Study of vibration severity assessment for Machine Tool spindles within Condition Monitoring

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Abstract

Today, machine tools are indispensable for production of manufactured goods. Several industries rely in this equipment to manufacture finished products by removing material through different cutting operations. Automobile, military and aerospace are just examples of industries where machine tools are used intensively. Today these industries strive for higher precision, narrower tolerances and more productivity in order to develop higher quality products using lesser resources and minimizing the impact on the environment.

Condition Based Maintenance CBM program has been proved as an effective preventive maintenance strategy to face these challenges. Reduction in downtimes, operation losses and maintenance costs are some of the benefits of adopting a CBM approach. The core of a CBM is Condition Monitoring CM, which refers to the surveillance of a suitable parameter for assessing the need of maintenance tasks in the equipment. These parameters are later compared with reference values to obtain a machine health assessment.

In machine tools, vibration level in the spindle units is considered a critical parameter to evaluate machine health during their operational life. This parameter is often associated with bearing damage, imbalance or malfunction of the spindle. Despite the importance of vibration levels there is not ISO standard to evaluated spindle health. This fact obstructs in some extend the planning of maintenance task for these high precision assemblies.

In the first part of this work, spindle components are studied and their function explained. Besides the main sources of vibration are listed, putting emphasis in three due to is importance when measuring vibration within condition monitoring of spindles. These are imbalance, bearing damage and critical speed. Later relevant concepts of vibration technology and signal analysis are introduced.

In the experimental part of the present study, controlled experiments were carried with the purpose of understanding which factors affect vibration measurements on the spindle housing. Control variables as spindle speed, accelerometer’s angular location, and spindle position were studied. Finally, a contactless excitation device CERS and its potential for industry in detecting bearing damage, is evaluated with two experimental setups.

The results indicate that vibration levels measured spindle housing depends on great extent on the angular mounting position of the accelerometers. Results also show that some vibrations severity indicators vary considerably along spindle speed range. It was also found that CERS could be potentially used on condition monitoring of machine tool spindles for detecting onset damage on bearings. However further research is considered necessary for this purpose.

Keywords: Machine tool spindle, vibration severity, vibration signature, condition-based monitoring, vibration standards, bearing damage, CERS

Sammanfattning

Idag är verktygsmaskiner nödvändiga för produktion av tillverkade varor. Flera industrier förlitar på detta utrusning för att tillverka produkter tack vare bortskärning av material i olika bearbetningsoperationer. Bild, militär och flygg-industrin är exempel av industrier där verktygsmaskiner används intensivt. Idag strävar dessa industrier för hög precision, trängre toleranser och mer produktivitet. Allt detta för att utveckla högkvalitet-produkter med mindre resurser och för att minska miljöpåverkan.

Tillståndsbaserdat underhåll (Condition Based Maintenance CBM) program har bevisats som en effektiv preventiv underhåll strategi för att bemötta de nämnde utmaningar. Minskning av stopptider, slöseri i produktion och underhålls-relaterade kostnader är flera av fördelar med implementering av CBM synsätt. Kärnan bakom CBM är tillstånd övervakning (Condition Monitoring CM), vilket hänför till bevakning av en lämplig indikator för att bedöma underhållsbehov av maskinen. Dessa parameter jämförs i efterhand med referensvärden för att inhämta en bedömning av maskinens hälsa.

I verktygsmaskiner, vibrationnivår i spindlar anses som en kritisk parameter för att utvärdera maskins-hälsa under dess operativt liv. Dessa parameter är ofta sammankopplad med lagerskada, obalans eller funktionsstörningar i spindeln. Trots dess betydelse, vibrationnivår i verktygspidlar är inte reglerad i form av ISO standard. Detta försvårar planeringen av underhåll för spindlarna.

I första del av den här arbete, spindels olika komponenter beskrivs och dess funktion förklaras. Dessutom de vibrations huvudsorsaker listas, med fokus på tre viktigaste som är relaterad till tillstånd övervakning of spindlar. Dessa är obalans, lagerskada och kritiska varvtal. Sedan viktiga begrepp inom vibrations mättteknik och analys introduceras.

I den experimentella delen av arbetet, kontrolerade tester utfördes i avsikt att förstå vilka faktorer påverkar vibrationsmätningar i spindelhuset. Kontrollvariabler som varvtal, acceloremeter vinkelställning och spindel positon undersöktes. Slutligen, utvärderas ”contactless excitation responses system” CERS samt sitt potential för att upptäcka lagerskador. Detta utfördes med två arrangemang.

Resultat indikerar att vibrationsnivåer påverkas i stor utsträckning av accelerometers vinkelställning. Resultat visar också att några vibrations indikatorer varierar betydligt med spindelns varvtal. Det konstaterades också att CERS skulle kunna användas för tillståndsövervakning av verktygspindlars med syfte av upptäcka skador i spindel lager. Däremot mer forskning behövs i detta riktning.

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Introduction to the thesis

The present work may interest different readers e.g. professors, engineering students and maintenance professionals. Therefore in order to facilitate the navigation and reading, the motivation of every section is briefly explained.

Section 1: Introduction. This introductory section gives a background behind this thesis work. First here is explained and supported why cutting machine tools are important in modern society and the reason of why spindles are critical component in machine tools. In subsection 2.3 a brief insight about different exiting modern maintenance strategies and the reason and why justified the condition monitoring of the spindle, using vibration techniques. In the following subsection a brief overview of two important vibration standards related to vibration is spindle is given. Finally in this chapter the research questions are introduced as well the limitations.

Section 2: Machine tool spindles. This section is the core of the thesis work and is aimed at introduce spindle technology and give an understanding about vibration on machine tool spindle. First industry applications, different spindle configuration and their use, are described. Later the main components of a spindle are listed and their function explained. This in order to understand the complexity and precision which spindle assemblies are required to perform well. Later, the main vibration sources of the spindle are briefly explained and categorized. This and the later subsections in the chapter are the key for understanding the relevance of the experimental part developed later in this work. Afterwards concepts as mass, stiffness, damping and dynamic stiffness are explained to understand their influence in spindle dynamics. Finally an overview of condition monitoring in spindle bearing and characterization of spindle is given, based mainly scientific articles. Their results and limitations in relation to the industry are also discussed.

Section 3: Vibration technology and signal analysis. This section is written to the readers which are not familiar with vibration technology and signal analysis. And it is intended to give the reader the basic concepts to understand the experimental part of this work. This is obviously focusing in mechanical vibration within rotatory machinery. The type of sensors available in the market, the mounting techniques and other instrument needed for vibration measurements. In the second part of this chapter, a brief overview of signal analysis is given, with emphasis in the important aspects when sampling the signals and different techniques for signals analysis used as FFT

Section 4: Experimental setup this part is dedicated to the experimental part of this work. Including methodology, instruments, results analysis and conclusions.

INTRODUCTION

1.1 Machine tools importance in modern society

Until today machine tools are considered the production equipment par excellence in modern manufacturing. They are the backbone in many industries and are capable of produce complex parts made in tough material as e.g. CFRP, titanium, and super-alloys. Most of these parts will become components in other subassemblies and subjected to demanding working conditions as high temperatures, large stresses and cyclic loads. Machine tools through the years have ensured specified dimensions and surface quality in order to withstand these requirements.

In order to comprehend what the concept *machine tool* comprises, a definition of this equipment may help. The Association of the Machine Tool Industries (CECIMO) which defines metal machine tool as:

> A power driven, not portable by hand, powered by an external source of energy, designed specifically for metal working either by cutting, forming, physico-chemical processing, or a combination of these techniques". (CECIMO, 2011, p. 11)

From the above definition it can be noticed that machine tools do not exclusively manufacture parts by a material cutting process, but also by forming and even aided by physco-chemical mechanisms. Hence it the three main categories of machine tools are identified: forming, cutting and physico-chemical based. The last group of machine tools are often associated with non-traditional machining process or NTM, including ultrasonic, water jet, electrochemical etc. Forming involves (but not limited to), bending, rolling, forging and extrusion machines. The present work will focus on the cutting machine tools ones as the one shown in Figure 1 and will refer to them as simply machine tools. Example of such widely used machines are turning, milling, and grinding, boring among many others.

Figure 1 Example of a cutting machine tool and its interior: CNC Turning machine. [DGM Mori Seiki]

Despite the rapidly growth of emerging manufacturing technologies, machine tools for cutting process are an stablished technology and seemingly with no replacement in the medium term. Additive manufacturing AM (3D printing), for example have been considered by many as a potential competitor to conventional machining processes. In 1996, AM was an industry worth 295 million, but by 2014 had growth 1300% achieving 4.1 billion (Wohler's Associates, 2015). As (Jun, 2002, pp. 741-742), points out AM compared to conventional machining have advantage as complex-shapes manufacturing in one setup and little production planning. Jun highlights also the advantage of cutting machine process in relation to AM mentioning accuracy, precision, excellent surface finish and mechanical properties of final products. The mentioned drawbacks of AM and the continuous improvement of machine tool technology, prevent AM become a substitute of machine tools in the short-medium term.

Today machine tools play an important role in the global economy. According to a survey (GARDNER Business Media Inc., 2015) which includes the 27 major countries that produce metal cutting and forming machine tools and covers 95% of the total production, the forecast of world machine tool production was 81.2 billion USD in 2014, see Figure 2. This figure suggested that the increase of machine tool production in the last decades is mainly due to industrialization in Asiatic continent.

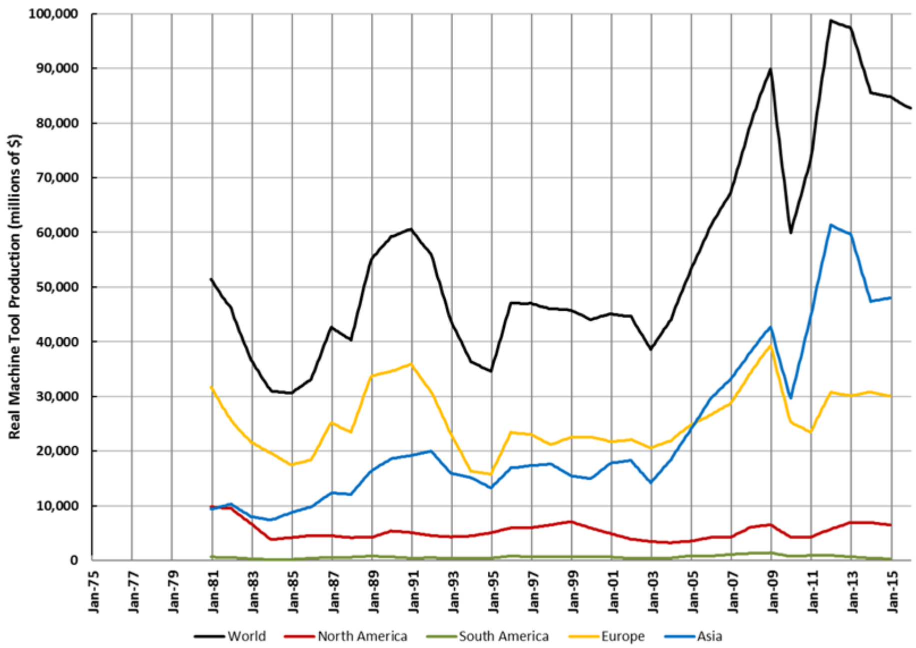


Figure 2 World machine tool (cutting and forming) Production. 2014 is estimated (Gardner, 2015)

Globally, China has become the largest producer of machine tools, surpassing Japan in 2010 (Masao, 2011). The following places in a top10-list are 2t^h Germany, 3^th Japan, 4^th South Korea, 5^th Italy and 6^th United States, 7^th Taiwan, 8^th Switzerland, 9^th Austria and finally 10^th Spain . China and South Korea have become strong suppliers of machine tool with significant growth in the last 30 years. In 2014 South Korea became number four in the world exceeding Italy (GARDNER Business Media Inc., 2015, p. 3).

Within Europe, machine tools accounted for roughly half of world machine tool production which equals almost 23 billion euro (26 billion dollar) in 2014 (CECIMO, 2015) and expected to growth 3% in 2015. Within the European production of machine tools, Germany and Italy together in 2010 accounted for 66,5% of production. In these two countries, machine tool manufactures consisted in several small to medium-sized enterprises concentrated in South Germany and North Italy (CECIMO, 2011, p. 15).

One would deduce from these figures that the traditional machine tools manufacturer (meaning German, Japanese, Italian and American producers) have seen their position in the global market, threatened by new competitors (Chinese and Korean machine tool manufacturers) in last 30 years.

Because there are not supply without demand, knowing the consumers of machine tools can give a more comprehensive picture of interaction in this industry. Interestingly most of the larger machine tool producers are also the larger consumers. Seven from the top-ten producer list are also in the to-10 consumer list according to (GARDNER Business Media Inc., 2015, p. 4). In this order 1th China, 2th USA, 3th Germany, 4^th Japan, 5^th South Korea,

6^{th} Italy, 7^{th} Russia, 8^{th} Mexico, 9^{th} Taiwan and 10^{th} India. Noteworthy is the fact that Sweden is in 25^{th} place.

Sweden holds limited influence as a customer and producer in the global market of machine tools. It accounted in 2010 for only 1.1% of the total machine tools production in Europe. In fact, in 2014 the imports of cutting machine tools (1.5 billion SEK in 2014) are approximately two times larger than the exports (Statistiska Central Byrå SCB, 2015). Based on these facts, Sweden can be considered minor consumer and producer of cutting machine tools in the global market.

Previous figures confirm the economic importance of machine tools considering its monetary value. For this reason one should consider that machine tools, as other any other production equipment, are assets to their owners. However, an additional crucial characteristic reveals importance of these assets: their capacity to produce other machines or other productive assets, including themselves. For this reason they are often referred as "mother machines". Besides machine tools are also present in the life cycle of almost all the common goods one sees daily. In the way that most of the equipment utilized for manufacturing those goods, were produced thanks machine tools.

As any other assets, machine tools generate value-added for companies but also create wealth for countries. In particular machine tools are used to fabricate highly manufactured products and parts such as turbines, gears, metallic frames, and machine components to name some (see Figure 3). Automobile, military, energy, aerospace, medical, oil and refinery are just examples of industries where machine-tools are used intensively nowadays.

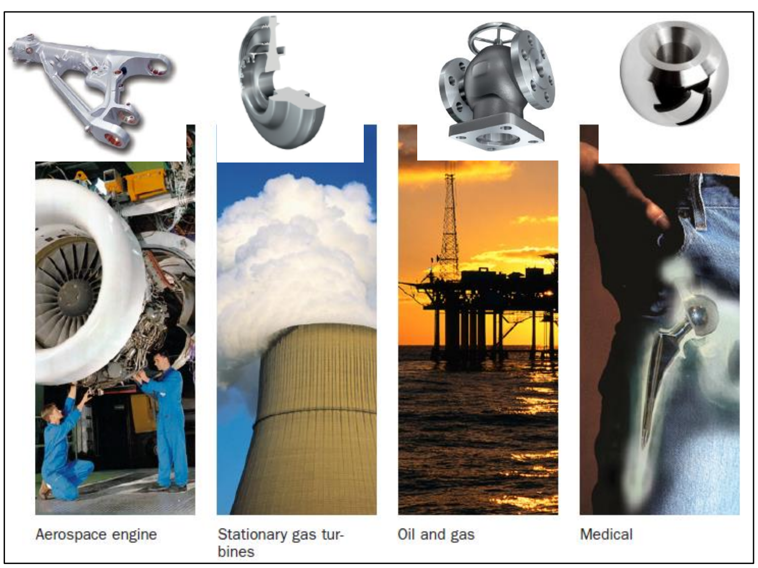


Figure 3 Some components manufactured using machine tools with application in different industrial sectors. From left to right: Landing gear, gas turbine disk, valve housing and femoral cap [Courtesy Sandvik Coromant]

1.2 The spindle unit: the heart of a machine tool

Machine tools are composed by several subassemblies where the spindle unit is an important structural component located in different position, depending on the machine tool configuration. In milling machines, for instead, they are mainly used in vertical position, and they hold the tool holder which, as its name indicates, sustain the tool. See Figure 4. Unlike milling machines, in turning machines spindle are horizontally placed and hold the chuck, which in turn holds the workpiece. However, many recent machining centers, capable of achieving multiple cutting operation, have swivel spindles that can work not only vertically and horizontally but in almost any angled position.

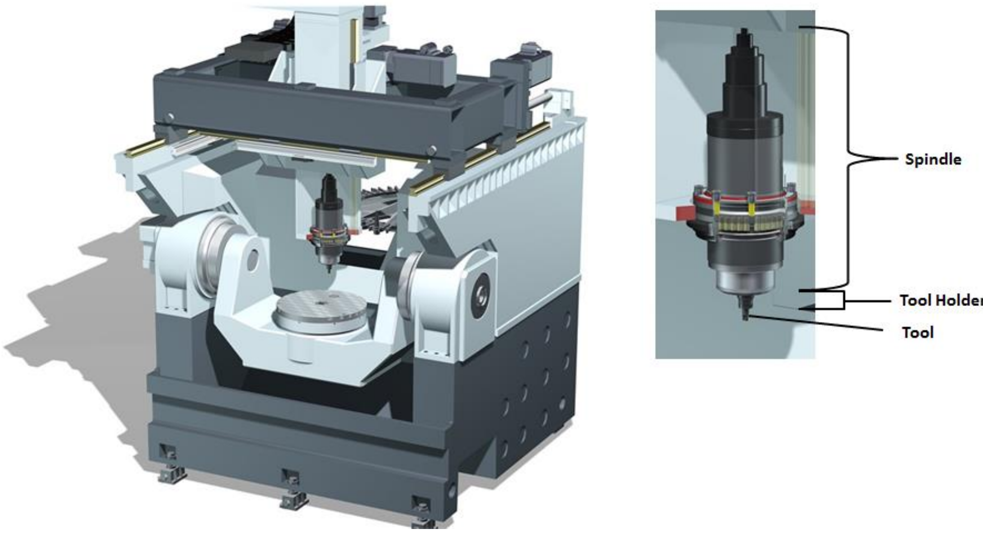


Figure 4 Position of the spindle in a milling machine adapted from [Jakobantriebs technik]

The spindle unit play an important role in the functionality and performance of a machine tool. This is well expressed by (He, 2015, p. 2) who states “The reliability of the spindle assembly influences the whole manufacturing effectiveness and stability of equipment”. Technically speaking, as (Abele, et al., 2010, p. 781) has synthetized, machine tool spindle accomplish two important functions: First, to give accurate rotational motion to the tool (e.g. drilling, milling and grinding) or part (eg. turning). And secondly, transmitting energy to the cutting zone necessary for material removal.

During machining the spindle is subjected to demanding load and thermal condition. Cutting forces loads transmit, in form of static and dynamic, through the tool system (tool holder& tool) to the spindle bearings. All this when the spindle is rotating at high speed and spindle/tool system are required to machine with required precision from 1µm (eg. grinding, boring) to 100 µm (eg. turning, milling) (De a Calle & Lamikiz, 2009, p. 11)

Problems in spindles are one of the main sources of machine tools downtime in manufacturing industry. In a German study (Fleischer, et al., 2008, p. 175) where maintenance information were gathered from 250 machine tools in the automobile manufacturing sector, four major subsystems being responsible most downtime where identified. Within them, spindle and tool changer accounts for 26% of downtime within these four subsystems, being the second major cause after drive axes as figure Figure 5 shows. Similar results are shown by (Häggblom, 2013, p. 33) in a work carried out in cooperation with the main Swedish-based car manufacturer. The study is based in analysis of maintenance data collected during two years from 15 machine tools of a production line. Within downtimes longer than three hours, problems related with spindle &tool changer accounted for approximately 21% of the total. This can be estimated from the information shown in the graph considering "magazine", "spindle cooling system" and "spindle/chuck". This two examples support what Neugebauer states about the negative effects of spindle failure within production systems: these failures "inevitably incurs idleness and high cost for repair and production losses." (Neuebauer, et al., 2011, p. 97).

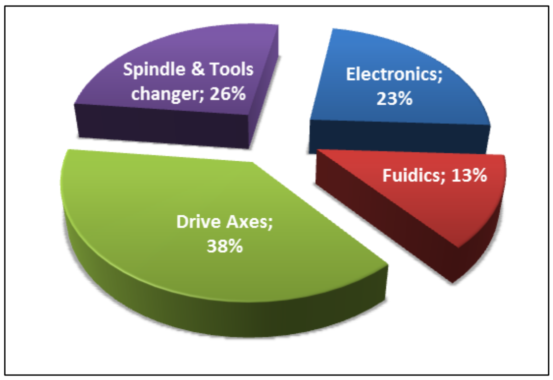


Figure 5 The top four major assembly components in machine tools that leads to downtime (Fleischer, et al., 2009)

Another example even clearer about how critical the spindle is as machine tool component is illustrated within the aerospace industry. In the context on an recent study (de Castelbajac, 2012, p. 27) carried out in the French aerospace industry which covered 130 motor spindle (of same model), the mean time to failure was randomly spread with a mean value of 2500 hours. What is remarkable is the fact that spindle manufacturer had specified 6000 hours for this model. In the study was part of a program where one of the goals was to increase competitively in the industry by decreasing cost during operation and maintenance. Important companies where involved, in the mentioned work including Dassault, Airbus among others.

Due to the mentioned high cost from spindle-related downtimes, it is it common practice in the manufacturing industry to have a spare spindle for critical machine tools in the production system (Quintana, et al., 2009, p. 109) (Archenti, 2015). By this way, the failed spindle can be rapidly replaced and sent to a spindle reparation shop, minimizing mean-time to repair and consequently the cost for non-production. In fact, today supply and delivery of spindle can take several months (Neuebauer, et al., 2011, p. 98).

Considering machine tools as productive assets, they are expected to function at their maximum capacity, avoiding costs in unplanned reparation and production losses due to downtime or quality problems. Because of this, maintaining them in optimal condition is a priority in highly productive production systems. This can be achieved by a preventive maintenance strategy of critical components of the machine tool. With this strategy maintenance task can be planned in advance based on the real maintenance necessity the equipment. For this reason, a well design maintenance plan for spindles that includes condition monitoring become important to ensure productivity, decrease downtimes which will result in a reduction of the life total cycle cost of machine tool.

1.3 Condition Based Monitoring CBM of spindles

The need for monitoring spindle condition can be comprehended by understanding the fundaments of Reliability centered maintenance RCM where Condition Based Maintenance CBM is integrated into. RCM is a methodology to select a suitable maintenance strategy depending on how systems fail (failure modes) and in which extend this failure affects the function of the overall system (consequences). It is widely agreed that a fail occurs, when the system is incapable of delivering the required function at required performance (Mikler, 2014, p. 16)

According (Smith, 1993 cited in Tsang, 1995, p.6) RCM has as a main goal "to preserve system function". However this goal should be accomplish at reasonable cost. Under this point of view, it is not justified to expend more resources for maintain the function of a system if these resources exceeds the benefits of achieving so. It is also important to know that the term system refers to a plant or equipment where other parts/components interact jointly to accomplish a determined function.

Maintenance strategies which RCM includes are corrective (reactive), preventive (schedule based) and Condition Based Maintenance CBM. This methodology should be selected after carrying out a seven-step analysis in a RCM methodology. By this way, CBM strategies are often considered for failure modes which can produce outrage of the system or economical costs (Tsang, 1995, p. 7). However these failure modes must be also detectable. Once the failure mode is identified and studied, a suitable parameter using a specific technique is monitor in order to estimate the need of maintenance task on the equipment.

The detectability of the failure modes will depend whether the components, associated with the failure mode, present evidence deterioration/wear or not. Example of this type of failure modes are fatigue, corrosion, erosion, pitting. This is important because, it is common knowledge within maintenance that some failures never present evidence of wear-out or degradation on the component. These are known as random failures and are more frequent. Several studies (NAVSEA, 2007, pp. 2-3) carried out in aeronautical industry, show that random failures accounted for 77% to 93% of the failures while wear-out associated failures represent between 6-23% of all the failures. This means that a nearly a quarter of failures can potentially be avoided implementing CBM strategy.

Within CBM, there are several monitoring techniques to surveillance the deterioration of the components towards an imminent fault. These include oil analysis, thermography, ultrasound, vibration among others. Each of these techniques has different capabilities detecting onset failure in advance. For example, as Figure 6 shows, faulty machines may show increased vibration before contaminating particles on the oil are detectable. On the other hand, ultrasound techniques allows detecting these problems at even an early stage. For rotatory equipment, as machine tools, mechanical vibration analysis has been the one of the most accepted and used because its versatility and reliability. As (Tsang, 1995) points out vibration monitoring allows detecting deterioration as wear, imbalance, misalignment, fatigue in parts under reciprocating or rotational motion.

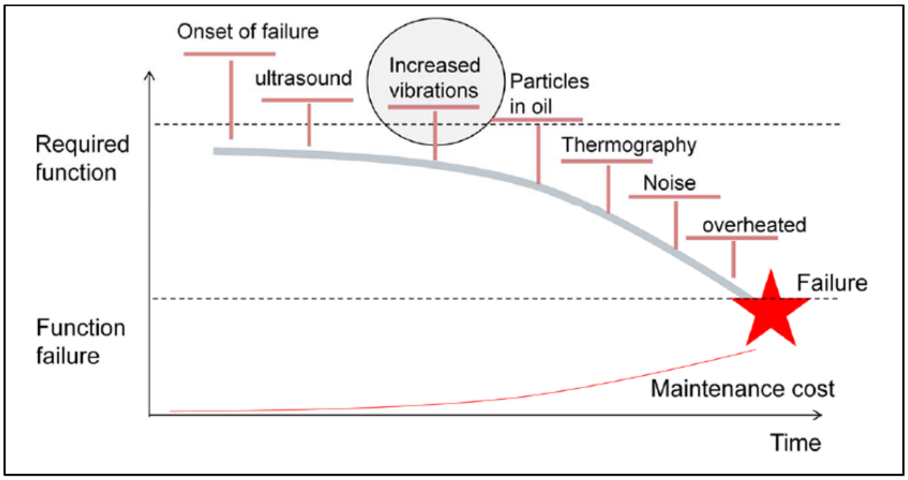


Figure 6 faulty signs of equipment until failure. (Hagberg & Henriksson, 2010 cited in (Mikler, 2014))

Regarding machine tools spindles, depending on the failure mode aimed to prevent, several monitoring techniques can be suitable from the economic and technical point of view. As (Neuebauer, et al., 2011, p. 97) highlights, problems in bearings is one of the main responsible in spindle failure, together with failures associated with clamping devices and rotatory union. The authors suggest that mechanical vibration (acceleration) and acoustic emissions measurement on bearing housing together with temperature monitoring of the bearing outer ring, may be suitable parameter to monitor bearing condition. Failures in other components, as clamping device and the rotatory union, can be detected using other sensors as shown in Figure 7.

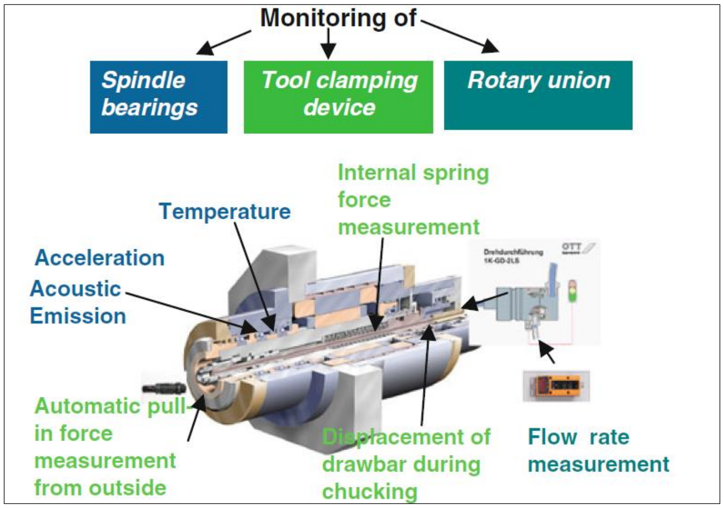


Figure 7 Used sensors for spindle monitoring (Neuebauer, et al., 2011)

Condition monitoring can be carried out in periodically or continuously depending on the critical role of the equipment in the production system (Ramachandran, et al., 2010). However the trend today seems to be in incorporating more sensors on the equipment for condition monitoring on real-time basis. Still many industries still carry out this measurement in fixed schedule, usually within week or months following the maintenance plan of the machine. Figure 8 shows a typical schedule for CBM where vibration is the condition parameter being monitored every a fixed time-lapse. Especially important is to carry out measurements in the equipment at its initial condition, unused.

These initial values measured in a CBM plan are often referred as the machine's "fingerprint" or "vibration signature". By this way, vibration signature represents the machine, in its closest state to the design. Therefore it provides a good baseline to evaluate the machine during its lifespan. For instead, when problems on the machine appear "its signature is modified" (Bóden, et al., 2014). These problems can include unbalance, bearing damage, motor problems, among many others. Depending on their severity can lead to an imminent failure (or breakdown) of the system (see Figure 8). As the figure shows in the vertical axis, the higher the vibration levels, the higher the probability of failure. For this reason, once a trend has been observed, it becomes important to increase the frequencies of later condition measurements in order to follow closer the development of these changes. If not maintenance task is carried out on time, this can lead to breakdown. In contrast, if the proper corrective maintenance is implemented, surely the machine will return to a condition close to its new condition with lower vibration levels, as observed in the figure.

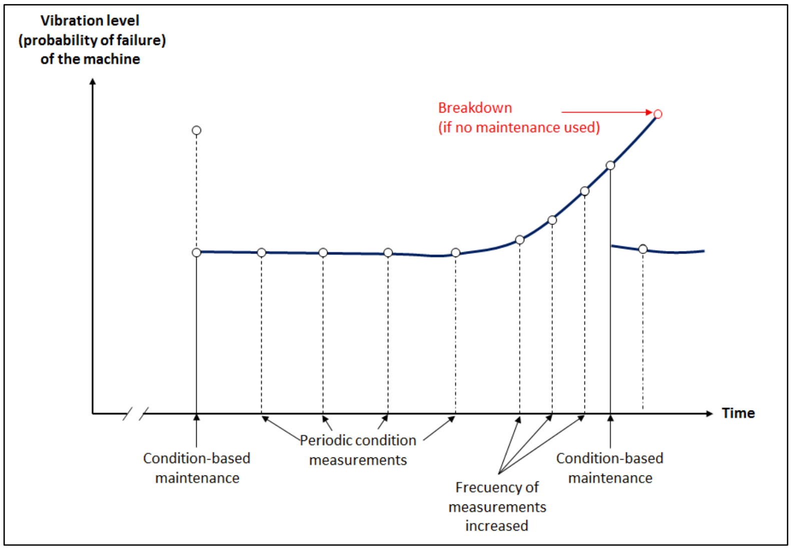


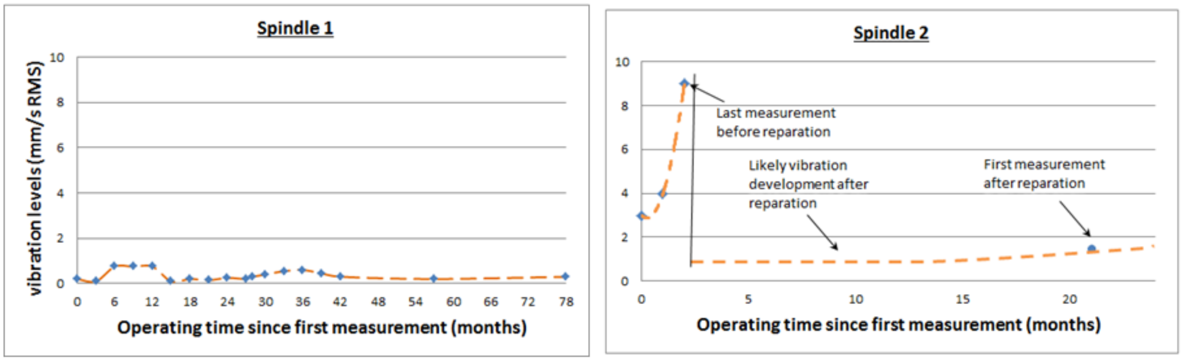
Figure 8 Condition Based Maintenance schedule (Rao, 2004)

One of the purposes of CBM is to “predict deterioration rate towards failure” (Mikler, 2014, p. 2). By knowing the remaining life of the component about to fail, maintenance task can be planned in advance preventing breakdowns and damage in other components. For complex system as spindle assemblies, determining accurately the remaining life of a spindle may be difficult.

Hopefully it is widely agreed that increasing vibration levels in machinery is associated with problem in machine components (misalignment, looseness, unbalance, cracks, bearings damage etc.). In other words, higher vibration levels, express worse condition or even damage of the machine tool. Machine tool spindles are not the exception to this

Empirical evidence from industry suggest that vibration levels may develops differently even for similar group of spindles and these vibration developments are strongly influenced by the initial vibration levels. In other words, higher vibration levels may be an indicator of shorter spindle life. While low vibration levels may vary moderately, yet remaining low under extended time period at normal operating conditions.

To illustrate this idea, real vibration measurements of a spindle are shown in Figure 9b. The data shown corresponds to vibration measured in the back end of the spindle housing in a milling machine with maximum operating speed of 6000 rpm. The initial vibration measurements are low and remain in this way for an extended period of time (about 4.5 years). On the contrary, as shown in Figure 9a, vibration measurement were carried out on a spindle housing in another milling machine (same max. speed as the former), shows rapid deterioration, when initial vibration levels are high. This sudden deterioration (within two months) may have led to breakdown if the spindle had not been sent to a repair shop. This example resembles Figure 8 (shown earlier) in several aspects: First increasing vibration levels indicate problems, which justifies spindle’s shipment for corrective maintenance. Secondly, after reparation the spindle showed comparatively lower vibration levels than initial values, indicating an improved condition. This is reflected in vibration levels after reparation.



a b

Figure 9 Vibration measurements for different spindles. Adapted from (ISO TC39/SC 2, 2011, pp. 6,17)

The question that arises by these two examples if it is possible to stablish absolute vibration limits for similar spindles in order to diagnose accurately spindle condition and avoid unexpected breakdowns. In fact, the graphs shown were based in data presented as supporting evidence for stablishing vibration limits for a specific group of machine tool spindles (ISO TC39/SC 2, 2011). This information was presented within the debate regarding standard proposal that resulted in the Swedish standard SS-728000-1, which will be discussed later in this work. In addition, the vibration data was collected within CBM programs among several Swedish industrial manufacturers including two important as Sandvik Coromant and Scania. The data corresponds to vibration measurements carried out on the front and back end of the spindle housing, nearby bearing locations.

The data collected in the mentioned report provides valuable information from existing machine tool in the industry, but interpretations based on this data should be formulated with prudence. Some measuring parameters which may influence vibration levels are not specified in the report. On the other side, the vibration tendency over time and its direct relation with spindle problems is rather clear and supported by literature.

One drawback when using vibration levels for assessing spindle condition is that they are often speed-dependent. One example of this is that some spindles may present high vibration levels (or peaks) at specific speeds along the operating speed range. These are known as critical speeds and are expected due to dynamic characteristic of the spindle. Despite their known existence, both spindle and machine builders do not recommend operating at these speeds because it can lead to damage on the spindle, in particular on the spindle bearings. In other words, even though high vibration level at critical speeds may not reflect the "truth condition" of the spindle, operating at these speeds can deteriorate its current condition.

It is interesting to observe that those peaks can differ greatly even among spindles with similar design characteristics. These differences in vibration levels are reflected in magnitude, critical speed location and “width of the peak”. One example to illustrate this is shown in Figure 10. The four cases correspond to motor spindles with a maximum operating speed of 25000 rpm. In most of them, the first critical speed appeared between 15.000 and 20.000 rpm. However the magnitude and extension of these peaks over the speed range differ significantly among them. For example, as in Figure10a this peak is rather sharp and high (about 3.5 mm/s rms), in spindle (c) seems to increase gradually even beyond the maximum operating speed and reaching a maximum of about 2.8 mm/s rms. Case d in Figure 10 is especially interesting because illustrate how these peaks can affect machine tool customers and how these peaks can be modified after proper measurements. According to the report (ISO TC39, 2011, p.31) the vibration levels represented by the dotted line corresponded to initial vibration levels of the spindle. The machine tool owner needed to operate the machine near to maximum speeds but because of the high vibration in this area, it was impossible. Therefore, the owner required the machine tool supplier to solve this issue. The solution was to improve the spindle balance quality after a procedure called field balancing. This resulted in vibration levels noticeable lower than the previous, which are shown in solid line.

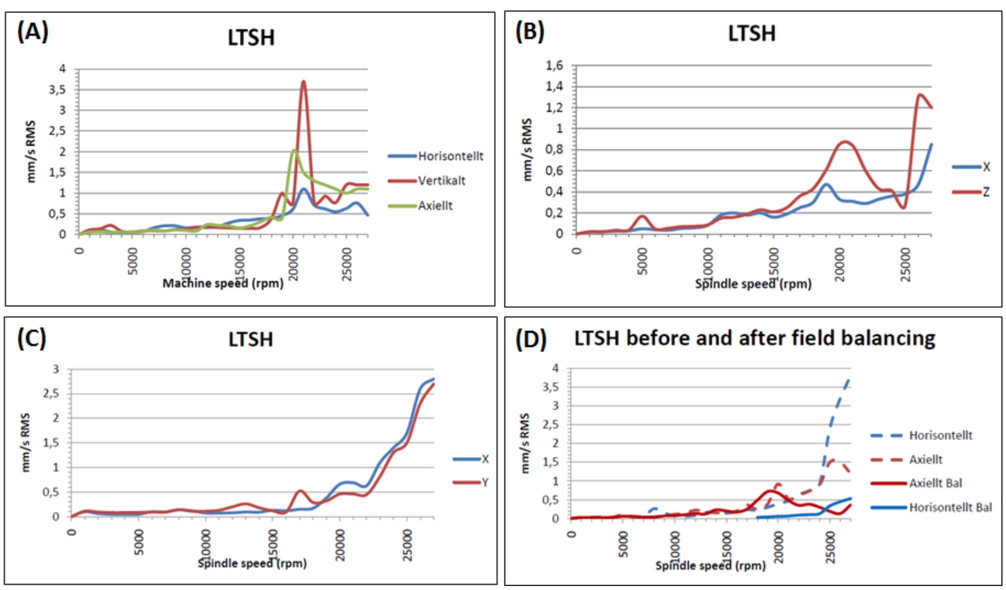


Figure 10 Vibration levels along the entire speed range for four different motor spindles (ISO TC39/SC 2, 2011).

In short, condition monitoring based on vibration measurements are a widely used technique within preventive maintenance of machine tools spindle. Increasing vibration levels are often attributed to a deterioration of the spindle condition.

Vibration peaks at critical speeds are undesirable because they prevent machine owners to take advantage of the full speed range of the spindle. This is because operating the spindle at these critical speeds can diminish its life by exposing spindle components to non-design working condition. In addition, high vibration levels associated to critical speeds can varied significantly even for spindles with similar design characteristics as mentioned earlier.

A major concern is, therefore, stablish reasonable vibration limits may be used not only as monitoring references but potentially as an acceptance mechanism to assess the new spindles. Machine tools consumers have become more demanding in the last decades because of the need for high productivity as a consequence of the increasing global competition. Thus machine tool availability demand has become a driving force for technology development of machine tools builders.

The difficulty of stablishing fixed vibration limits is due to the fact the existence of different indicators and interpretation of results is not unique. In the next section, two important standards related to spindle condition are briefly reviewed to acquire better understanding of this concern.

1.4 Guidelines and Standards for assessing spindle condition

There are countless techniques and criteria for evaluating condition on rotatory machinery by vibration techniques. They differ mainly in the way the vibration signal/data is analyzed and interpreted. (El-Thalji & Jantunen, 2015, p. 235) implies that there are simple and advance signal processing techniques for performing condition monitoring and diagnosis of machines. The simple techniques are preferred in industry and they includes as root means square (RMS), kurtosis, Fast Fourier transform, crest factor, cepstrum, correlation functions, among others. Advance signal processing techniques include wavelets, fussy etc. Some of these process techniques are explained in section 3 of this work.

When using simple signal processing techniques, severity guidelines are practical tools to interpret machine condition, based on characteristic values of the signal. For example, peak, rms to name a few. These guidelines were the first tool developed to assets the condition of a machine. The first vibration severity guidelines are generally attributed to American engineer Rathbone in the 30´s. He developed these guidelines based on his professional experience in turbo machinery and wrote a paper titled "*vibration tolerances*" where his introduced the basic concepts used until today (Mitchell, 2007, p. 2). Today these guidelines have been refined but they are based on the same principles Rathbone developed.

Today vibration severity guidelines, as Rathbone's, are often built based on experience and cover general rotatory machinery. These guidelines have been created by different stakeholders, including associations, machine manufactures, standard institutions, vibration equipment suppliers among others stakeholders. In (Cease, 2011), an overview of existing criteria for guidelines is presented. In the present work, two of these severity guidelines, relevant for machine tool spindles, are briefly described in the appendix.

This large number of guidelines to evaluate machinery condition based on vibration measurements creates a problem: results, interpretation and thus condition assessment may differ even for same machine. This is because these guidelines use different indicators as (peak-to-peak, rms, crest factor, etc.) to evaluate severity, furthermore they also stablish different vibration limits. This makes almost impossible to compare different judgments or determine their equivalence.

Another issue with severity guidelines is that most of these guidelines aim at rotatory equipment with differ greatly with machine tools. Two main difference are identified: firstly, what differ machine tool spindles from other rotatory equipment (e.g. pumps, blowers, generators, fans), is the fact that machine tool spindles are high accurate assemblies, which must have low vibration levels to ensure precise machining on the workpiece. This low vibration requirement may require a more detailed specification when measuring and assessing spindle condition based on vibration levels and therefore it might be inappropriate to apply a general criterion. Secondly the majority of the rotatory equipment these guidelines are aimed at, work at an specific nominal speed. This is not the case of machine tool spindles, which are expected to perform precisely within the whole speed range in order to adjust cutting parameters based mainly in the workpiece material characteristics.

An alternative to vibration severity guidelines provided by individual actors, are standards, which are respected mechanism to unify different point of view of different actors within a specific technical field. This it is particularly truth within international standards published by The International Organization for Standardization ISO. This standardization authority works in parallel with member bodies, which are standardization institution worldwide and represent ISO in a specific country. For example, this is the case of DIN in Germany, SIS in Sweden or ANSI in USA (ISO, 2015a). Among these members, standards are proposed, developed and voted by a group of experts in a certain matter. These groups are referred as technical committee (TCs) and they represent different stakeholders interested in a specific area. There are more than 250 TCs within ISO covering almost all the industrial sectors including Agriculture, Building, Mechanical Engineering etc. Stakeholders include industry, ONGs, governments among others (ISO, 2015b). These committees holds annual meetings, when all the participating members in a TC, discuss and vote relevant standards proposals. Standards proposals required 2/3 of votes, among the P-members, to become ISO standards.

Standard are not perfect but perfectible tools aimed to be practical for the industry. Standards are discussed involving interested stakeholders, and then updated based in the technology development. For examples some standards are withdraw and replaced by improved standards which take into consideration variables that have not considered before, because with the former technology, they have not been an issue. Standards are also practical tools; this means that they may suitable for the industry based on the training and time required. Standards are good enough mechanism. For example, a very complicated standard taking into consideration all the variables will be difficult to read, interpret and apply and may require a lot of time, effort and maybe higher expertise, thus economical costs which may turn it in an impractical and expensive tool.

Despite the of spindle as a critical component in machine tools, there is not yet international standard that determines the vibration limits for evaluating their condition. This is because different ISO P-members have not reached an agreement about an optimal manner to measure vibrations and which will be the allowable levels they should be limit to (Stenmark, 2014). However is interesting, for the present work, to review two important standards related to machine tool spindles condition assessment.

1.4.1 ISO 10816

There are two main ISO standards which stablish evaluation for condition of rotatory equipment based on vibration levels. These are ISO 10816 and ISO 7919. These are improved version of the former standard ISO 2372 withdrew in 1995. The difference between ISO 10816 and 7919 is that while ISO 10816 is used to evaluate machine vibration by measurements on non-rotating parts, ISO 7919 is used by measuring vibration in rotating shafts (commonly by non-contact displacement sensors). The correct selection between the two standards will depend on whether or not is reliable to measure vibration on the shaft or bearing support (also called pedestal) of the equipment, depending on its dynamic characteristics. ISO has published a guide for select correctly between ISO 7919 or ISO 10816 depending on the equipment. This is described in detail within the technical report ISO/TR 1901:2013 (ISO, 2013).

Because vibration measurements in machine tools are often taken in non-rotatory parts, ISO 10816 seems more appropriate for spindles. However ISO 10816 is divided in several parts which every of those give guidelines mostly focus in turbomachinery (steam and gas turbines, compressors, generators, and fans) (ISO, 2009). Obviously this machine performs functions not comparable with the accuracy demand of machine tools. For this reason, this standard can be a seen as a very "rough" guide for defining vibration limits in machine tool spindles.

To exemplify the criteria used in this standard, a severity chart extracted from ISO 10816 is shown in Figure 11 . The parameter used to indicate severity is vibration velocity in rms (root mean squared). This is displayed in the right vertical axis of the chart. The classification of severity will be determined by the letters A, B, C and D. This is standard is aimed at give an evaluation criterion not only for "operational monitoring" but also for "acceptance testing".

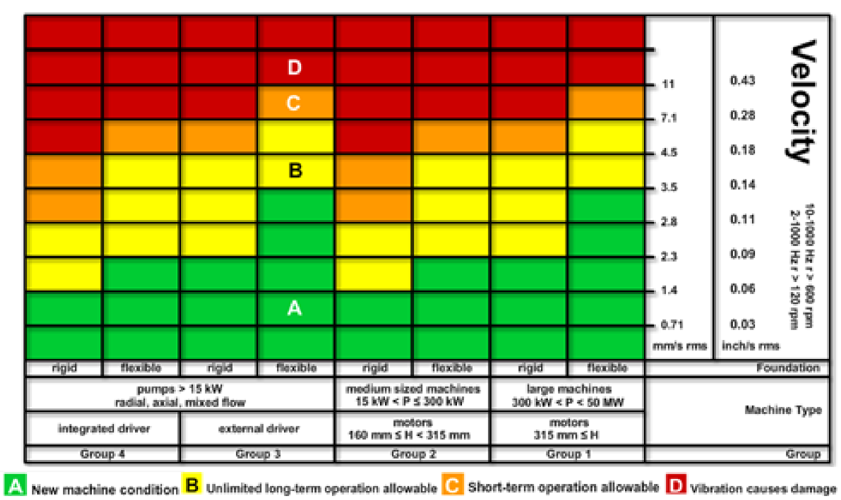


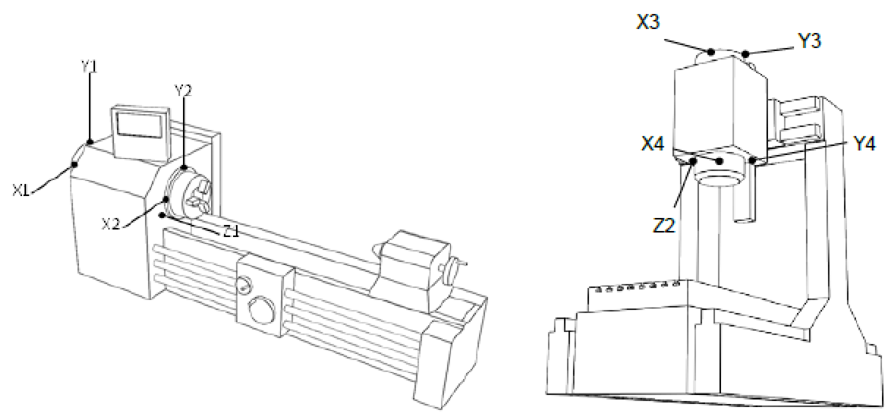
Figure 11 Severity Chart ISO 10816 Velocity (MOBIUS INSTITUTE, 2008)

1.4.2 Swedish standard SS728000-1/ (ISO TR 17243-1)

Sweden, as a machine tool consumer, has a Swedish standard since 2014, regarding machine tool spindle assessment based on vibration measurements. This standard is known as SIS 728000-1 and is titled:

"Evaluation of machine tool spindle vibration by measurements on spindle housing-part1: spindle with rolling elements bearing and integral drive operating at speeds between 600 min^{-1} and 30 0000 min^{-1} "

The standard is non-normative, meaning that it can be considered a recommendation but not a requirement. The Swedish Standard Institute published later, ISO TR17243-1 as a national standard denominated SIS 72800-1. This standard is today used in the Swedish industry for evaluating attributes of new spindles based on vibration measurements, but also for spindle condition during their operation life. All this by measuring vibration on the spindle housing as it illustrated in Figure 12.



Key

X1, X2, X3, X4 Preferred radial measurement locations in X-axis direction of machine.
Y1, Y2, Y3, Y4 Preferred radial measurement locations in Y-axis direction of machine.
Z1, Z2 Preferred axial measurement locations in Z-axis direction of machine.

Figure 12 Preferred vibration measuring position on spindles according SIS 728000-1

This standard, as Figure 14 shows, is similar to ISO 10816, in a sense that stablish four different condition zones depending on vibration level measured (A,B C and D). These zones are aimed not only to evaluate the spindle during its operating life but also served as acceptance criteria. In the same way as ISO 10816, the Swedish Standard 72800-1 stablishes that vibration measurements to be taken on the housing of the equipment (spindle in this case) as close as possible on the bearing housing, specifically at back and front end of the spindle. Nonetheless this standard refers, as the title suggests, to very specific population of spindles. Besides the standard is applicable to different cutting machine tool including: machining centers, milling, boring grinding and turning machines (SIS, 2014, p. 5).

Another difference with ISO 10816 is that in SIS 72800 vibration data is collected at different speed along spindle speed range. Besides, two parameters are used to evaluate spindle condition. These parameters are denominated Long Term Spindle Condition (LTSC) and Short Term Spindle Condition (STSC), which correspond to RMS (root mean square) value of vibration velocity and acceleration respectively. According to the standard, LTS represent only problems detectable at low frequencies (10-5000 Hz), associated with e.g. misalignment and unbalance. According to the standard, these problems can degrade spindle bearings in the long run. On the other side STSC represent vibration within high frequency spectrum (2000-10000 Hz) and therefore reflects better problems associated with bearing condition, which can develop rapidly (from days to six months).

In addition, the standard allows 10% the exclusion of vibration data up to two zones within the speed range of the spindle. However, this is allowed only on spindle in the range 6000-30000 rpm. The reason is that because its physical configuration, high speed spindle present usually one to two critical speeds within the operating speed range (SIS 72800) as it is shown in Figure 13.

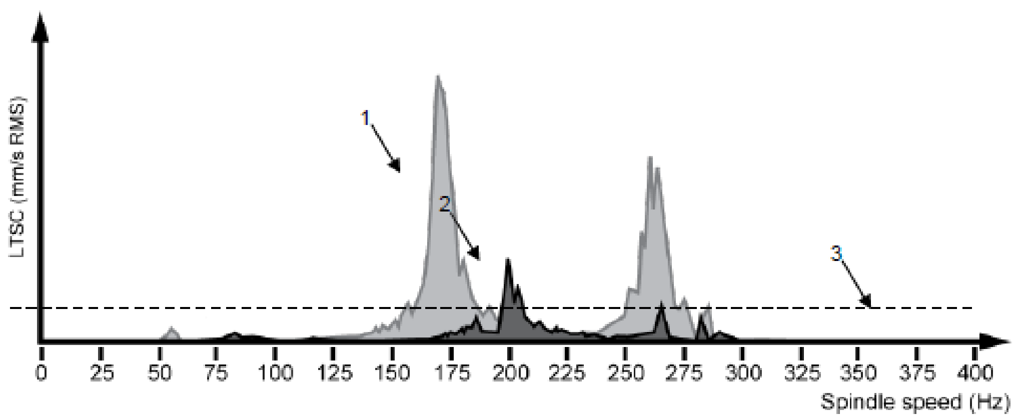


Figure 13 Exclusion criteria for LTSC according to SIS72800

Based on these two parameters measured along the operational speed range of the spindle the standard stablishes allowable limits as it shown in Figure 14. Worth noticing is that the table require the same vibration limits for LTSC regardless the maximum speed of the spindle, which is grouped in 4 classes (600<6000 to (18 000<30 000) rpm. On the other hand, when it comes to STSC, these limits are more “allowable” depending on the spindle maximum operating speed. This means that in the case spindle with higher maximum operational speed, higher values of STSC will be tolerated.

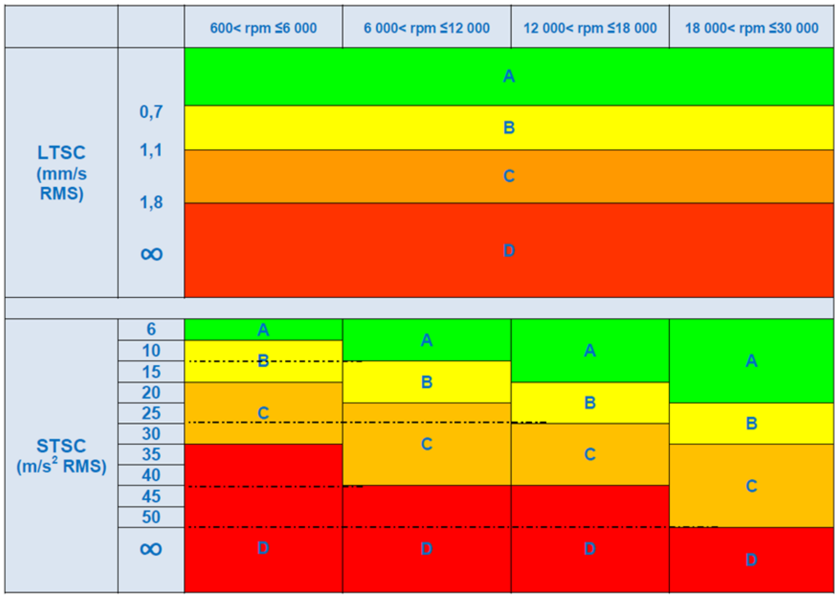


Figure 14 Table for evaluating vibration levels on motor spindles (up to15 kW) according to SIS 728001

It is important to understand that specifying absolute values can be controversial because they may be interpreted as unmovable rigid requirement to machine tool builders, the standards is quite flexible in this aspect. In fact it states that “Numerical values assigned to the zone boundaries are not intended to serve as acceptance specification”. And clarify that even when the spindle exceed certain limits specified on the standards, they can still be accepted for the consumer with necessary arguments of the machine tool supplier about these specific feature of the spindle that explain these deviations. In fact the standard states that these values “provide guidelines for ensuring that gross deficiencies or unrealistic requirements are avoided.” In fact the standard highlights that is intended to provide a useful tool to evaluate spindles.

1.5 Objectives and research questions

The main objective of this work is to contribute to the discussion on the topic, giving the reader clarifying some concepts and enhance the understanding of vibration on machine tool spindles. But also investigating some factors that may influence vibration measurements in machine tool spindles

After acquiring a basic understanding of the topic, motivation and background, is appropriate to introduce the relevant research question which the present work is intended to answer:

RQ1 Is there any relation between vibration fingerprint of a machine tool spindles and their lifetime? In other words, can high initial values of vibration be an indicator that a spindle wears out rapidly?

RQ2 What are the main sources of vibration in a machine tool spindles? Besides which of these sources are related with condition monitoring of the spindle?

RQ3 At which extent vibration measurements at the spindle housing represent the truth condition of the spindle?

RQ4 Which factors may affect the reliability of vibration measurements on the spindle housing?

RQ5 Is it possible to establish vibration limits for machine-tool for condition monitoring by measuring vibration in spindle housing?

RQ6 Do current standards provide and effective tool for condition monitoring of integral spindles in machine tools? Could they be improved further to meet this point?

1.6 Scope and limitations

This work focuses on machine tool spindles used in conventional machining processes within manufacturing industries as automotive, aerospace, and mould&die). Other spindles used in other industries are not treated here. For instead, micromachining or ultra-precision machining used in optical, photonics or telecommunication manufacturing industry. This is because they required much higher accuracy than conventional machining so different spindle technology is used (e.g. air bearings). However, they may be mentioned to illustrate the challenges of conventional machining when accuracy and speeds required increase.

Other topics surpass the scope and objectives of this work are:

- Torsional vibration on the spindle components
- Failures in the spindle that cannot be monitors by vibration techniques
- Spindle failure root causes and related corrective maintenance tasks
- Estimation of the residual life of the spindle

2 MACHINE TOOL SPINDLES

Machine tool spindles are used widely in a large range of industries as it seen in Figure 15. These industries include Automotive, Aerospace, Mold and Die, and Electronics among many others. Different industries carry out diverse machining operation at different stage of the manufacturing process. For this reason spindle configuration (e.g. max. speed, power, accuracy) will depend on the requirements of the processes they are intended to carry out. For example, as it is shown in Figure 15, automotive industry require spindles up to approximately 17,000 rpm with high power (up to 150 kW) in order to perform roughing machining of power train manufacturing or cast iron components . In the same way aerospace industry utilizes spindles also with high power to machine high strength materials as Titanium: However they also require more speed capacity in order to achieve high accuracy in large Aluminum frames (Abele, et al., 2010). On the contrary, Electronics industry require low power (1-5 Kw) spindles but whit extremely high speed (up to 370,000 rpm) for Printed Card Board PCB drilling and routing (Wester Wind, 2013).

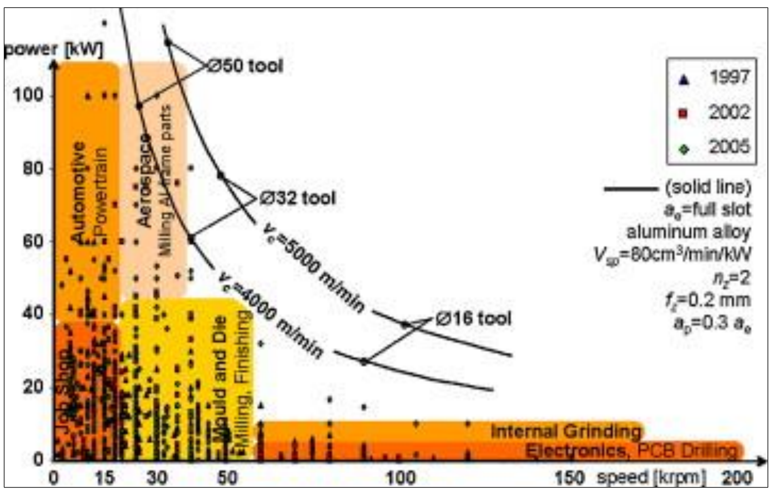


Figure 15 Spindles available in the market [Courtesy PTW] cited in (Abele, et al., 2010)

Power and speed range are decisive for shaping the capability of spindles, however they are not the only characteristics. Accuracy, vibration level, thermal behavior, torque and energy efficiency is also important. The transmission mechanism or drive system will limit importantly these characteristics. As Figure 16 shows, there are four main transmission drives for machine tool spindles: 1) belt driven, 2) direct driven 3) gear driven and 4) integral.

Belt driven spindles use cogged or v-belts to transmit power from motor. This drive system is cheap and easy to maintain but the required tension of the belt necessary to transmit rotation to the spindle produce unwanted effects as energy losses, noise, vibration and additional load on spindle bearings. Their efficiency is about 95% and can reach speed of about 15,000 rpm with good performance. Spindle with direct drive receive the power direct from the motor by means of a direct coupling system. Because there is not possibility transform power to torque, they are capable of working at high speed but only at low torques. Their transmission efficiency is about 100%. Gear-driven so not perform well at high speed because at these speeds energy loses and vibration increase and guarantee poor performance. However, they can transmit significant torques at low speeds and therefore are suitable for heavy duty machining. Their efficiency is less than 90%. Integral spindles also known as electro spindle, built-in-motor spindle and motor spindles contain an electric motor incorporated its assembly, which is centrally located between bearing arrangements (front and rear). This configuration occupied less space in comparison with other spindles but it more expensive and intricate in its construction. However their advantages are evident, for example they achieve high speeds at low vibration and noise with exceptional mechanical performance (Quintana, et al., 2009). Integral drive spindles are also characterized for high acceleration capability, wear-free, high rigidity and excellent results in finishing operations (Siemens, 2009, p. 14)

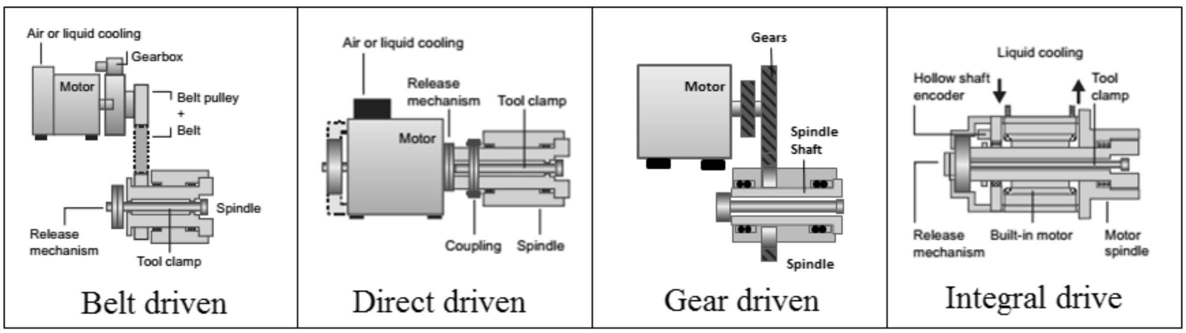


Figure 16 Transmission systems for spindle in machine tools. Adapted [Courtesy: Siemens]

Traditionally, the working position of the spindle has been related with specific machining process, but this is rapidly changing due to the development of CNC machine tools. Milling process for example that has been traditionally carried out with the spindle located vertically is used today in horizontal position in milling large aircraft structural parts in aerospace industry. (SAE, 2010). Another example is turning, which is conventionally executed in vertical position. Today is possible to find vertical lathes for automobile applications, which catches and clamp the blanks from above without the need of external assistance (EMAG, 2015). The main advantage of working on vertical position is that favors chips removal from the cutting zone, which facilitates subsequent machining operations.

Another consequence of development of CNC Machine tool is the introduction of rotatory headstocks in recent machine designs. These headstocks can rotate in one or two axis, which enables the machine tool to achieve complex milling features in a single setup. Reducing lead times and increasing productivity. Examples of rotatory headstocks are twist, swivel (Figure 17, left) bi-axial swivel (center) and parallel kinematic headstocks (right). Two main disadvantage of these heads, although, as (Quintana, et al., 2009, p. 81) point out, are the insufficient rigidity for high power application and the need of more sophisticated numerical control systems to direct their motion.

Apart from re-orientate conventional machining process and equipped spindle with several axis, a third strategy in new designs is integrate different machining cycles in a single machine. These multitasking machine means that they can accomplish different machining process and/or the same machine process in independent work areas. As the reader of this work may suspect, the idea behind this is productivity increase due to several machining cycles in a single machine.

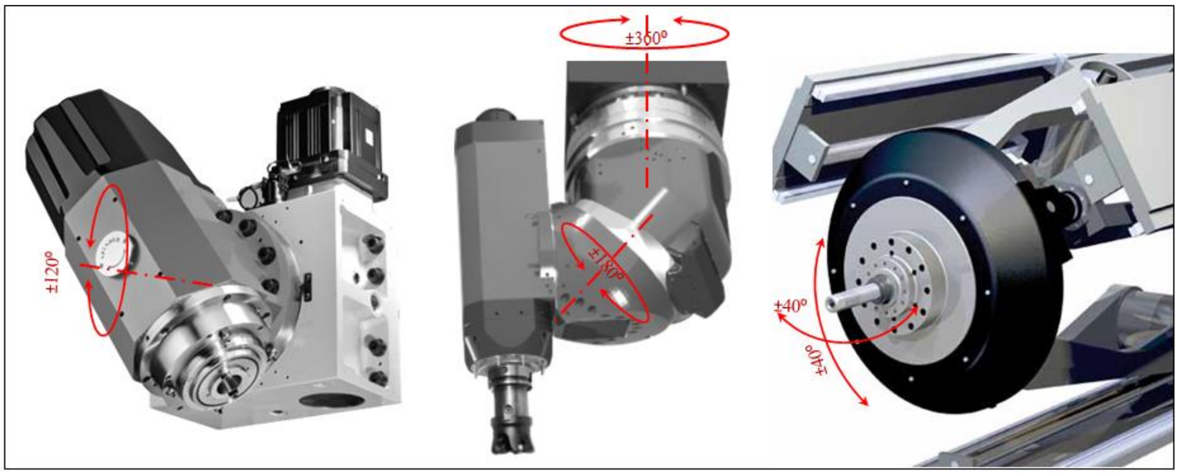


Figure 17 Different rotatory headstocks: Swivel RS4, Bi-axial swivel M21 45º and parallel kinematic Sprint Z3. [Courtesy: Spinder Technology Co, Cytec.and Starrag, respectively]

Multitasking machine tools are equipped with several spindles. These additional spindles can perform the same machining operation as the main spindle (DUO, TWIN or TRIO concepts), different machining operation than the main spindle (turning/milling, turning/grinding etc.) or execute other complementary function. In modern CNC lathes for example where tooling turrets have often their own spindle in order to drill or perform simple machining operations. DUO, TWIN, TRIO designs, however often add additional spindle to carry out same machining process, but not necessarily the same operations. (see Figure 18).

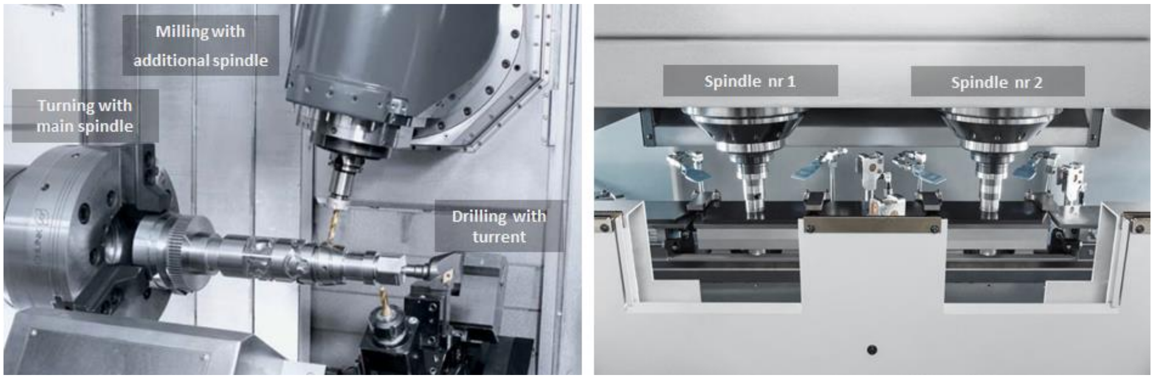


Figure 18 Multitasking machine tool: Milling/turning lathe CTX TC and VSC twin 250 [Courtesy: DMG Mori and EMAG]

It seems that spindle physical dimensions will depend on manufacturing sector they are used for. The larger the part to machine in one operation, the more power the spindle will have in order to achieve a significant removal rate. By this way, productivity requirement are meet. This will tend to result in a more massive spindle. For example based on spindle manufacturer catalog ((HSD Mechatronics, 2015, pp. 4,5), typical spindles used for milling are within the range of 1-36 [kW] can have a shaft diameter of (80-300 mm) and a length of 300-700 mm, with a weight from 8-150 [kg]. However, in aeronautic industry, large aluminum/titanium frames of airplane structural components are machined. This required even more power and speed to achieve high removal rates according to (Noel, et al., 2012). Therefore massive spindles mounted in extensive structures are used as gantry machine tools.

2.1 Main components

Machine tool spindles are sophisticated precision assemblies that have a significant number of components. Each of them plays a crucial role in the functionality of the spindle. Components' individual performance will contribute at certain extend the overall performance of the spindle. Figure 19 illustrates the main component of spindle with integral drive.

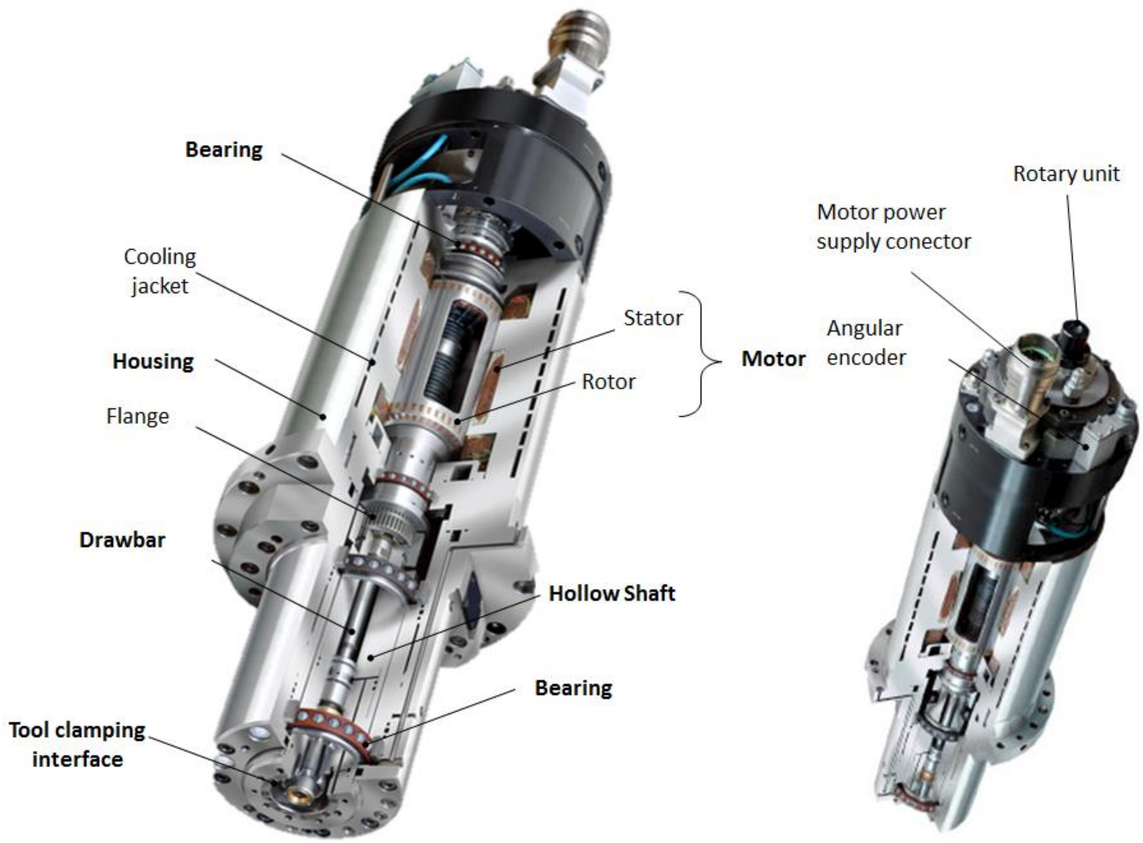


Figure 19 Main components of an integral spindle. Adapted [Courtesy: SKF]

As mentioned earlier, the main function of a machine tool spindle is to give accurate rotational motion to the tool necessary for material removal. In order to achieve this, spindle components interact as following:

The spindle ***housing (section 2.1.1)*** locates all the other components of spindle. Electrical power supply is provided to the spindle by wire. This electric energy is transformed to mechanical energy (rotation) by the ***motor (2.2.2)***, which in the case of integral drive spindle, is built inside. The *stator* generates a rotating magnetic field when current flows through its winding. This magnetic field produce that the *rotor*, integrated with the ***shaft (2.2.3)***, start rotating. Due to shaft is supported by rolling ***bearings (2.2.4)***, inner ring and rolling elements of the bearings start rotating as well. The ***drawbar (2.2.5),*** located inside the hallow shaft, pull the tool holder into the tool clamping interface. The pulling force is such a high magnitude that results in secure transmission of rotatory motion from the shaft to the tool holder. Motor and bearings release heat which is extracted by a built *cooling jacket* on the spindle housing***. Encoders (2.2.6)*** record the angular position of the spindle at any time, while ***sensors (2.2.6)*** give information about drawbar status (clamped/ unclamped) and other important parameters to evaluate spindle performance. The rotary union can provide different pressurized media to the spindle, including coolant, lubricant or air.

2.1.1 Housing

The housing is the structural part that contains all the other parts of the spindle and separates physically the spindle from the machine tool structure. Their external appearance can vary (see Figure 20) depending in in internal configuration and the required interface for the machine tool body where it will be installed. However, the most common are square and cartridge-type, the later has a cylindrical shape. Most integral spindles have cartridge type housing with flange mounting (to right of the figure).

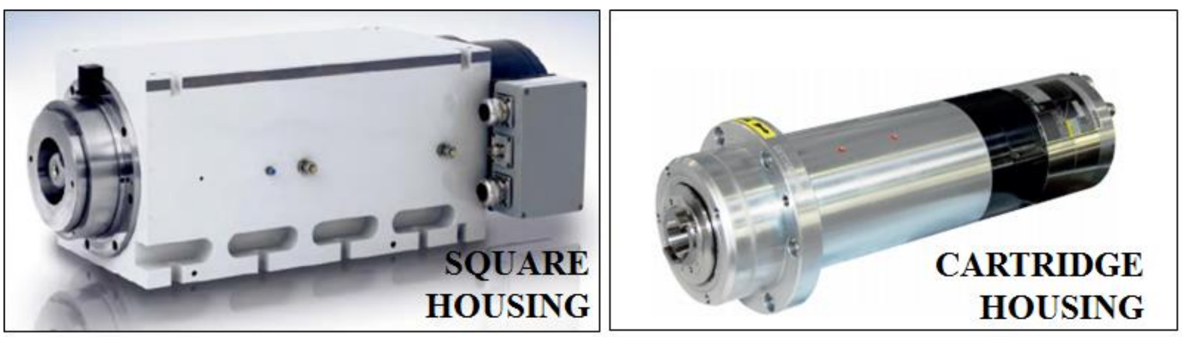


Figure 20 Common housing shapes for machine tool spindles [Courtesy: Spindeltechnik and HSD Mechatronics]

The principal function of the housing is to locate accurately and protect their components from environment. Geometry accuracy of the housing is especially important for assembly and location of spindle bearings. In fact, machine tool spindles require high precision bearings, which are classified by tolerance class. For this reason the housing has to comply with geometry and dimensional tolerances as runout, angularity, circularity and concentricity according to ISO 1101 (NTN, 2012, p. 10).The housing also protects the internal components against contamination from cutting operation (mainly cutting fluid and chips). This protection is given thanks to a sealing arrangement at the end on the spindle nose. This is often labyrinth-type which has several narrow grooves tightly adjusted to housing walls, providing interacted path to keep away contaminating particles. But in several spindles models, apart from having labyrinth seal, air sealing is achieved by a constant and controlled feed of pressured air into the spindle [Fisher] [GMN] [Franz Kessler GmbH].

Secondary functions of the housing are to provide the necessary channel network for lubrication of bearings and cooling of bearings and motor in integral spindles. Besides, in some motor spindles, the housing provides internally the cutting fluid to the tool tip necessary for machining operations. In addition, spindle housing provide location of sensor and electric network (Quintana, et al., 2009).

2.1.2 Motor

The motor is the element that transform electrical energy to mechanical energy in form of rotation. This motion is tranfer from the motor to the shaft by a transmision system, in external driven spindle (belt, gear, or direct drive) or inside the spindle in spindle with integral drive. In machine tool applications, electric motors are used. The electric motor consist in two main parts: a rotor and stator. The stator is the stationary component that generates a magnetic field which produce suficient electromagnetic forces on the rotor to make it turn.

There are a large range of electrical motor on the market. They can be powered by either direct Current or altenat current wich give the motor their designation DC and AC, respectively. Within AC motors, there are two types: sycronhous or asynchronous. In synchronous motors the rotor rotates at speed determined by power supply frequency and number of poles, know as syncrhnous speed. In synchronous motors (denominated also induction motors), the rotors rotates slower than synchronous speed (Groschopp, 2012). In spindles, the speed is regulated by varying the frecuency thanks to frecuency converter that comunicates with CNC system of the machine tool.

The trend in the industry today is shifting from asyncronous motors to permanent-magnet synchronous motors. The later type presents comparatively better characteristic as: high power density and better power-to weight ratio, which enables smaller and lighter spindle constructions (Abele, et al., 2010, p. 782). Figure 21 shows a typical induction and synchronous motor and how their parts are assembled with the spindle shaft in intergral drives.

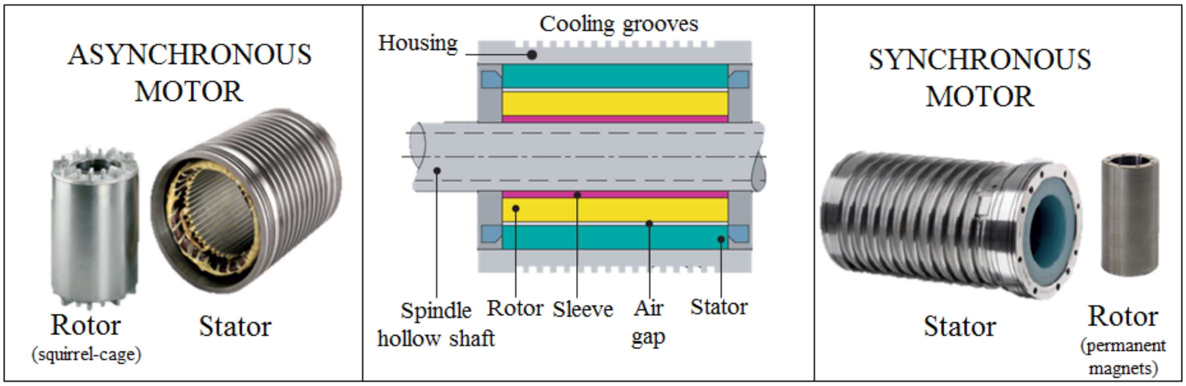


Figure 21 Examples of AC motors components for integral spindle. Modified [Courtesy: Siemens]

As the drawing of Figure 21 shows, integral spindles contain the motor built inside in such way that the rotor is integrated with a hollow shaft, which will transmit the necessary power to the tool. The stator contains the rotor without physical interference thanks to a thin air gap in between.

Because their nature, electrical motors are not 100% efficient in transforming electrical energy to mechanical energy. Some of this energy is losses in form of heat, raising the temperature of the stator or rotor depending on their configuration. In fact, as (Abele, et al., 2010) points out, the motor is one of the major a source of heat in integral spindle. Because the negative effects of thermal effects in spindles as axial displacement (of microns order of magnitude) of the shaft (GMN, 2015) or damage components due to overheating (CADEM, 2014), machine tool manufactures controlled these effects mainly using two strategies. The first is, with thanks to temperature sensors, machine tool operators are alerted by alarms or power supply is disconnected automatically from the motor. The second strategy to avoid temperature increase is to provide cooling system for motor. In particular, integral spindles are equipped with in external cooling systems to the buit-in motor. This is achieved by providing a “cooling jacket” (see Figure 22) which is a helical cavity that surrounds the stator. Through this cavity, a coolant (e.g. water, oil, glycol solution) is circulated by an external cooling unit. In many spindle designs this cooling circuit also is responsible for extracting the heat from bearings (GMN, 2015, p. 16).

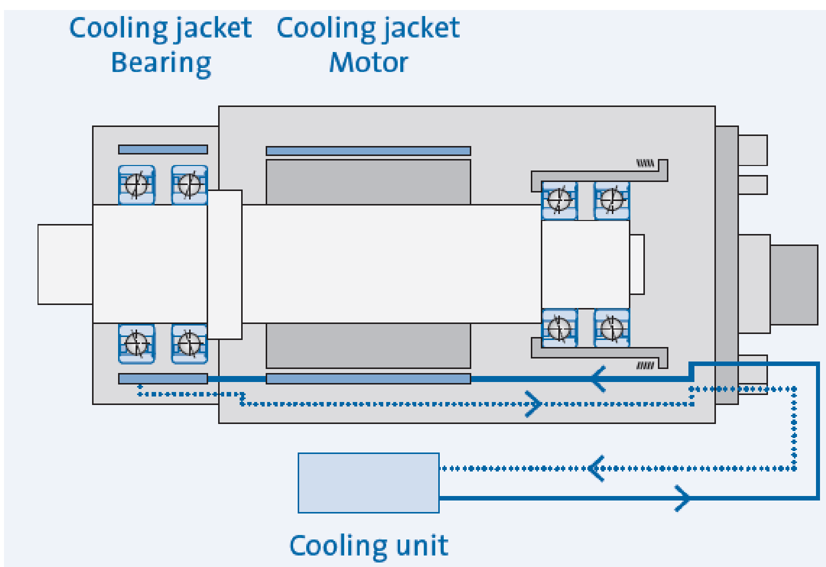


Figure 22 External Cooling unit for spindle motor and bearings [Courtesy GMN]

In order to understand the spindle capabilities for machining, it is important that they depend in large extend on the motor..Because torque and power are speed-dependent in electrical motors, the majority of spindle manufactures includes motor performance diagram of the motor along the entire spindle operating speed range. Figure 23 shows a typical torque speed diagram for characterizing spindle performance. In most of the motors, they deliver maximun torque at low speeds until reaching “base speed". Runing the spindle at higher speed than then base speed result in a rapid decrease of the torque available for the spindle. The higher the speed the less torque available will result. In the last part of the speed range, the power decreased while the torque is constant. Inportant to know is that traspasing torque-power curves can result in motor stalling (unrotationa state) that can scalate to overheating, which can cause motor failure, know as “motor burning”.

In these torque-power diagrams, S1 and S6 refers to two different duty cycles specified by. the standard IEC 60034-1 of the International Electrotechnical Commission. Because the performance of the motor wil be limited by temperature increase, the different duty cycles referst how the working load on the motor is distributed during a certain period of time.

- S1: *Continuous duty* : the motor works at a constant load for duration long enough to reach equilibrium temperature
- S6 *Continuous operation with periodic load*” Sequential, identical duty cycles of running at constant separated by period of not load at zero speed .

In reality, as is explained in (CADEM, 2014, p. 3) this duty cycles are just a reference, because in real machining operations the spindle will be subjected to load of different types for different period of times. For example, the spindle in a 2 minutes-timeframe, can perform roughing, change tool, milling and so on, varying cutting parameters (feed, depth of cut, velocity) and working conditions wich affect the resulting cuttting forces, and then load, in the spindle.

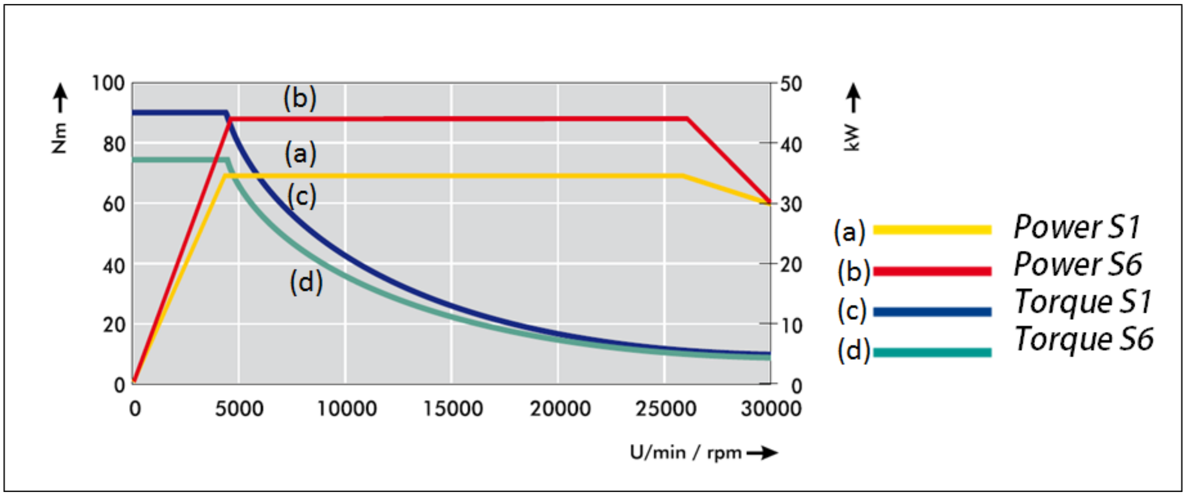


Figure 23 Typical torque power diagram of spindle over the entire operating speed [Courtesy:Cytec]

Finally, the rotor has to have the adequate balanced in order to perform well and not damage other components. When it comes to finishing quality, specially important is in spindles used for grinding, because they usuallly work with extremely demanding tolerances (at magnitud order of microns).Thats why these devices are equipped with internal or external balancing mechanisms to correct their inbalance. Other problem with imbalance is that can have negative effects in other components. Bearings, for instead, due to excesive imbalance on rotor-shaft are subjected to additional dynamic loads reducing their nominal life (Taneja, 2012). Therfore the inbalance must be limited to rasonable levels. Example of this is the standard ISO 1940-1 which stablish spindles for precision grinders required balancing grade of G0.4 (IRD Balancing, 2009).

2.1.3 Shaft

The shaft main function is to receive the power from the rotor and transmit this power to the tool holder. The shaft usually rests on two to three bearing arrangements depending on the spindle configuration. These bearings locate accurately the shaft, limiting its movement to only rotation. Shaft's exact angular position is registered by encoders situated in the back end of the spindle. The shaft is hallowing in order to contain the drawbar and in some configurations allow the circulation of required media as cutting coolant.

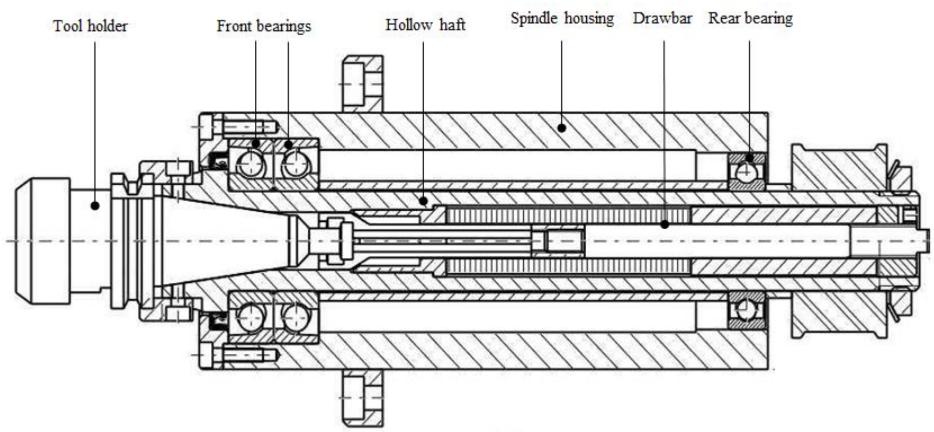


Figure 24 Typical spindle assembly for belt driven spindle.Edited from (www.cnccookbook.com)

Geometrical and dimensional specifications are required for this components as concentricity, parallelism, surface finish and runout (NP, s.f.). Specially important is the so called runout because, similar to the spindle housing, the shaft must be designed and manufactured to meet a specific tolerance classes for ensuring bearing mounting specifications.

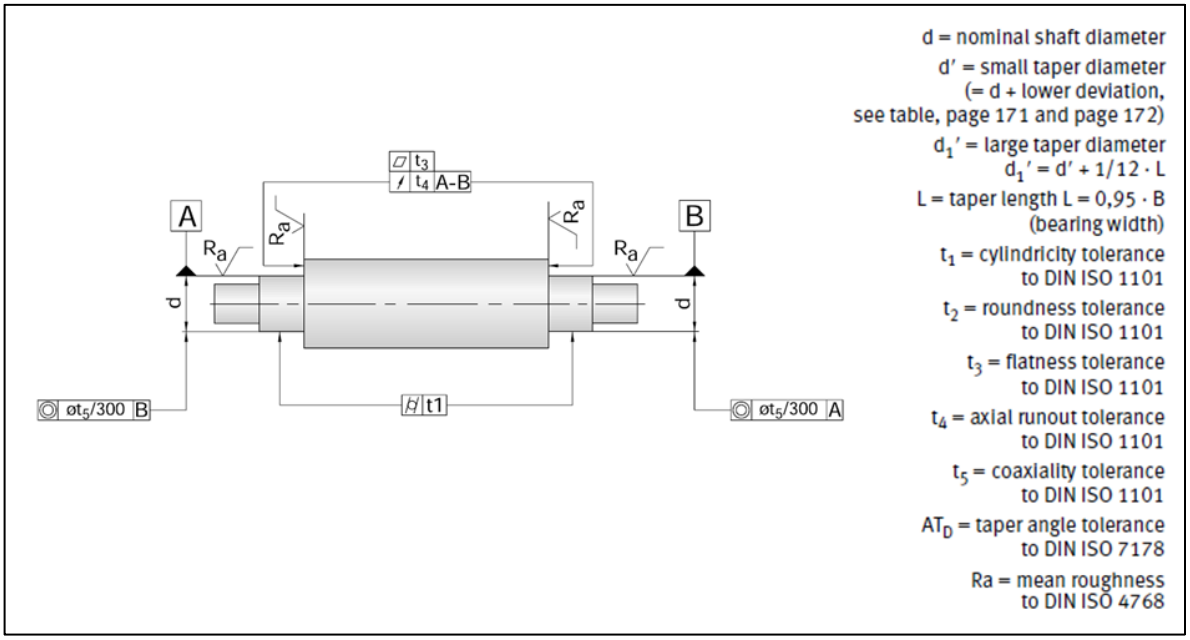


Figure 25 Typical Geometrical tolerance for spindle shaft (INA FAG, 2014, p. 39)

According to mechanics theory, the shaft or shaft-rotor (in integral spindles), as any solid body will have natural frequencies associated with specific vibration modes. These modes can be excited, for example, by unbalance forces at certain rotational speeds. These rotational speeds are called critical speeds. Usually, spindles are design so the critical speed falls out from 50% of the maximum speed (Smith, 2011a). However, because its broad speed range, most of high speed spindles (from 12000 rpm) presents one or two speed within the operating speed range (SIS 728000-1).

2.1.4 Bearings

Bearing main functions are to reduce the friction and transmit loads, while locating and supporting the shaft (McDermott, 2011). There are several bearings types as rolling bearings, aerostatic, magnetic, hydrostatic and hydrodynamics bearings. However, as (Weck & Koch, 1993) exposes, rolling bearings have overall better characteristics compared with each of the other type, within machine tool spindle applications. One main disadvantage of rolling bearings though, is the lack of high damping properties. Nonetheless, rolling bearings stand out in low price and easy lubrication. Apparently that would be the main reason, they dominated the market for machine tool spindle applications today. Because its importance, this subsection will be dedicated to rolling bearings. Figure 26

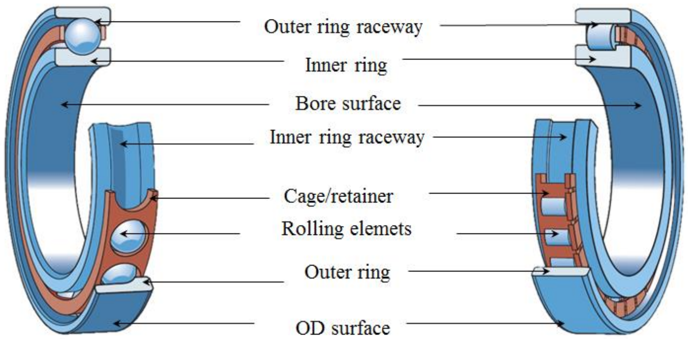


Figure 26 Main parts of two rolling bearing commonly used in spindles. Angular contact ball bearing (left) and cylindrical roller bearings (right). Adapted from [Courtesy:SKF]

In order to function, every single component of the rolling bearing (See Figure 27) is important. In order to understand the functionality of each bearing components, the functional operation of a bearing is explained next. In operating condition, the bore surface of the inner ring is in direct contact with the shaft. Hence the shaft and bearings inner ring rotate at equal angular speed, causes rolling elements rotate through the raceway, while being constantly loaded (axially and radially) by the shaft. These rolling elements are equally separated by the cage, which prevents the mutual contact and minimizes friction and heat generation. Besides, the cage guides the rolling elements from the loaded to the unloaded zone of the inner ring and prevents the harmful sliding motion of the rolling elements (SKF, 2013, p. 37). While rotating, the rolling element also requires a constant oil film for reducing friction with the outer and inner ring raceways. At the same time, the outer ring rests on bearing housing without rotating.

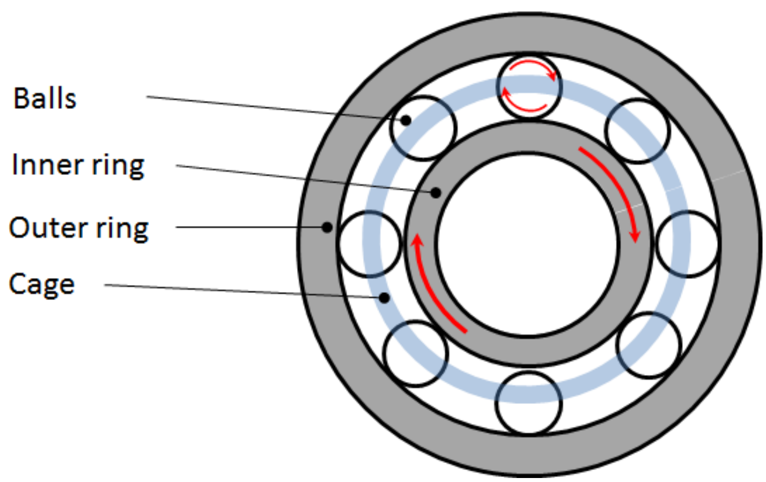


Figure 27 Rolling bearing diagram

In order to preform properly ensuring the adequate operating life, bearings components are made from a vast range of materials. These materials provide the required hardness, fatigue resistance, dimensional stability among other necessary properties. Bearing rings and rolling elements are often made of special steels as carbon chromium steel, stainless steel and chromium-nickel alloyed steels. Some of these steels are subsequently heat treated to improve surface hardness. Ceramic, specifically silicon nitride, is another material used for rolling elements due to thermal and mechanical advantages in comparison with steel. Cages are mainly fabricated (machined or stamped) with hot-roller carbon steel or brass. Brass is preferred when lubricant can react with steel. Polymer cage, injection molded, are also used. Because they provide good strength/elasticity rate. However high temperatures and reactivity with the lubricant can reduce its usage life. (SKF, 2013, pp. 151-153)

There are countless rolling bearing types available on the market and their selection will be based, at the first stage, on the performance requirements and operating conditions. However other factors as dimensional constraints and environmental condition are also essential (NSK, 2013, p. A16). In other words, the selection will be mainly based in the specific technical application. In machine tool spindles, the requirements are usually high durability, accuracy, damping, rigidity and while having easy lubrication and low friction (Weck & Koch, 1993, p. 445). As Weck and Koch mention another important demand on spindle bearings is the so called speedability or speed factor or n*dm with [mm/min] units.Where n represent rotational speed while dm the bearing mean diameter.

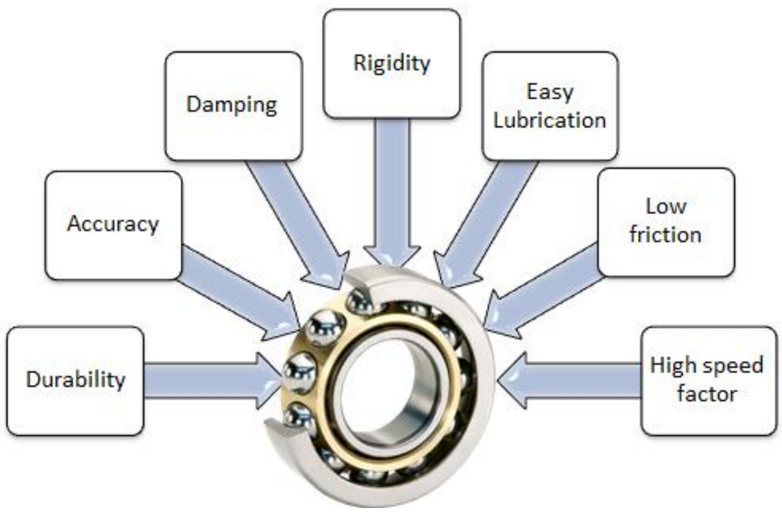


Figure 28 Main requirements of bearings on machine tool spindle applications Recreated from (Weck & Koch, 1993, p. 445)..

In order to meet these demands, rolling bearing, angular contact ball bearings and cylindrical roller bearings are preferred in machine tool spindle applications (See Figure 26). Angular contact bearings provide punctual contact with the outer and inner rings, thus low friction is achieve, generating less heat and allowing higher speed. Besides they can carry axial and radial loads. In the other side cylindrical rolling bearings CRB (single or double row), have linear contact with the raceway and therefore generate more friction than ball bearings. However they carry radial loads better and are more rigid or stiffer than ball bearings. Because these two rolling bearing types complement each other, they are often combined in spindle designs, which will be explained later.

One of the factors that influence durability of bearing's life is the amount and type of load it carried compared with the nominal loading conditions. The load type that bearings are designed to withstand is in close correlation with bearings' geometry. In angular contact ball bearings (ACBB) the contact angle α will influence the amount of axial or radial load the bearings can carry and the allowable speed of the bearing. Higher values of α implies the bearing can withstand more axial load, but at lower speeds. For this reason ACBB with small contact angle are more suitable for radial load and high speed (NSK, 2009, p. 11). Typical contact angles for angular contact ball bearings are 15º, 25º and 30º

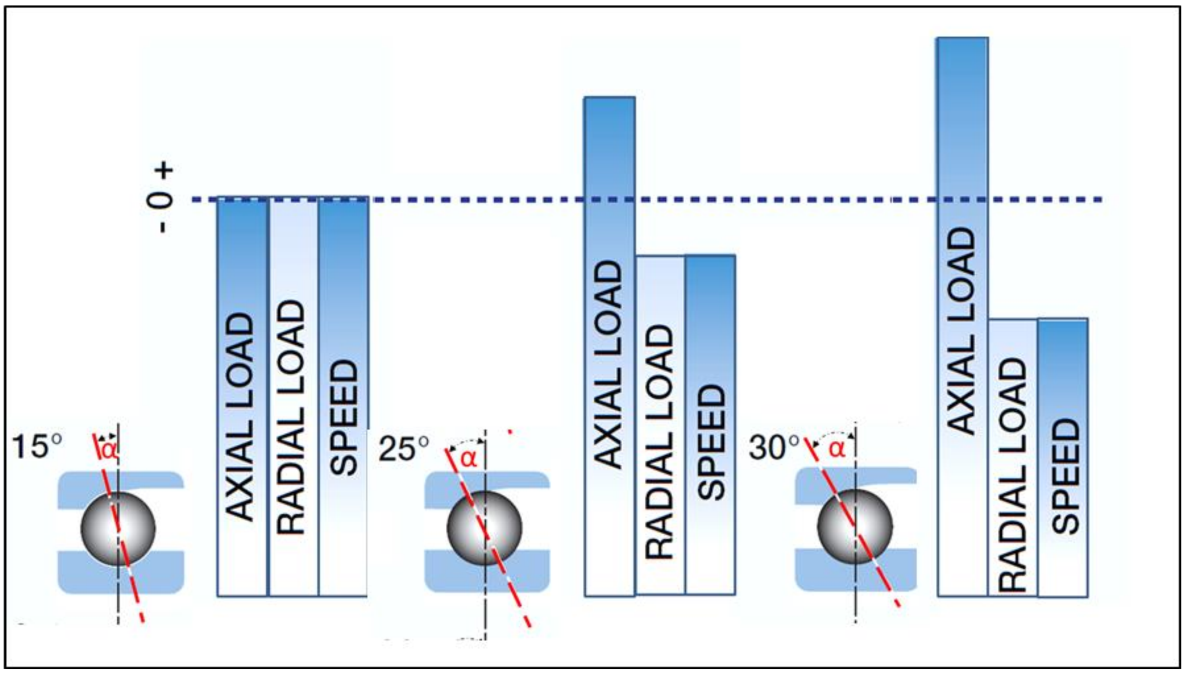


Figure 29 Influence of contact angle in bearing capabilities. Adapted from (NSK, 2009, p. 11)

Bearing sets

In order to carry axial load in both directions (NSK, 2009, p. 11) and increase spindle axial and radial rigidity (INA FAG, 2014, p. 194), ACB bearings are combined and arranged in in different ways along the spindle shaft. These arrangements are often referred to as bearings set. The most basic configurations are back-to-back (DB), face-to-face (DF) or tandem (TD) referring to the relative position of the bearing faces with respect to a second bearing (see Figure 30). However they can be combined for creating more arrangement with more than two bearings next to each other.

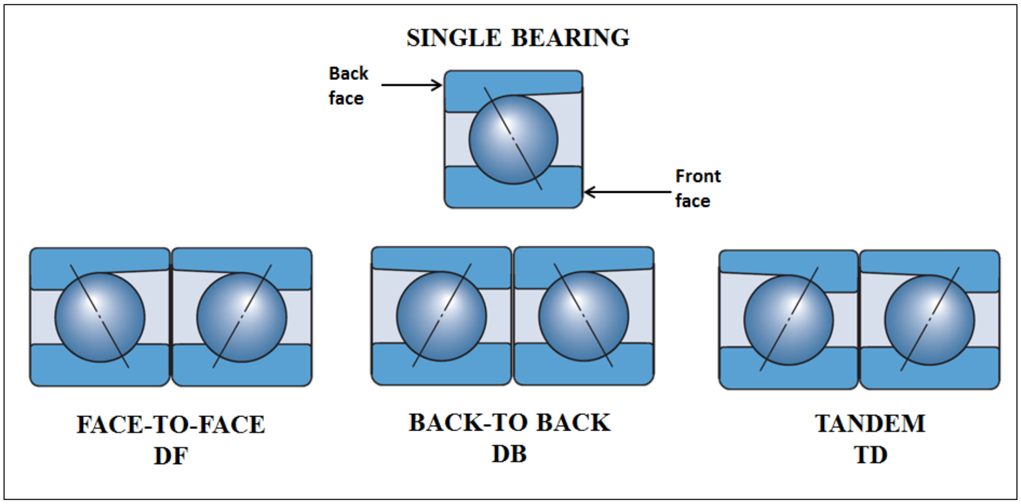


Figure 30Different types of bearing arrangement

To illustrate how different bearings arrangements are used in spindle application, refer to Figure 31. It shows a belt-driven spindle used in a CNC lathe. The spindle has two bearing arrangement. In the front end there are three contact ball bearings. Two of them are in tandem and the third back to back. The pulley needs certain tension for transmitting the torque to the spindle shaft. This tension resulted in radial loads on the back end of the spindle. This explains the double row roller bearing in the back end. Roller bearings are more rigid in the radial direction and withstand better radial loads than ball bearings.

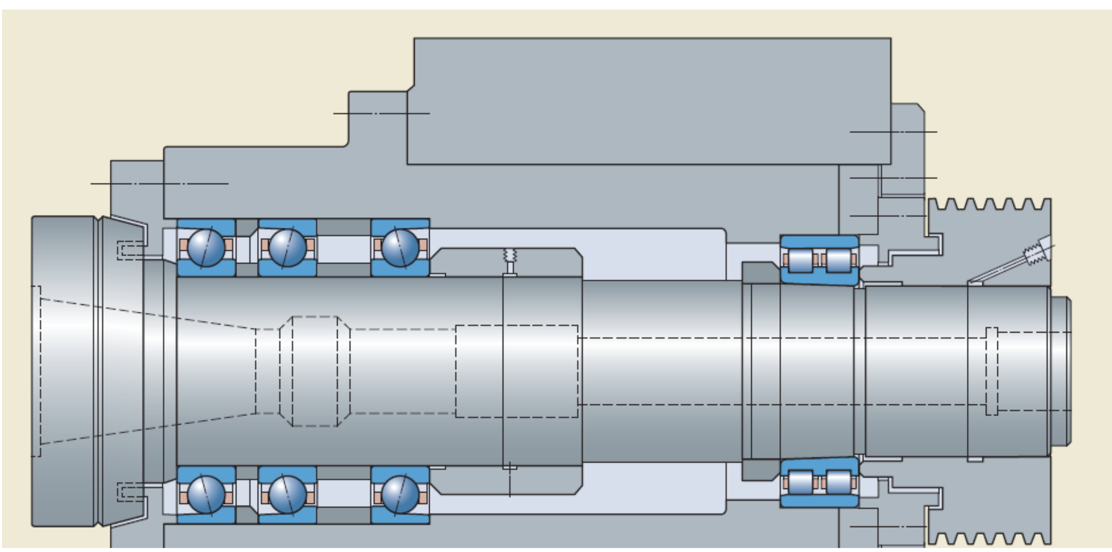


Figure 31 Belt driven spindle of CNC lathe (SKF, 2014, p. 58)

Another example of common bearing arrangement in spindles is illustrated with Figure 32. It shows an integral spindle (motor spindle) used in machine center. In contrast with the last example, in this spindle do not carry heavy radial load at the back. Therefore a single row roller bearing is would be convenient. Another characteristic is that while the bearings in the front are axially constrained, the bearing in the back of the spindle is adjustable to accommodate thermal expansion of the shaft.

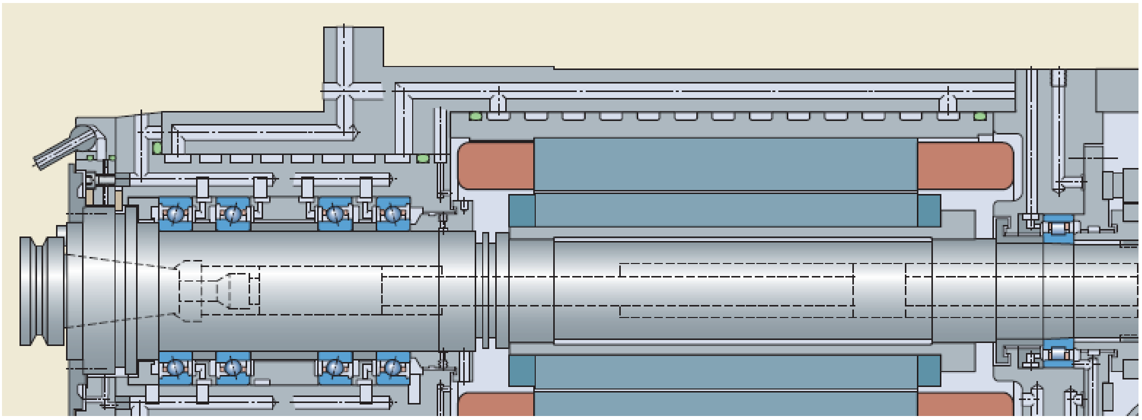


Figure 32 Integral spindle for machining center (SKF, 2014, p. 62)

Bearings Precision

As mentioned earlier, accuracy is also another important requirement of spindle bearings. For this reason, bearing manufactures refer often to bearing for machine tool application as "High precision bearings". This is because spindle bearings have to comply comparatively with more demanding tolerances than other bearings. Because different application will require different degree of accuracy, also known as tolerance classes. These tolerance classes are stablished by different standards as ISO 492, DIN 620 and ANSI/AMBA Std. 20 among others depending of the bearing type (NTN, 2012, p. 7).

In particular, these tolerances classes refer to dimensional accuracy and runout of inner and outer ring of the bearing. For most applications the accuracy of the inner ring is a determining factor for selecting a suitable tolerance class for the bearing (SKF, 2014, p. 25). This seems to be the case of spindle bearings due to the importance shaft accurate motion to ensure precise machining. When it comes to dimensional accuracy however, the critical aspect is the adequate preload of the bearing.

Boundary accuracy of outer and inner ring of the bearing, together with the bearing housing and shaft will determine the interference or fit. The fit influence the internal clearance or preload of the mounted bearing SKF p25.

Bearing rigidity and clearance

Rigidity on the spindle system depends greatly on bearing preload. Preload values are often given by bearing manufactures and they represent "the axial force required to press together the rings or washers of new unmounted bearings" (SKF, 2014, p. 50).

Bearing preload will also important because will determine the bearing life, runout and quality of surface finish (Abele, et al., 2010, p. 797). High precision bearing as used in spindles are high sensitive to preload. As (Mannan & Stone, 1998, p. 889).highlight, too low preload can result in low rigidity (stiffness) leading to poor machining while. Excessive preload can result in bearing life reduction.

Radial clearance and thus preload can be influence by thermal effects during operation as Figure 33 illustrates. Temperature difference between the inner and outer ring can reduce radial clearance. This lead to increasing preload on the bearing, which results in higher friction (SKF, 2014, p. 37), which eventually will diminish bearing life.

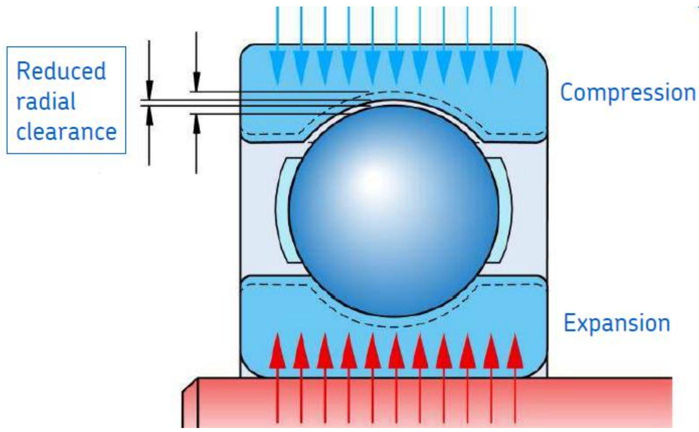


Figure 33 Result of thermal effect in bearing radial clearance (McDermott, 2011, p. 19)

Because of the importance of bearing preload in bearing life several spindle includes preload system in their design.

Lubrication

Easy lubrication and friction are intimal related with bearing life. According to (McDermott, 2011, p. 22) approximately one third of premature failure of bearing is due to poor lubrication. The oil film can be as thin as 0.2 µm and prevent metal-to metal contact between the rolling element and the raceway. If this oil is affected, thermally or by contaminants, it can lead to premature wear of the bearings. Figure 34 illustrate the above mentioned. Because the relative large size of dirty particles compared to oil film, they can easily interpose between the rolling element surface and the oil film.

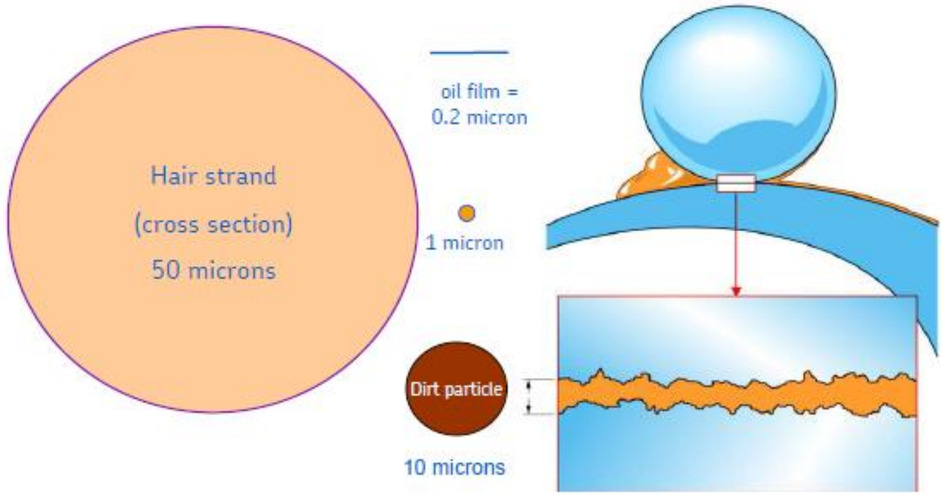


Figure 34 Relative size of oil film on bearings

There are three main methods for providing lubrication to bearings. These are oil minimum quantity lubrication; oil injection lubrication and grease lubrication (Weck & Koch, 1993). Each of them have their advantages and limitations. In machine tool spindles, they are often chosen by spindle designers based mainly on the required condition of the spindle but also how the lubrication system integrality in the overall design of the spindle.

Grease lubrication is the simplest solution and provides lubrication for an extended period of time to the bearings. In this case the sealing of the bearing is critical for preventing contamination on the lubricant. This is suitable for low speed, (speed factor $<10^6$ m/min) according to Weck & Koch and (speed factor $< 8{,}5\text{x}10^5$) according to (Quintana, et al., 2009, p. 93). Weck & Koch means that higher speed may difficult oil diffusion into the rolling contact while Quintana et al. states that higher speed than specified can degrade the grease rapidly.

Oil injection lubrication is another alternative, where the oil is injected at high pressure directly into the bearings. This creates a surplus of oil which ensures permanent lubrication film and do not limit rotational speed. Some disadvantages includes: transmission power loss (can reach up to 1/3 at high speeds) and the need of difficult-to-manufacture return conducts, according to Weck &Kock .

Oil minimum quantity lubrication or also known as oil-air lubrication is more widely used in high speed spindles than the other two mentioned. for providing oil at regular basis. It consists in mixing air and oil (provided by a compressor and pump respectively) for injecting into the bearing. See Figure 35. According to Weck & Kock this system allows speed factors up to $1.5\text{x}10^6$ /min for steel bearings. However this lubrication method required a more complex system to provide lubrication compared to injection lubrication.

This can be perceived as disadvantage when comparing to other solutions. This lubrication strategy required also monitoring, control and additional equipment as filters and pumps.

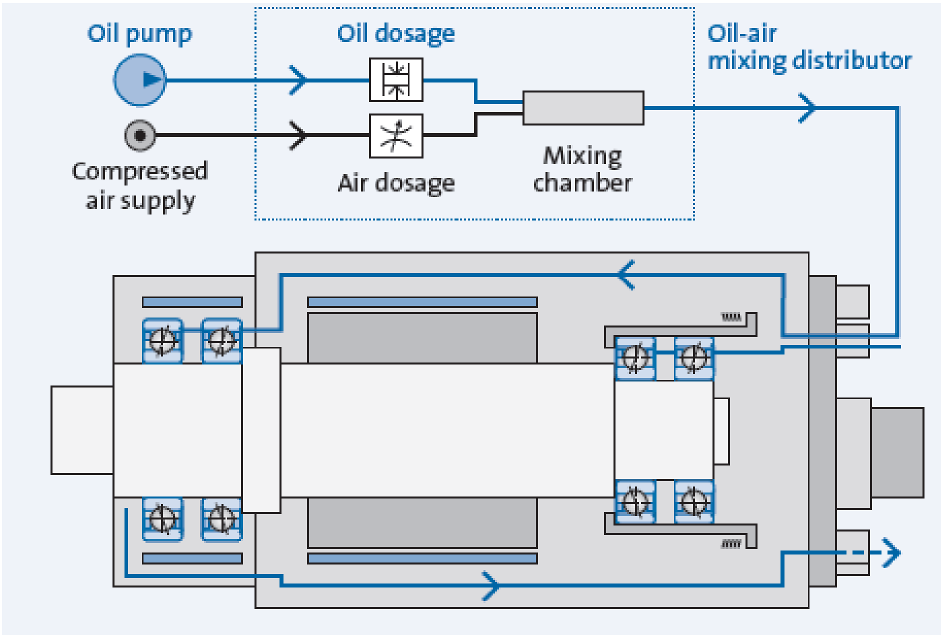
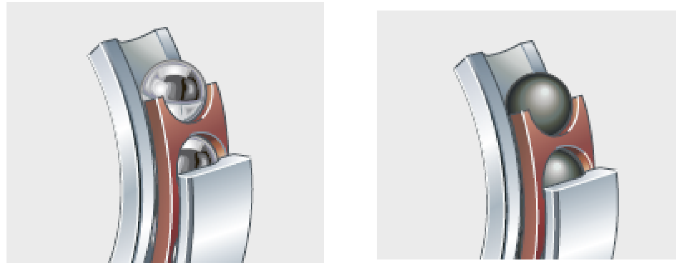


Figure 35 Oil-air lubrication system in motor spindle (GMN, 2015, p. 6)

Hybrid bearings

In order to cope with negative effects in bearings produced by thermal and dynamic phenomenon due to high speed in spindles, hybrid ball bearing has been used as a widely accepted solution. These bearings have the characteristic of using ceramic balls instead of steel. As Figure 36 shows, ceramic balls (generally grade silicon nitride) are lighter due to low density (40% lower than steel) this is a desired property when dealing with centrifugal force at high speeds. Besides, ceramic balls have a thermal expansion four times lower coefficient than steel. This allows minimal dimensional change when thermal conditions are unfavorable. Hybrid bearings are also more rigid because their higher elasticity modulus. Another positive property point out by (Smith, 2011b) is that ceramic balls have lower friction coefficient with the raceway and therefore they generate less heat. Heat and high temperatures may increase the preload, generating even more heat that can result in the degradation the oil film.

In the other side hybrid bearings have also some drawbacks. They are significantly more expensive than conventional bearings. Besides they are poor heat conductors which may be disadvantageous for heat dissipation generated in the shaft as (Smith, 2011b) suggest. He also explains that this is because usually the bearings are the only path for heat dissipation from the spindle. (Quintana, et al., 2009, p. 92) explain that hybrid bearings are especially sensitive to collisions of the spindle. Collisions can easily cause brittle fracture in ceramic balls.



Properties (units)	Conventional steel bearing	Hybrid ceramic (Si_3N_4) bearing
Young's modulus (GPa)	208.00	315.00
Hardness (VickersRc)	60.00	78.00
Density (g/cm^3)	7.80	3.20
Max. usage temperature (°C)	120.00	800.00
Coefficient of expansion ($10^{-6}/K$)	11.50	3.20
Poisson's ratio	0.30	0.26
Thermal conductivity (W/mK)	45.00	35.00
Chemically Inert	No	Yes
Electrically conductive	Yes	No
Magnetic	Yes	No

Figure 36 Comparison of material properties steel and ceramic. Adapted from (Quintana, et al., 2009, p. 92)

Bearing durability

Bearing life in designed-condition can be express as the bearing nominal life. There is a general formula[1] for calculating this life which is often found in bearing catalogues or books of machine components design. However this formula provide a basic estimation and not consider all the factors that may influence bearing life. In an attempt for refining this bearing rating formula, some important bearing manufactures have specified additional ”correction factors” which compensate for other aspects. For example SKF and NTN include a_{SKF} and a_{NTN} correction factors. These factors are aimed at including lubricant condition, contamination and fatigue load ratio on the basic bearing life formula (SKF, 2013, p. 65) (NTN, 2012, p. 23).

Even with this correction factors, the mentioned formulas may not suitable for spindle bearings. This is because they often work at different condition compared to other rotatory machinery. Spindles bearing are subjected to high axial and radial load, which may be

[1] According to ISO 281, the formula for bearing life:

$L_{10} = \left(\frac{C}{P}\right)^{p}$ Where:

L10= basic rating life (at 90% realiability) for one million revolutions,C:= basic dynamic load rating, P:=Equivalent dynaic bearing load and P= exponent of the life equation (ball bearing=3, for roller bearing 10/3).

difficult to determine, thus complex to calculate. At the same time spindles operates at not fixed speed, due to their requirement to adapt its speed depending on the tool and the material being machined.

When operating, bearing life is a function of bearing components and lubrication conditions See Figure 37. This means that every part of the bearing has great importance in determining the bearing life ((SKF, 2013, p. 62). For example if the cage is damage, then probably its failure will dictate the life of the bearing, regardless the condition of the remaining components.

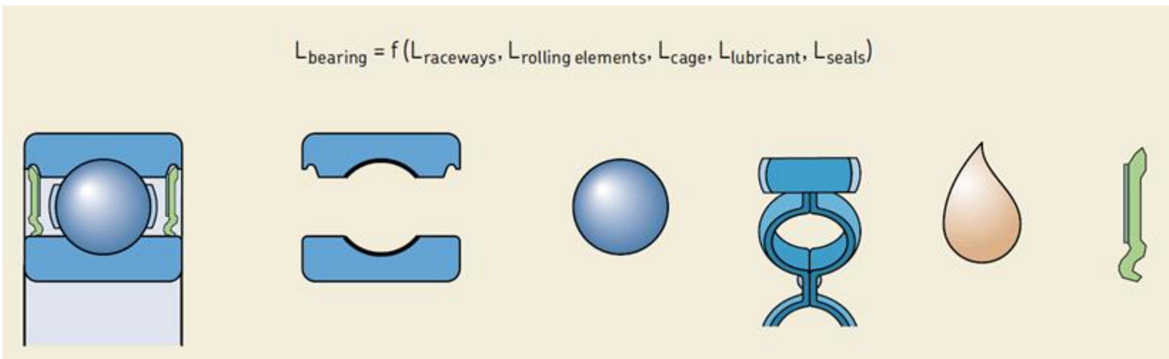


Figure 37 Bearing life as a function of individual life of its components (SKF, 2013, p. 62)

Obviously the life of every component will be affected by the working condition. If these conditions are in accordance with the designed working condition, the bearing will expect to reach its nominal life in 90% of the cases according to bearing life rate formula. In the other hand, if the bearing is subjected to conditions was not designed to work under, an early fail can be expected.

Inadequate condition for spindle bearings can be the result not only due to inadequate working conditions but also of poor design, manufacturing or assembly of the spindle. For instead, insufficient preload on the bearing due to poor assembly quality of the spindle, can reduce the nominal life of the bearing significantly (NSK, 2009, p. 17). Or excessive imbalance in the shaft can reduce bearing nominal life greatly (Taneja, 2012). Mainly because the additional dynamic force the bearing has to withstand.

Inadequate working condition as chatter (resulted from inadequate cutting parameters) can damage the bearings irreversibly. In the same line, a clash of the spindle with the workpiece can result in a overloading in the spindle bearings, resulting in permanent damage.

In summary, bearing rating life is may be difficult to calculate with accuracy and it still a topic for research. In fact, recent developments in modeling fatigue damage may be promising for calculating better bearing rating and incorporating in the basic formula ((Morales-Espejel & Gabelli, 2015)

2.1.5 Drawbar

The drawbar allows the clamping and unclamping the tool holder. This mechanism is embedded totally or partially in the spindle shaft depending on the configuration. Its function is lustrated by Figure 38, which shows a drawbar partially embedded in the shaft for steep taper (ISO taper) tool holder type. When the system is activated, hydraulically or pneumatically, the drawbar shaft is displaced inwards, loading the spring stack. At this moment, a griper (located at the front end of the drawbar) grips and pull the tool holder´s retention knob (or pull stud), against tooling interface on the spindle shaft. The drive key on the tool holder´s flange contact is adjusted against the shaft counterpart.

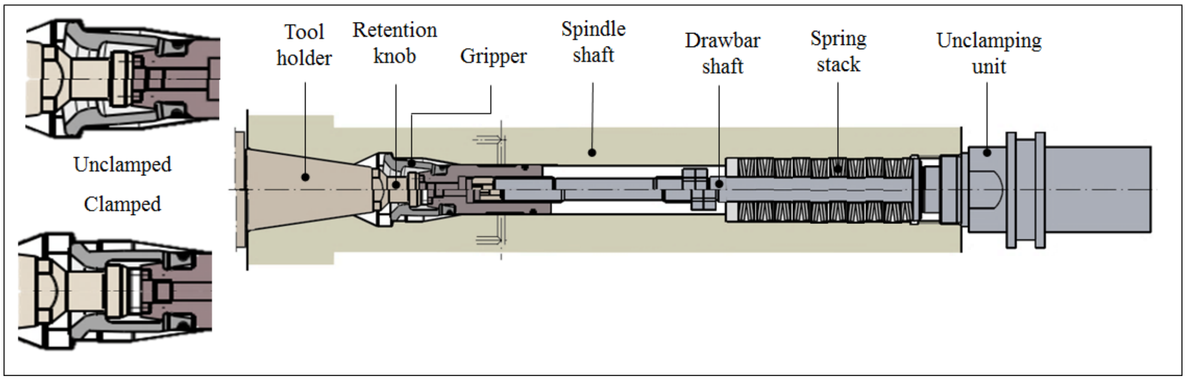


Figure 38 Drawbar under unclamped and clamped position inside the shaft. Adapted from (Otto-Jakob)

Depending on the drawbar characteristics, different transfer media can be supplied through the drawbar in several ways. For instead, during operation cutting coolant can be supplied axially to the tool holder's knob or axially to the tool holder flange as is shown in Figure 39. Besides air can be supplied for cleaning when tool is being changed.

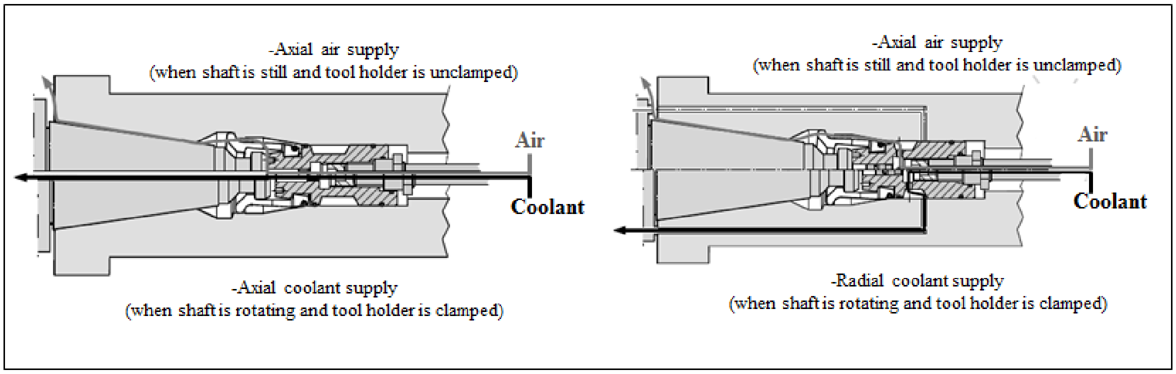


Figure 39 Media supply possibilities through the drawbar Adapted from (OTT-JAKOB Spanntechnik, 2013, p. 21)

In order to achieve good spindle performance at high speeds, higher clamping forces are necessary. The bigger the spindle and thus the tool holder, the larger pulling forces are required. Depending on the tool holder type, the pulling force of the drawbar can vary from about 5 up to 150 kN (TAC Rockford, 2015).

The drawbar, as well other components in the spindle, is susceptible to wear, specially their spring arrangement (commonly disk spring or Bellevile) which is constantly load and unload during tool changing. For this reason monitoring of the drawbar clamping forces are necessary to achieve proper machining results and avoid premature wear on the spindle and tool holder (Precision Spindle & Accesories Inc., 2015).

2.1.6 Tooling interface

The tool holder, in cutting process like milling, drilling or boring, is "…the interface between the cutting tool and the spindle and it ensures the transmission of the rotational movement of the spindle to the cutting tool" (Quintana, et al., 2009, p. 95) .

Strictly speaking, the tool holder does not form part of the spindle unit, but is determining in spindle performance during the cutting process. Once is clamped the tool holder can be considered as an extension of the spindle and therefore they are often studied together. It is important to remember that axial and radial forces as well as vibration will be transferred from the cutting zone to the spindle components through the tool holder.

The front end of the shaft will have a determine geometry to allow the clamping of tool holder. This geometry will depend of the type of tool holder chosen for the spindle. Figure 40 shows a typical tool holder and its main parts and how it is connected to the spindle shaft thanks to the drawbar.

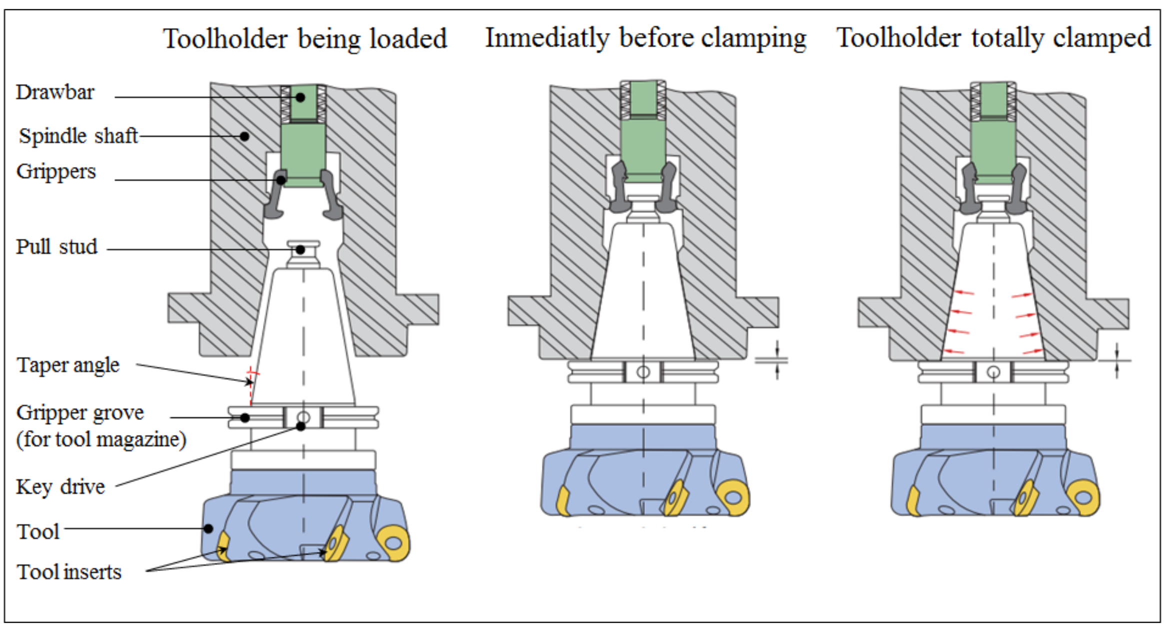


Figure 40 Tooling interface for a tool holder ISO. Adapted from (KENNAMETAL, 2013, p. 18)

Today there are a large number of tool holders in the market as a result of several developments by both companies and standardization bodies. The first known tool holder used was the Morse Taper, which dated from 1800’. Later in the 20’s, the ISO steep taper (or 7/24 ISO taper) was introduced and several variations were developed in the 60’s, which resulted in MAS-BT, ISO/DIN and CAT-V. These variations were introduced considering their use in Asia, Europe and America respectively. They basically differ in the dimensions of gripper groves and pull studs, but they all have the same taper angle. Thereafter, in the 90’s HSK type was develop by the German DIN. This tool holder differs from the ISO taper in different aspects: eliminate the need of pulling studs and increases the bending stiffness by allowing face contact of the flange. Besides it was possible to increase the clamping force required for high speed applications (Sandvik Coromant, 2015).

Apart from these solutions, other tool holder solutions have been developed by companies related to machine tools industry. This is the case of the Capto (patented by the Swedish-based tool manufacturer Sandvik Coromant). This solutions provides a polygonal- taper face connection, eliminating the need of key drives and allowing high clamping forces and improved torque transmission. Another example is the Big Plus introduce by the Japanese-based Big Daishowa, this is very similar to the ISO taper but provides flange contact thanks to improved dimensional tolerances.

	ISO/CAT	MAS BT	CAT-V	BIG PLUS	HSK	Capto©
Taper angle	16.26°	16.26°	16.26°	16.26°	5.7°	2.88°
Flange contact	NO	NO	NO	YES	YES	YES
Clamping method	Pull stud	Pull stud	Pull stud	Pull stud	Internal segment clamping	Internal segment clamping
Torque transmision	Drive keys on flange contact	Drive keys on flange contact	Drive keys on flange contact	Drive keys on flange contact	Drive keys on taper	Polygon

Figure 41Main tool holders types used in industry. Adapted from Sandvik Coromant

As it is with the spindle, the selection of the tool holder will depend on the application. The selection will depend on variables like operational speed, tool interchangeability requirements and multitasking capabilities of the spindle. Other factors as bending rigidity and torque transmission characteristics and balancing grade are also important (Sandvik Coromant, 2015).

Within every holder type there is a number of variations depending on the application and dimensions of the spindle interface. For instead HSK type holders are found in types A,B, C, D, E, F, T, and A/C/T in different diameters. Some of these types are for manual or automatic tool change, with different balancing grades, suitable different machining operations. In addition some allow coolant supply though the flange or through the center.

2.1.7 Encoders and sensors

Encoders are devices which provide information about the shaft position. They communicate with the Numerical Control System, allowing exact recording of the shaft position. This is especially important in turning operations when the spindle provide the rotatory movement to the workpiece.

Sensor, contrary to encoders can provide information about different physical units and not only position. In spindles, they can give information about clamping status of the tool, temperature of the bearings or motor, etc.

As it will be noticed later in this work, monitoring of machine tool condition is mainly carried out by with external equipment, which is often challenging. The main drawback of procedure is that some components cannot be monitored because of the difficult access due to machine design. For example, in some spindles the mounting of vibration sensors on back bearing housing is nearly impossible, because the "spindle box" armor almost the entire spindle. Hopefully the incorporation of several sensors is one main trend within spindle manufacturing industry (Abele, et al., 2010, p. 782). The aim is to facilitate continuous evaluation of the spindle status and also for monitoring the cutting process. Some of the sensors included are bearing and motor temperature, motor current and speed and vibration sensors.

2.2 Spindle vibration sources

Before exposing the main vibration sources in spindles, it would be reasonable to provide a definition of vibration. According to (Mobley, 2000, p. 24) "*…is a periodic motion, or one that repeats itself after a certain interval of time…*" . While (Rao, 2004, p. 13) states vibration is*" Any movement that repeats after an interval of time…*". Finally a more simple definition given by (Girdhar & Scheffe, 2004, p. 13) states that *"… is the motion of a machine or its part back and forth from its position of rest*.". From the above definitions, it is clear that mechanical vibration refers to movement.

Vibration is a movement often unwanted, considering the accurate performance at the cutting point expected from the spindle-tool assembly. As (Marsh, 2008) highlights "*An ideal spindle allows motion in a single degree of freedom: Pure rotation. Any movement in the remaining five degrees of freedom is undesired…*" Of course, this utopic notion of spindle would be impossible to achieve, because all the existing uncertainties in design, manufacturing and assembly process of the spindle and its individual components. But also because the thermal and dynamic influence when spindles are operating.

Spindle vibration or vibration between the tool and the workpiece can have negative effects in the cutting process. For instead, it can diminish the quality of a product by leading to poor surface finish. (Vafaei, et al., 2002) states "These untoward motions can in turn induce a host of problems such as inaccuracies and surface irregularities in the form and finish of the surfaces". Besides it can even damage the tool edge, reducing tool life (Rivin, 2002, p. 40.11). An extreme example of this is chatter, which is a regenerative vibration phenomenon with devastating effects on the workpiece, tool and spindle (Kennedy, 2004). For this reason vibration levels may be tolerated until at certain extend depending on the required performance of the spindle. In particular as Rivin expresses *"The tolerable level of relative vibration between tool and workpiece…is determined by the required surface finish,…machining accuracy and…. detrimental effect of the vibration on tool life…"* (Rivin, 2002, p. 40.1).

Based on the literature research conducted, several sources of vibration related to the spindle-tool system, were identified and shown in Figure 42.

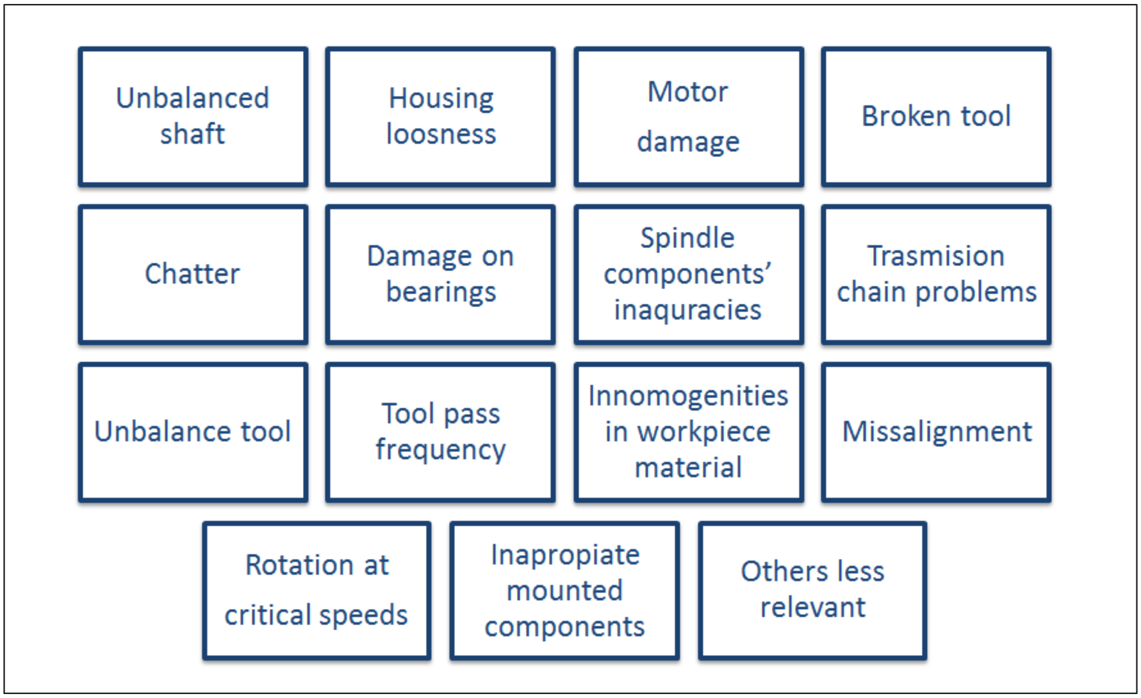


Figure 42 Main vibration sources on spindle

However the above list does not provide enough information about the nature of every vibration source. It would be necessary to classify these sources in a different manner in order to comprehend better their origin and facilitate its treatment.

One alternative classification of vibration often provided on vibration text-books is based on external energy sources, vibration can be classified as free vibration, forced vibration and self-excited vibration. A tentative classification of some of the mentioned sources it is shown in Figure 43. This classification may seem rather theoretical but in fact, it is useful for understanding better these sources.

Free vibration is related to shock or impact. For example hardness due to homogeneities in the workpiece material, may cause that the tool experience high impacts when machining. This will produce the tool to vibrate freely until the next cutting edge hit the workpiece.

Force vibration, means that the vibration it mainly determined by an external and periodic force acting on the structure or system. Unbalance is an example of this, because it repeats synchronized with shaft speed. Damage on bearing present also force vibration characteristics, because they repeat after a determined period of time, when the rolling elements hit the defects on the raceway, followed by a of the rolling element (free vibration).

The third type of vibration is self-excited and can be exemplified by chatter phenomenon which for definition is regenerative, this means that it is amplified in magnitude.

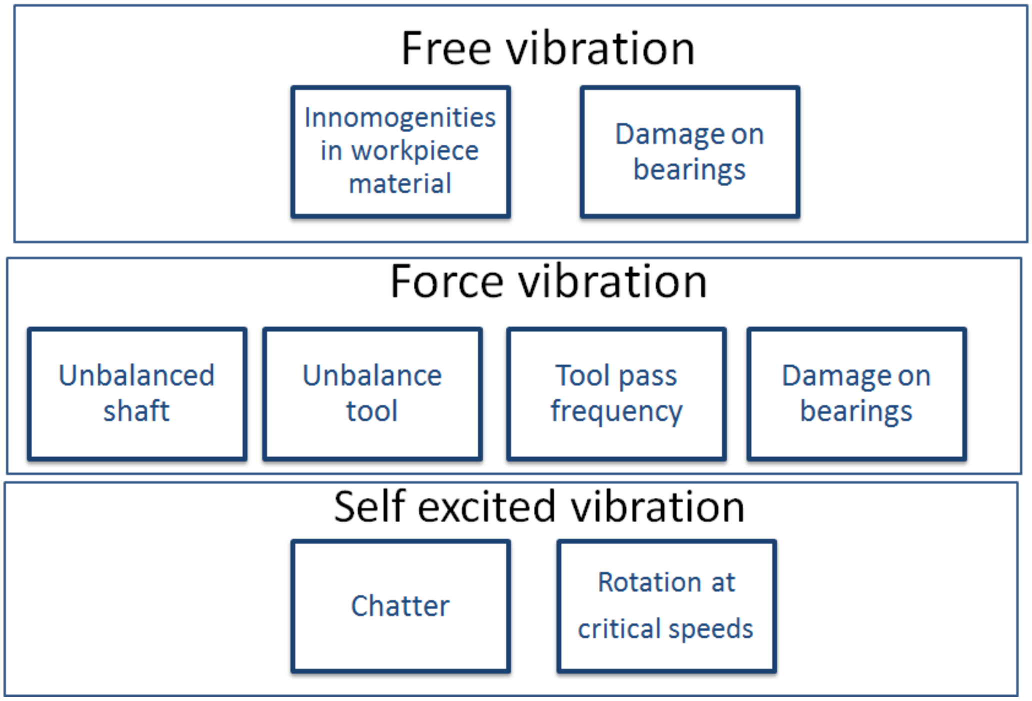


Figure 43 Tentative classification of some vibration sources in spindles

As the reader may notice, many source of vibration interact during the cutting process, which is not directly related to condition monitoring. However it is important to understand that these sources of vibration may determine in an important extend, the life of the spindle. For example according to (Castelbajac, 2016)., within the aerospace industry, it have been found that unforeseen condition in the process, as chatter or induced vibration near spindle harmonics (originated by tooth pass frequency) have been two of the main causes related to premature spindle failure. It can be said that is the response of the machine tool to working loads (expected or not) an important factor in evolution of deterioration in spindle components.

In Production engineering, the relation between manufacturing process and machine tool is known as Process Machine Interaction PMI. This concept is rather contemporary in Engineering and it may help engineers to understand better the process but also designers to have a more integrated approach when developing new machines and components. As (Brecher, et al., 2009, p. 589) points out the importance of PMI approach can be exemplified by the evolution in the understating of chatter phenomenon. The authors emphasize that chatter has been better comprehend, modeled and predicted by using PMI approach, instead of just analyzing the machine and the process separately. According to (Rivin, 2000 cited in (Archenti, 2014, p. 5) PMI approach is intended so “design, monitoring, control and optimization of statics and dynamics of machining must include the entire machining system and not only the process or machine tool, as separate entities”.

Figure 44 illustrates the concept of PMI approach looking at spindle as a component of interest. The cutting process generates loads on the machine structure and its components, in form of dynamic, static load and also “thermal load” meaning heat generated during the cutting process. The machine structure will respond to these loads in form of displacement due to elastic deformation of its structure due to load and thermal changes. This interaction is continuous basis. As Brecher et al. suggest, these interactions between process and machine will define workpiece quality, wear of machine elements and productivity of the machine.

By PMI perspective, it may be reasonable to consider that the condition of the spindle will affect in the way it respond during the cutting process. For example, high vibration (cyclical displacements) due to poor condition of the spindle may be transfer to the tool through the tooling system affecting the accuracy of cutting operation.

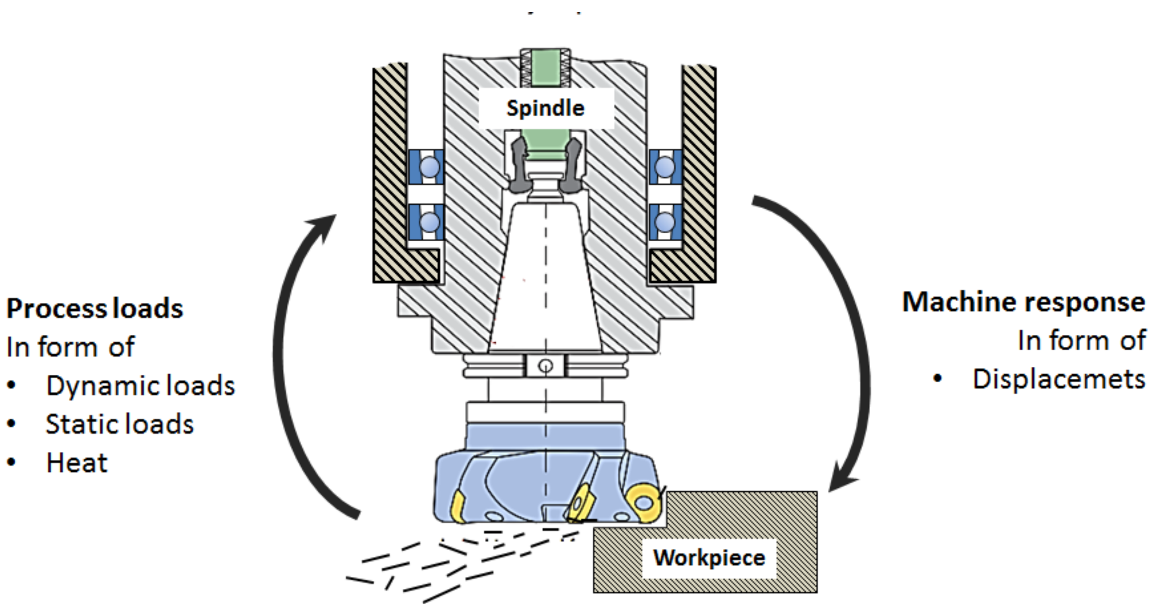


Figure 44 Interaction between process and machine tool. Adapted from (Brecher, et al., 2009)

It is important to highlight that vibration measurements in spindles can have different proposes and procedures which will be best suitable for each of them. For instead, for processing monitoring, it may be interesting to monitor vibration on the tool tip, workpiece or even bearing housing in a continuous basis. On the other side, spindle designers may investigate vibration response of the new spindle prototypes by conducting different tests on different structural parts of the assembly. Finally, for maintenance proposes as in condition monitoring, it may be suitable to carry out periodic vibration controls with defined running parameters (e.g. speed), which evidence the actual health of the spindle in order to facilitate planning maintenance task in advance in will the minimal possible cost.

Because of the time limitations, it is impossible to investigate each of the vibration sources related to spindle health. Many of the other sources which are not cover in this work may be neglectable when it comes to stablishing vibration limits for machine tool spindles. This is because they may have little influence in the overall magnitude of vibration and may be masked for other substantial sources as unbalance or bearing damage. Thus, based on the literature conducted, and the current standards relevant for machine tool spindles, three sources were identified and will be better explained next.

2.2.1 Imbalance in spindle-tool

Imbalance is probably the most common source of vibration in rotatory machinery according to (Scheffer & Girdhar, 2004, p. 90) . It is produced when the axis which the shaft is rotating around is not coincident with shaft´s axis of inertia (Stadelbauer, 2002, p. 39.2). It is also defined as 'The uneven distribution of mass about a rotor's rotating centerline' (Scheffer & Girdhar, 2004, p. 90). A final but important definition is given in the standard ISO 1925: "condition which exists in a rotor when vibration force or motion is imparted to its bearings as a result of centrifugal forces".

There is different type of imbalance and its classification seems to be more important for correction purposes than monitoring within rotatory machinery. However this classification will be briefly explained because it may help to understand better this vibration source. As Seffer & Girdhar point out, there are three types of imbalance. These are defined considering how the rotating axis (geometric centerline GCL) deviates with respect to the axis of inertia (principal inertia axis PIA).

-Static unbalance. PIA and GCL are parallel.
-Coupled unbalance PIA and GCL intersect in the center
-Dynamic unbalance PIA and GCL do not touch or coincide

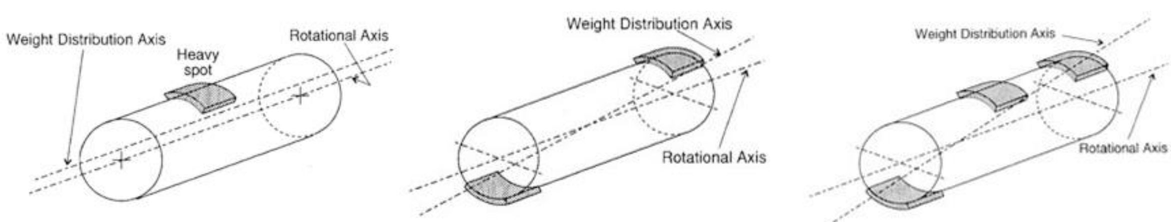


Figure 45 Classification of imbalance: Static, Dynamic and Coupled

Significant imbalance can reduce the life of shaft supporting bearings. An imbalanced shaft generates additional forces on bearings, increasing the equivalent dynamic load expressed in the basic bearing life formula. This dynamic load (commonly referred as unbalance force) is of centrifugal nature and therefore changes direction when shaft is rotating as it shown in Figure 46. For this reason it "synchronized" with the shaft rotational speed.

Therefore imbalance is easy detectable at rotational speed when vibration spectrum is analyzed as it will be explained in section 3.

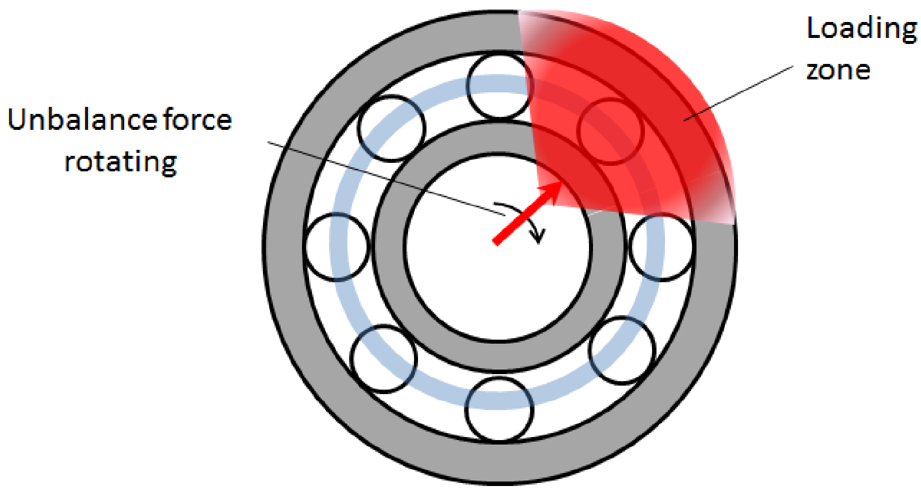


Figure 46 Unbalance force due to shaft imbalance acting on the bearing

These centrifugal forces will theoretically grow quadratic with rotational speed. Therefore balance tolerances are especially critical in motor spindles when high speeds are reached. Even though balance quality is not yet directly specified for machine tool spindles, balance tolerances for individual rotating components, are standardized by ISO 1940-1. This standard is aimed at providing a common base for understanding between users and manufacturers of rotating machinery. This is achieved by giving recommendation of balance quality for rotors (in rigid state) considering "the machinery type and maximum speed services' based on worldwide experience (ISO, 2003).

The explanation of standard ISO 1940-1 regarding balancing tolerances falls out from the scope of this work. However it will be briefly introduced considering its relevance for understanding unbalance in spindles. This standard specifies different balancing grades (from G04 to G4000) depending on the rotatory component of interest. The lower the G number, the more demanding balance will be expected. For example, for centrifugal pumps G6.3 is recommended, while machine tool drivers are required G2.5 and G1 for grinding machine drivers (see appendix). Based on this G number is possible to determine the acceptable residual unbalance (remaining unbalance after balancing), taking into consideration the maximum operating speed of the mechanical component being evaluated. Because unbalance force is speed-dependent, the larger the speed, the less residual unbalance will be tolerated with a fixed balancing grade. This residual unbalance (in [g mm]) can be calculated either with a formula or by a table (see appendix). In the case of using the table, residual in balance per kg must be later multiplied by rotor total weight. In both cases the rotor weight is taken into account (IRD Balancing, 2009). This means that more residual unbalance will be allowed on heavier rotors compared with lighter ones, for the same rotational speed and balancing grade.

There are several sources of imbalance in spindles-tool system. As Shen et al, suggest, every component of a spindle assembly can potentially be a source of imbalance. Operational factors and errors in design, material, manufacturing, assembly of the spindle components contribute to the total imbalance of the spindle-tool system. Figure shows some suggested source imbalance in spindle tool system according to (Shen, et al., 2011).

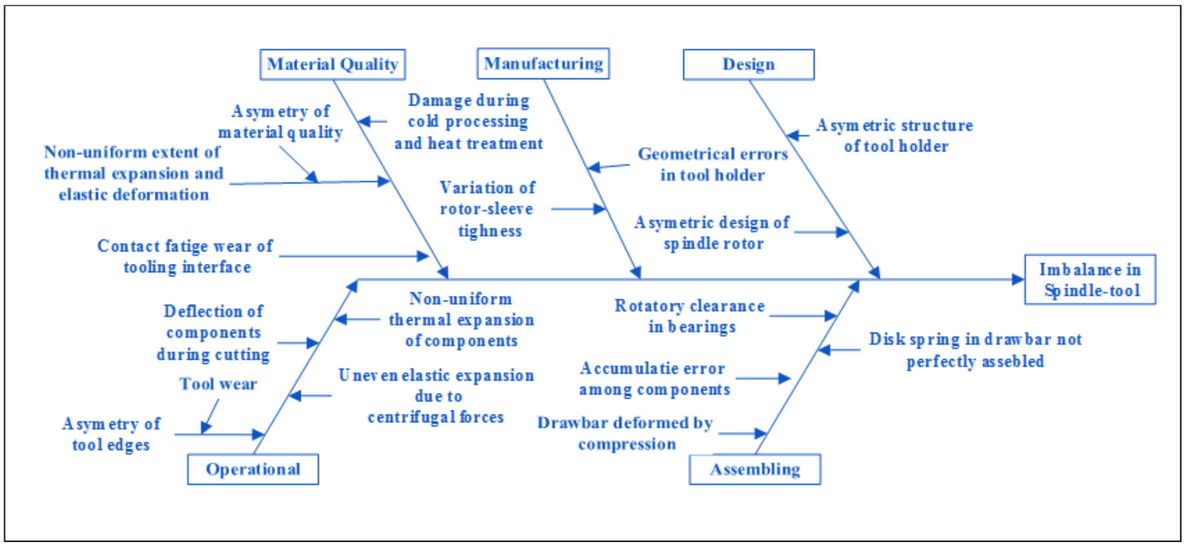


Figure 47 Possible causes of imbalance in spindles according to (Shen, et al., 2011)

In the mentioned work, the authors calculated theoretically the equivalent balancing grade of the spindle tool system, considering different balancing grades for three main components: spindle unit (meaning shaft-rotor), drawbar and tool system. Besides they considered a fixed balancing grade of G0.4 for the spindle unit , G1 to G16 for the drawbar and tooling system. In the worst case, this resulted in a balancing grade resulted in G2.97 for the entire spindle-tool system. Based on these results they concluded that because of the large mass of the spindle unit in comparison to drawbar and tooling system, the resulting balancing grade of the system is highly influenced by the spindle unit. Later the scholars experimented with the spindle running at different speeds with a tool capable of being imbalance-adjustable. They concluded that the imbalance (residual) on the tool, will increase significantly the vibration measured at the front bearing housing of the spindle.

Because tools are a source of imbalance they are often required a balancing grade of G 6.3 or even G2.5 according ISO 1940-1 but some critics have arisen concerning the applicability of this standard for tools in machine tool spindles. The use of these standards in the tool system, results in balancing tolerances difficult to achieve by the current balancing technology (Schulz & T., 1998). This is because; tool holders have a mass much lower compared to the rotors the standard is intended to. As a result, the allowed residual imbalanced moss restrictive for the tooling system. To illustrate this concern, consider a quality level of G 2.5 at 25 000 rpm, for a tool of 1 kg weight, this will correspond to a permissible residual unbalance of 1 g mm, which is equivalent to a permissible eccentricity of gravity of 1µm (Sandvik Coromant, 2015a). This requirement is even tougher for higher speed and lighter tools. Besides this (Sandvik Coromant, 2015a) (Schulz & T., 1998) point out that factors like constant tool changes, clamping inaccuracies and individual fit tolerances of components, prevent securing a repeatable balancing condition for the spindle-tool system.

Another problem with imbalance apart from dismissing bearing life is that leads to disruption in spindle shaft motion in (Fan, et al., 2013) .The authors in the mentioned work, measured the shaft motion in two directions while generating different degrees of imbalance in the spindle rotor by an electromagnetic balancer. The severity of this undesired motion due to unbalance can be illustrated by understanding the grinding process which required precision in the order of 1 µm (De a Calle & Lamikiz, 2009, p. 11). The imbalance in this case is mainly caused by the natural and sometimes uneven wearing of the grinding wheel. Therefore in order to prevent increase vibration at the cutting zone generated by this imbalance, grinding machines are often equipped with balancing systems.

In short, imbalance is one of the main sources in rotating machinery. Whether the imbalance come from the spindle unit or tooling system must be avoid securing precision during cutting and nominal life on bearings.

2.2.2 Bearing damage

As a difference with imbalance which produce vibration mainly at the same frequency of the shaft speed, damage on bearings produces often vibration at high frequencies in the vibration spectrum as Figure 48 illustrates. Their characteristics will depend on different factors. For example, the frequency which these defects will be visible will have direct relation with bearing geometry (eg. number of rolling elements, inner ring diameter, ball contact angle etc.). Another important factor which determines the frequency of this is the running speed of the shaft. At higher shaft speed, the defects will appear at higher frequencies. The extension of the damage in contrast; will be often related to magnitude of the vibration. In other words high vibration amplitude may be interpreted as more severe damage.

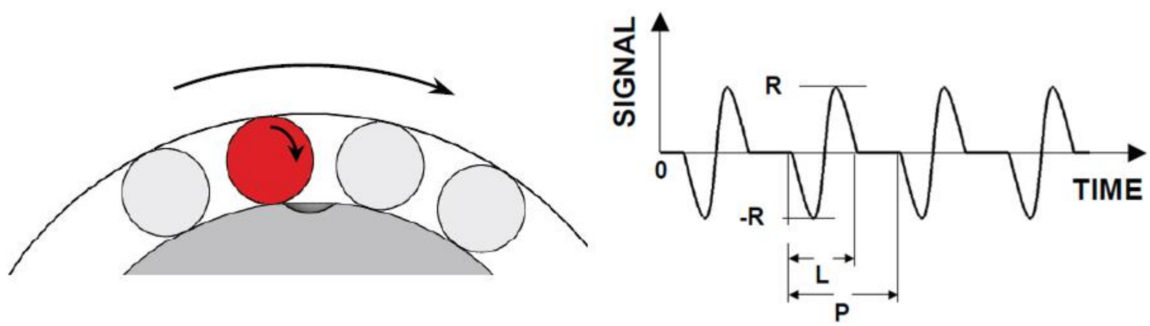


Figure 48 Acceleration signal generated by the passage of ball train over a raceway defect (Hoshi, 2006, p. 1)

Several authors suggest that bearing damage is the main cause of spindle failure. Within the industrial sector, spindle bearings are considered “the most sensitive components..” (Hendén, 2015). In fact, damage on bearings is far the most common (over 40%) within motors spindles according to (Abele & Korff, 2011, p. 426). Another study, within the aerospace industry, cited in (de Castelbajac, 2012, p. 27) indicated that bearing damage account for 60% of the motor spindle breakdowns. Because the importance of bearing damage on spindle life and thus condition monitoring, this will be further discussed in in next subsection.

Bearings can be damaged in many ways. Besides, this damage can be concentrated in a single or several of their components. These damages and their causes had been largely confusing. This is because every bearing manufacturer used to describe these issues in their own terms. Fortunately, the main twelve bearing manufacturers, including the Swedish-based SKF (Bergbom, 2015, p. 17) published in 2004 an ISO standard in this matter. ISO 15243 classifies damage, for rolling bearings, in six groups: as it seen in Figure 49. According to the standard six groups are identified: Fatigue, Wear, Corrosion, Electrical corrosion, Plastic deformation and finally Fracture and Cracking. Each of these groups are then divided in subgroups of two or three, were a more detailed classification is provided.

In the case of corrosion, this subgroup have even a third sub classification originated from frictional corrosion which is divided into “fretting corrosion” and “false brinelling”.

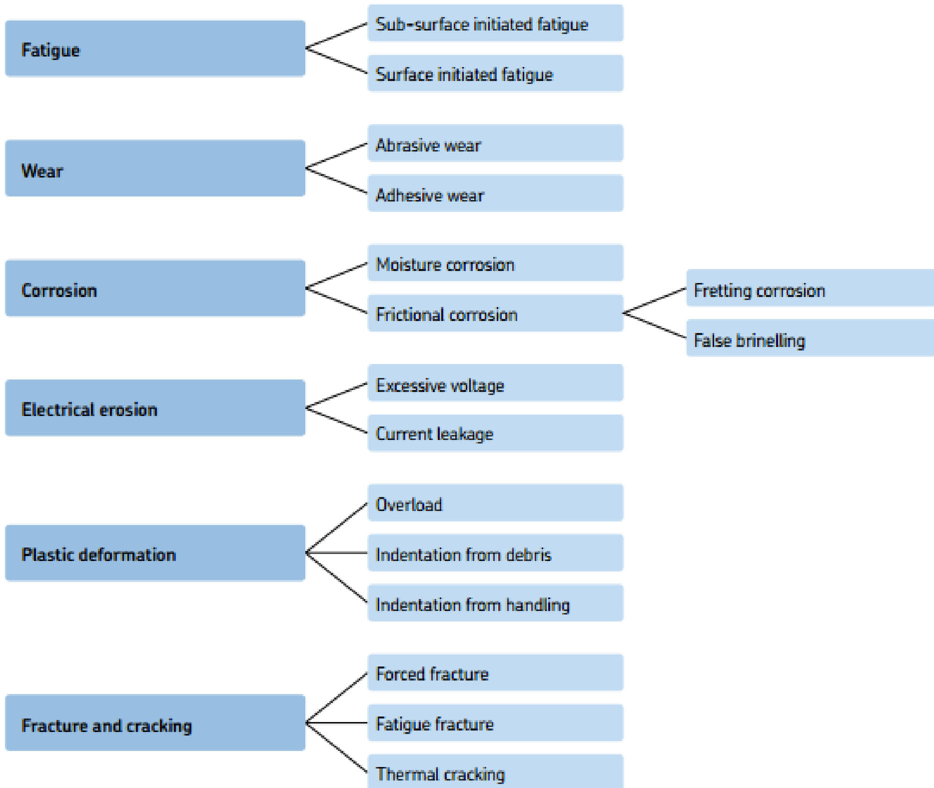


Figure 49 Classification of bearings damage according to ISO 15243

Figure 50 illustrate how this initial damage specified in ISO 15243 can develop over time. In this case, the inner ring has been affected by sub-surface fatigue. This type of damage can be expected after a long running time of the bearing under normal condition but even at early stage due to abnormal working condition. Regardless the root cause, this damage is originated due to repeated stresses. These stresses generate micro-cracks which are developed underneath and growth continuously until reaching the surfacing, leading to spalling (material break) (SKF, 2012, p. 127)

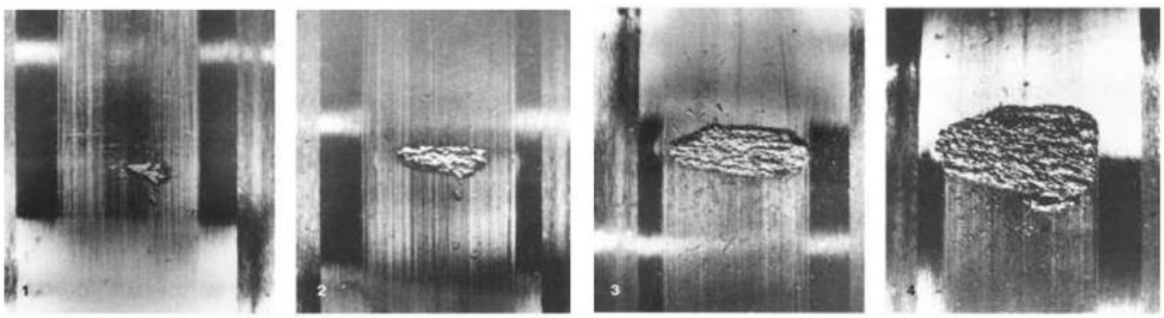


Figure 50 Development of sub-surface fatigue on the bearing inner raceway (McDermott, 2011, p. 37)

Even though initial damage on the bearings can be of one type described in ISO, the development of the damage can be very complex, involving other types of damage. This is usually the case when servicing spindles after breakdown. (Hendén, 2015) points out that is difficult to track back the initial damage in a bearing when the spindle is received for corrective maintenance. Usually because spindles are sent to repair shop when the damage has evolved and is extended after experiencing total failure. He explains that in these cases the spindle bearings are usually "burned up", which may refer to the high temperatures bearing experience when operating faulty at high speeds.

Even though it may be difficult to track initial damage in bearings once the spindle has failed, information gathered during process monitoring systems may be a better option. This is the approach that Castelbajac et al. used for understanding better the bearing damage in their work. In motor spindles for example, they reported that the deterioration of rolling elements may have been the result of cleavage development that escalated to wear (de Castelbajac, et al., 2014). Figure 51 illustrates the severity of the bearing of a spindle used monitored in the mentioned work. Specifically, it shows deterioration of a ceramic ball in a motor spindle bearing. The first picture to the left shows one of the bearing ball in its new state, while the last picture to the right shows the ball at the final stage once the spindle have failed. The raceways have been reported being damage as well due to the abrasive wear when of loose ceramic particles are passed by the rolling elements.

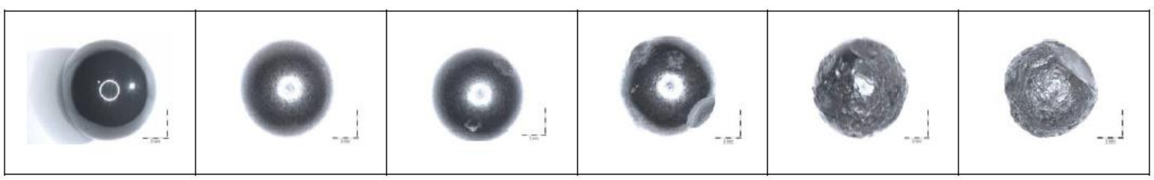


Figure 51 degradation of a ceramic ball in a motor spindle bearing until spindle failure (de Castelbajac, 2012, p. 133)

In general, premature damage on bearings can be attributed to pre-operational and operational causes. Possible pre-operational causes can involve careless handling or improper assembly. While operational refers mainly to abnormal operation conditions. This means conditions the bearing was not designed to cope with. For the interested reader, in the appendix it is found a matrix with possible root causes to damage in bearings.

In general, poor lubrication or contaminated lubricant is a common cause of bearing damage associated with the operational stage of the bearing life. These causes can lead to surface fatigue. This is because the contact surface of the rolling element and ring will no longer separated by a proper lubrication film (SKF, 2012). This contaminants can be from external source or even be the result of wear in the bearing components. In both cases theses particles can encrust in other components (e.g balls or raceways) generating further damage Figure 60

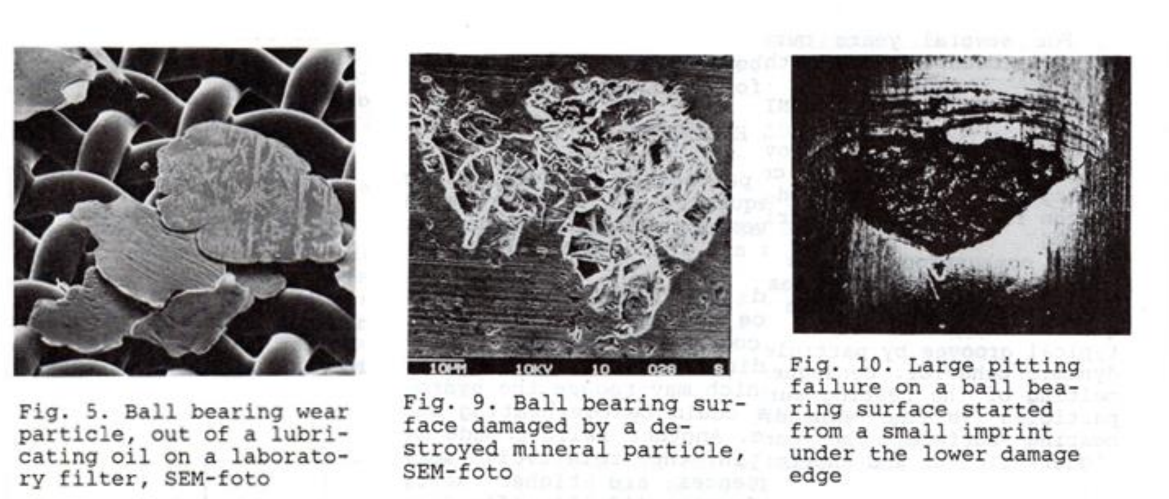


Figure 52 Contaminating particles on bearing and their potential effect in other bearing components (Jantzen, 1990, p. 8)

In machine tool spindles, one example of abnormal operational condition that can lead to premature fail on bearings is overloading due to collision of the shaft. In spindles, it is common that collisions compromise tool together with workpiece or clamping devices. In fact, collisions may be one of the main causes of spindle damage according to (Abele & Korff, 2011, p. 426). In their study, these authors claim that collisions account for 60% of damage cause on motor spindles (60 %). This collision may be caused mainly for errors in NC code when programing the machining operations. These collisions subject the shaft and thus spindle bearings to extreme high forces in a short period of time.

Another example of abnormal operating condition in spindles that results in bearing damage, is tool breakage. As (de Castelbajac, et al., 2014) point out in their real-time monitoring of three motor spindles under a period of seven months in aerospace industry, tool breakage was attributed as root cause in the front bearing of one of the spindle. The authors explain that during tool breakage (lasting about 0.5 seconds) high forces were generated which caused indentation of the ceramic balls on the steel raceway. This indentation result in plastic deformation. This damage is further developed by strong vibration due to cutting with broken tool (during several minutes before being discovered) and harmonics produced by passing tool frequency.

2.2.3 Critical speeds

Critical speed refers to the speeds when vibration levels become comparatively high due to forced excitation near to natural frequencies of shaft-bearing system. The American Petroleum Institute API defines critical speed as:

A shaft rotational speed that corresponds to the peak of a non critically damped (amplification factor > 2.5) rotor system resonance frequency. The frequency location of the critical speed is defined as the frequency of the peak vibration response as defined by a Bodé plot (for unbalance excitation).

The amplitude of vibration on these critical speeds will depend on the damping properties of the system (See Figure 53) If the damping is low, the system may not be able to dissipate energy thus high amplitude will be expected. On the other hand, if the damping is strong, the amplitude at that speed will not increase significantly or even become hardly noticeable.

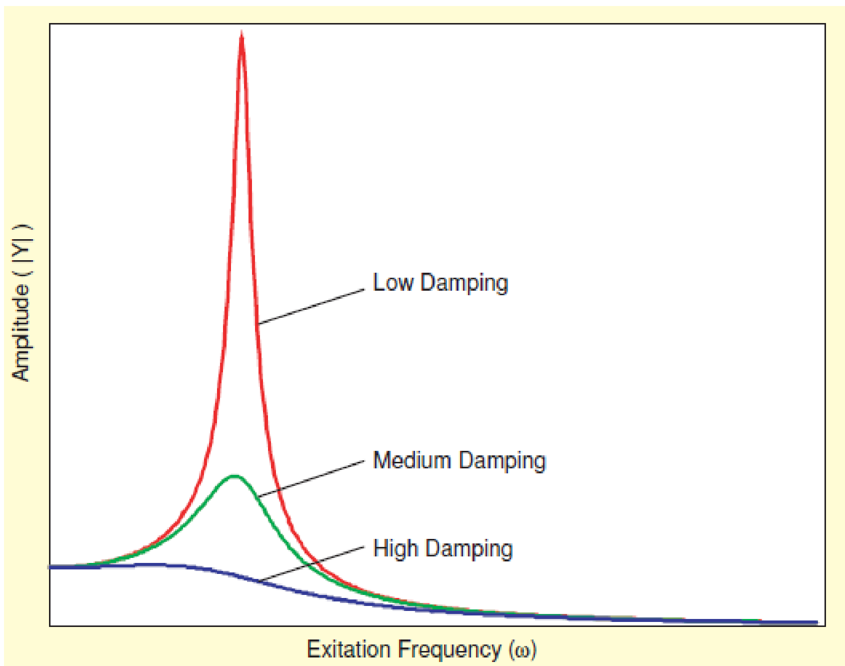


Figure 53 Effect of the damping properties on vibration amplitude of a system at critical speed

Critical speeds have mainly relation with how the system shaft bearings interact. The shaft-bearings system will present different mode shapes related to a specific natural frequency. The mode shape is strongly related to the rate bearing stiffness/shaft stiffness. If the bearing has high stiffness, the shaft may be considered flexible and will acquire a different shape mode than if the bearing had low stiffness.

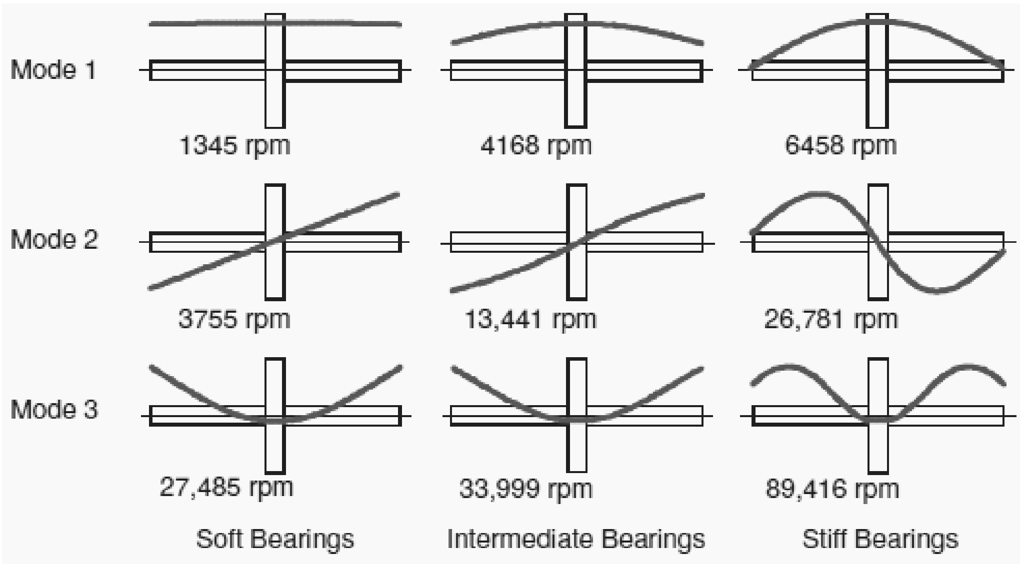


Figure 54 Examples of shapes mode v/s relative bearing stiffness for a simplified rotor system (Swanson, et al., 2005)

These modes (Swanson, et al., 2005) are often excited by imbalance. As mentioned earlier unbalance force often rotate synchronized with the shaft speed. For this reason the damped natural frequency of the rotor system can be reached easily detected with a speed sweep as Figure 55 illustrates. Rotatory machinery often presents as much as three critical speeds before reaching its maximum operating speed.

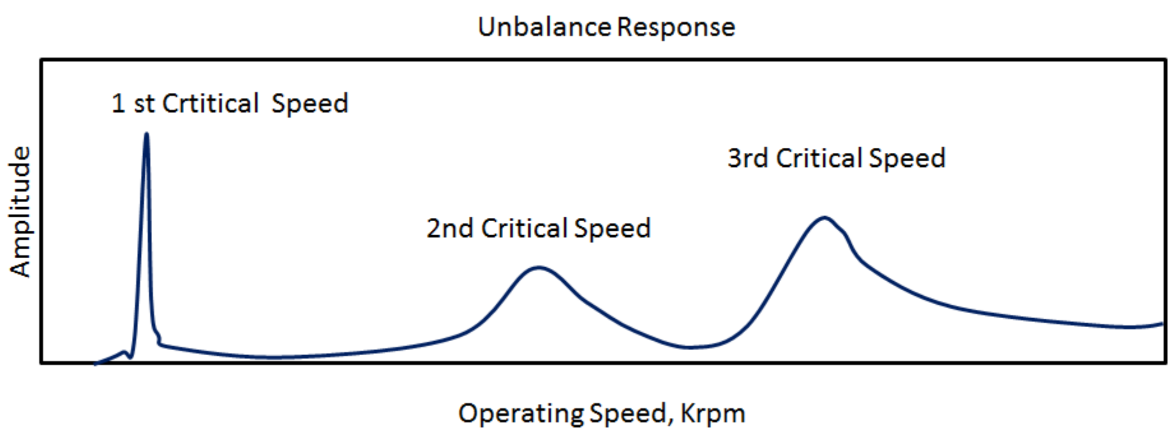


Figure 55 Critical speeds of a typical rotor system

2.3 Survey of bearing failure detection and diagnosis on spindles

Bearings are critical components in rotatory equipment and machine tool spindles are not the exception to this. As mentioned earlier, several researches and experienced professionals, attribute bearing damage as the main cause of spindle breakdown.

Luckily, bearing failure has been widely research over the last decades so there is plenty of literature research about the topic. Although only little research have been conducted focusing in bearing damage detection on machine tool spindles. This subsection intend to give and overview of different approaches for detecting and assessing bearing damage on spindles. The emphasis will be put in the methods, findings and the limitations on these studies.

Hoshi proposed Power (in g^2 units) of the vibration spectrum as an indicator for detecting localized bearings damage on rolling bearings (Hoshi, 2006). The researcher used a test bench where the test hybrid bearing was mounted and overloaded axially thanks to a hydraulic loading cell as it shown in Figure 56. Then, the driving shaft was run to high speed in order to generate a defect on the test bearing. The defect aimed to generate was small cavities caused by seizure phenomenon which resulted from a combination of everlasting and high temperature in the interface ball-raceway. All this procedure was carefully monitored by an accelerometer mounted axially in the spindle housing.

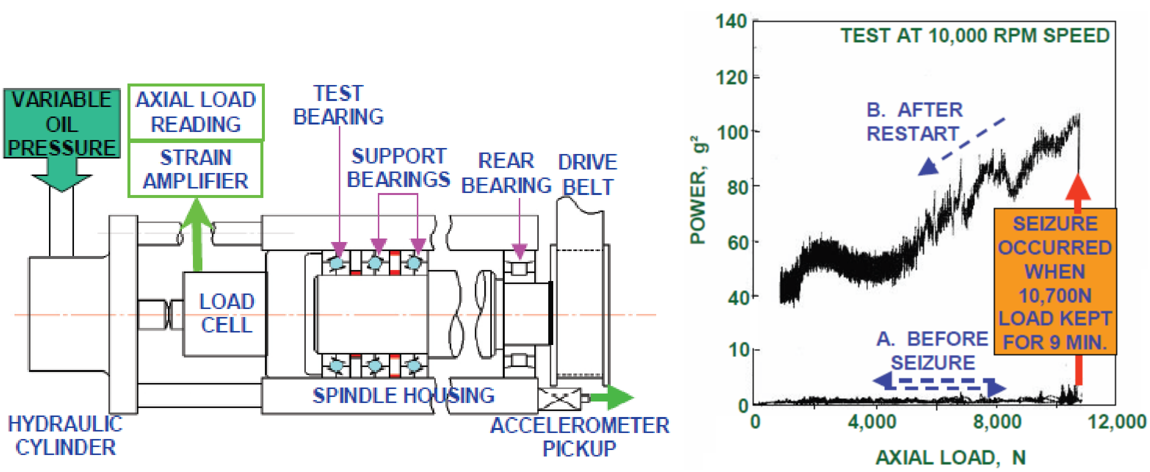
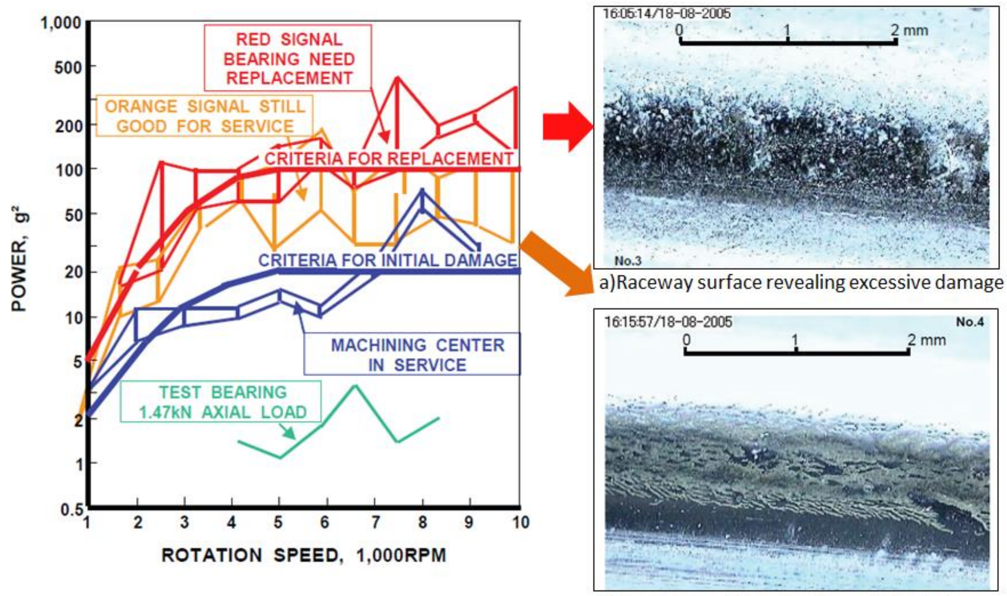


Figure 56 Experimental setup by (Hoshi, 2006) and failure detection

Using the proposed indicator and based on further tests on real machine tool spindles, Hoshi suggest four zones of damage for assessing bearing condition on spindles. One of the drawbacks of Power indicator used for Hoshi is its speed at lower speeds as it shown in Figure 57 (left).



b) Raceway surface revealing substantial damage but still good for service

Figure 57 Severity zones suggested by (Hoshi, 2006)

In another study (Neuebauer, et al., 2011) different monitoring techniques for spindles bearings are compared. These included acoustic emissions, vibration and temperature. In this case, as in Hoshi's work, the damage was artificially generated by overloading the spindle bearing. However in this work a motor spindle was used and static axial as well as radial, dynamic loads were applied as Figure 58 illustrates. The measurements of the spindle condition were made without load (idle speed of 3000 and 7000) weekly. The spindle was run until failure which resulted in 8600 hours which can be seen in Figure 59 left).

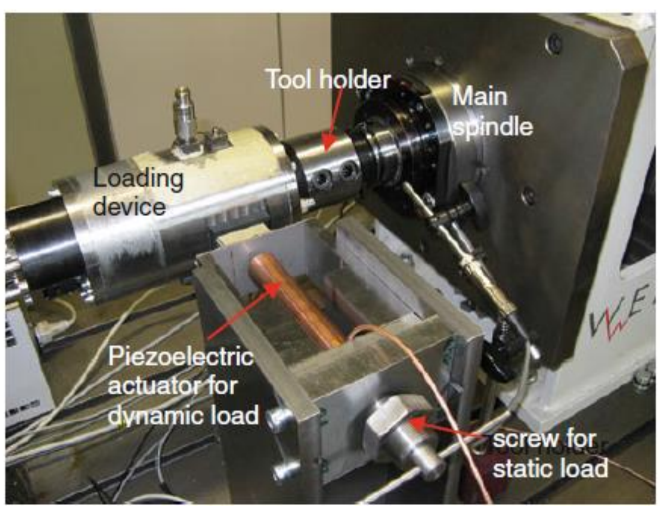


Figure 58 Experimental setup in (Neuebauer, et al., 2011)

The results suggest that by using rms of vibration velocity, the damage could be detected at less than 10% of the remaining life of the bearing. However by using envelope techniques of the vibration acceleration signal, this could be improved up to 15-20%. With the parameter suggested by the authors using acoustic emission AE, the damage can be detected at soon as 50% of the remaining life of the bearing. This resulted are summarized in Figure 59 (right).

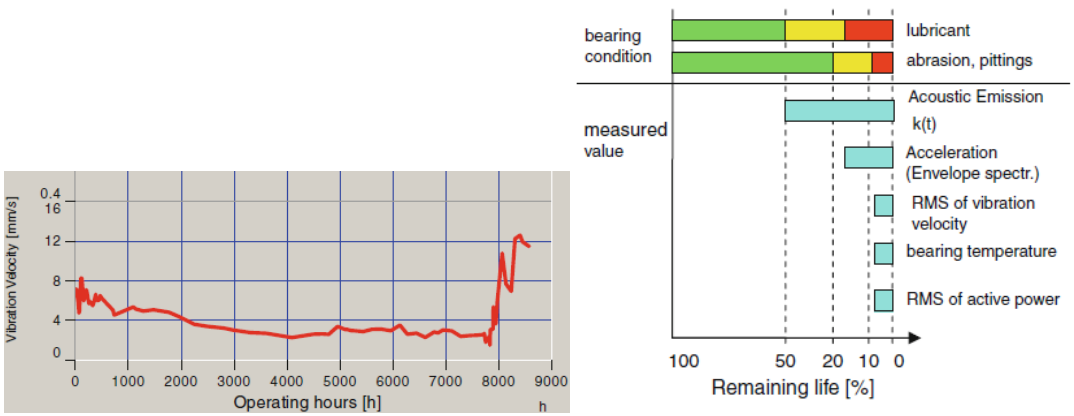


Figure 59 Velocity vibration trend until bearing failure. Performance results of different monitoring techniques (Neuebauer, et al., 2011)

In their work, Nauebauer et al. veryfied spindle failure (caused by bearing failure) by demounting and visually inspecting the spindle bearing as it shown in Figure 60

Figure 60 Failure on the inner and outer ring after disassembling spindle in (Neuebauer, et al., 2011)

Other researchers as (de Castelbajac, et al., 2014) have also proposed improved indicators to evaluate bearing conditions. In contrast to Hoshi and Neugebauer et al, De Castelbajac et al. carried out their investigation in an industrial environment with bearing failures originated by real operating conditions. The authors studied the vibration development of three motor spindles of the same model used in the aerospace industry. The spindles where used for milling structural parts of aircrafts. The mentioned spindles (24 krpm /70kW) were monitored by four accelerometers (two in the back and two in front spindle) close to the bearings arrangements. In this case the accelerometers had been embedded by the spindle manufacturer in its design as it shown in Figure 61.

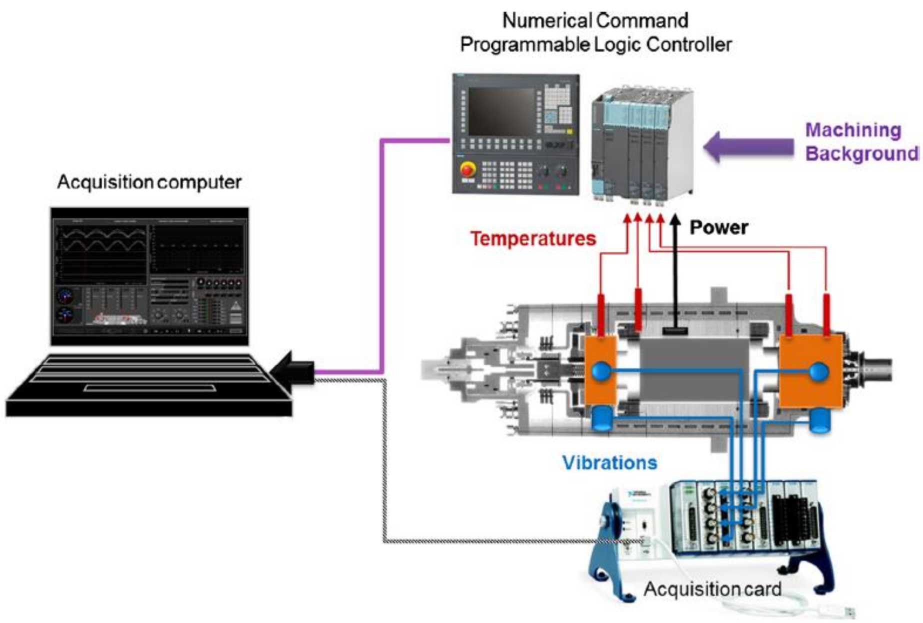


Figure 61 accelerometers placement on the spindle in (de Castelbajac, et al., 2014)

The authors monitored continuously the spindles until their failure (under several months). Vibration measurements were taken daily and automatically (triggered by PLC system) in moments when the spindle was not being utilized for cutting operations. Then, the spindles were run with a balanced dummy tool in four specific rotational speeds, close to the ones using during cutting. The speeds selected were (15, 17.5, 19.5, 21.1 and 23.75 krpm).

As mentioned before, authors of this work create and indicator for evaluating the severity of the vibration signature of the bearings, which was designated as Relevancy of Spindle Noise or SBN. This dimensionless indicator can be considered as the rate between initial and current vibration signature distribution of the spindle. In other words, a comparison between the vibration signature of the spindle in its new state and current condition. SBN is based in the assumption that the extended defects on the bearings will generate a noise component on the frequency spectrum of the vibration. They will be predominant in relation with other vibration sources.

The effectiveness of SBN in assessing bearing damage was validated by the authors through comparison of consistency against others indicators. The indicators SBN was compared against the most common severity indicators used in industry as vibration acceleration (rms) , kurtosis, crest factor and the magnitude of the ball pass frequency of the outer race of the bearing. (Some of them will be explained in section 3). The aforementioned researcher has measure bearing damage at specific speeds. This fact makes vibration values to be comparable during the life of the spindle.

The authors made this comparison based in three aspects: Firstly how the values disperse over the life of the spindle. Secondly the increasing trend over the time. And finally, the consistency when comparing the value of the three spindles once they have failed, as shown in Figure 61.

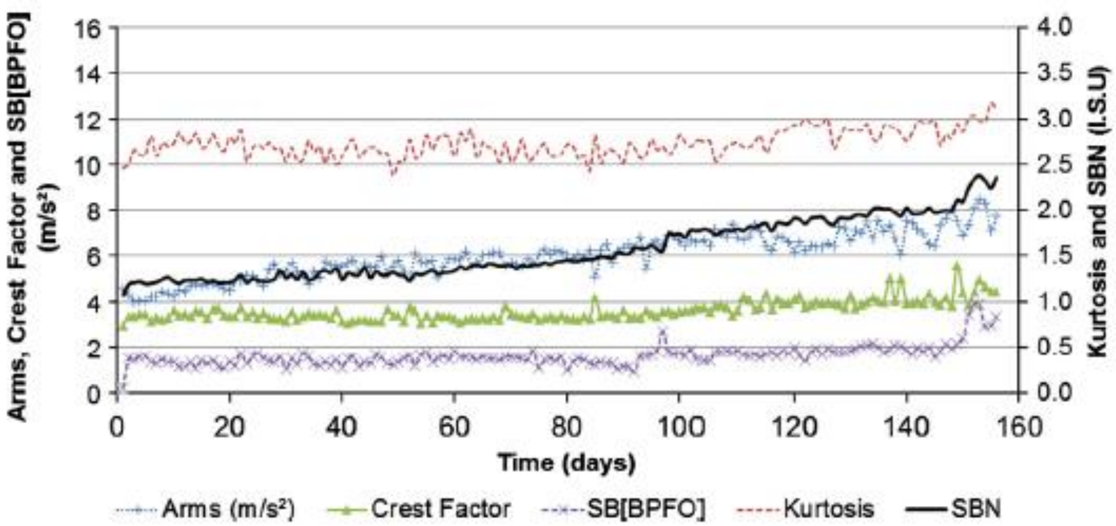


Figure 62 Comparison of SBN with other criteria for one of the spindle studied. The measurements correspond to the rear bearing in spindle 2. (de Castelbajac, et al., 2014, p. 167)[2]

Based on their results the researches claim has found a threshold for this spindle model. The results suggest a value of 2.5 for SBN indicator when the spindle is close to fail.

[2] By looking at Figure 62, it can be tempting to compare the rms value when the spindle failed, with the STSC indicator (in the Swedish standard SIS 728000-1). This because the spindle studied matches the characteristic of spindles the Swedish standard deals with (motor spindles with rolling bearings). However it must be noticed that this value correspond for the testing running speed of 15 000 rpm. Which may not be the speed when the maximum vibration level on the bearing housing is reached, considering that spindles often present its maximum value at its maximum speed. Besides, in the mentioned work, it not specified how the acceleration indicator was filtered. Interesting to notice is also how the rms value goes up and down with a rising overall trending.

Other academics (Vogl & Donmez, 2015) suggest a different approach than the previous works cited. These researchers developed a method and a metric to diagnose bearing condition on spindles. They authors suggest their metric (called Spindle Defect Metric) is comparatively better than other severity indicators used because only consider the condition of the bearing and is not speed dependent. The authors are critical regarding LTSC used in TR17243.-1 because according to them, this procedure will classify into C or D zone[3] “good condition” - spindles. They explained that indicators like LTSC can be “corrupted by the system dynamic”.

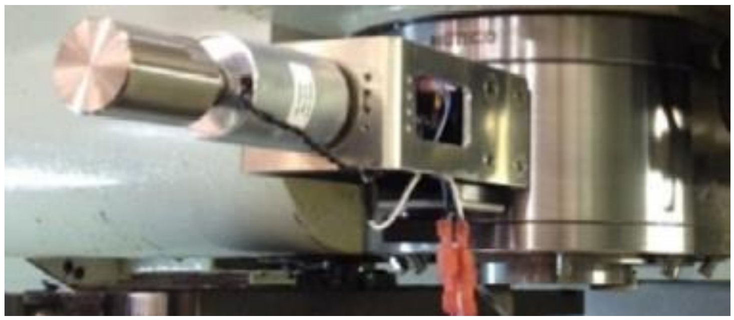
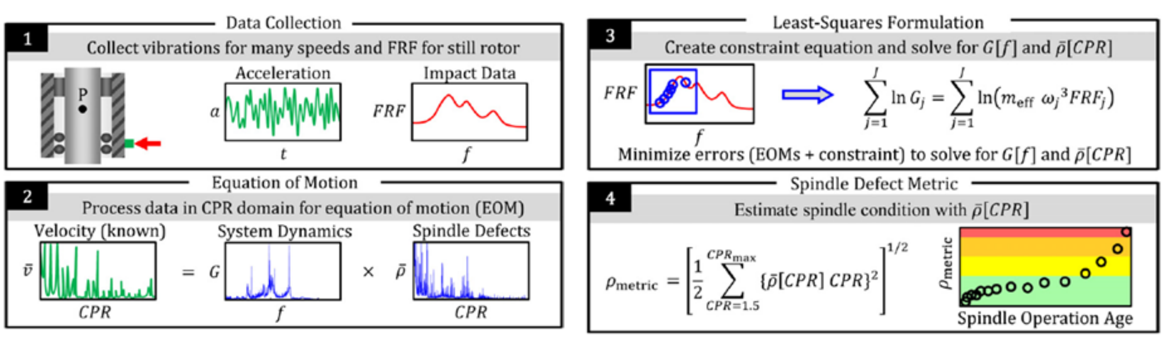


Figure 63 measurement carried out by (Vogl & Donmez, 2015) on the a spindle (front bearing housing)

The author start collecting their data with the instrumented (designated SCED) mounted magnetically on the spindle housing, close to the bearings as it shown I Figure 63. First impact test are carried out with non-rotation of the spindle to obtain FRF. Then vibration acceleration is collected within the whole operating speed of the spindle. In the next steps equation of motions are proposed, then solved to generate the indicator proposed with is expressed in longitude units µm. they authors explain that this mathematical formulation in necessary to remove the dynamic influence of rotor system when measuring vibrations on the housing.



$$\sum_{j=1}^{J} \ln G_j = \sum_{j=1}^{J} \ln(m_{\mathrm{eff}}\, \omega_j^{3} FRF_j)$$

$$\bar{v} = G \times \bar{\rho}$$

$$\rho_{\mathrm{metric}} = \left[\frac{1}{2}\sum_{CPR=1.5}^{CPR_{\max}} \{\bar{\rho}[CPR]\, CPR\}^2\right]^{1/2}$$

Figure 64Methodology by (Vogl & Donmez, 2015) to estimate spindle condition

[3] It is worth noting that the authors of the mentioned study compare their metric with LTSC of TR17243. However, as the TR 17243 states, this indicator (vibration velocity rms) is filtered 10-5000 Hz and it is aimed at detecting problems near to shaft speed frequency (misalignment, unbalance, etc). In contrast, the proposed metric is used to evaluate bearing condition. A more reasonable comparison could have been to compare the proposed metric against STSC (vibration acceleration rms) because it covers the frequency range of 2000-10 000 Hz and it aimed at reflecting bearing damage.

In fact they mainly validate their metric robustness by correlating it magnitude with the operating time of the spindle[4]. As it illustrated in Figure 65

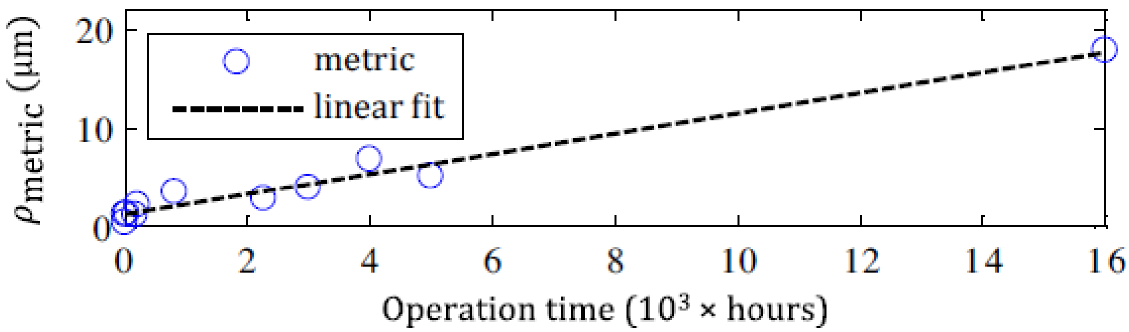


Figure 65 Validation of the metric proposed in (Vogl & Donmez, 2015)

Their method proposed by the approach is significantly more sophisticated and time demanding than the previous researches. This could difficult its implementation in an industrial context, taking into account the number of measurements necessary and the data processing tools. In addition, Vogl & Donmez's approach requires the custom instrument and software used to capture and process the data. Another difficulty observed for this method is the large size of the measuring device (compared with modern accelerometers), which may difficult its mounting on the back end of the spindle for condition monitoring of rear bearing. This part of the spindle is often hard to access.

[4] Apparently an example given in the mentioned work (Vogl & Donmez, 2015, p. 380) table 2, was accidentally omitted in Figure 64. In this case one of the milling spindle with 14 200 hours obtained a metric of 16.7 while a second spindle with approximately the same operating life (147 00 hours) obtained a metric of 7.02. This exemplifies however how very similar machines (same model and configuration) can have very different condition metrics.

3. VIBRATION MESUREMENT TECHNOLOGY AND SIGNAL ANALYSIS

3.1 Vibration units: Displacement, velocity and acceleration

As it has been mentioned, vibration can be measured in terms of displacement, velocity or acceleration. The selection of one or another will depend mainly in the frequency of the signal being measured. As it illustrated in Figure 66, low frequency signals reflect better its amplitude in terms of displacement. This is below 10 Hz (600 cycles per minute CPM). In the case of high frequency signals, "acceleration values yield more significant values than velocity or displacement". For this reason acceleration is preferred for signals over 1000-1500 Hz (60-90 kcpm). Finally velocity is preferred between 10 Hz and 100 Hz ((Scheffer & Girdhar, 2004, pp. 21,22). For this reason, bearing damage generate high frequency vibration signals is often expressed using acceleration units.

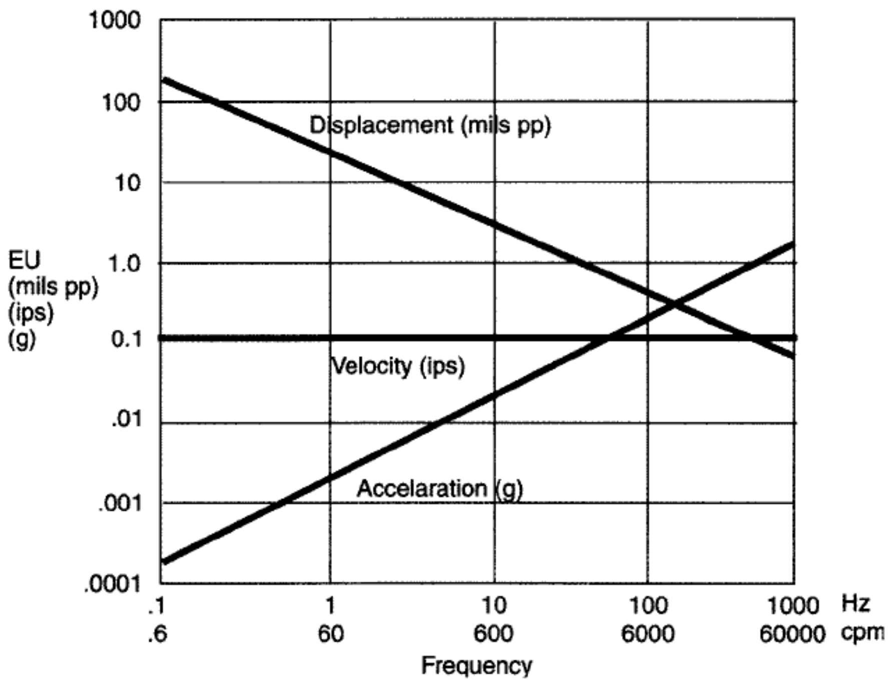


Figure 66 Relationship between displacement, velocity and acceleration at constant velocity.EU, engineering units. (Scheffer & Girdhar, 2004, p. 21)

3.2 Vibration Transducers and operating principles

The correct selection of vibration sensors is vital for ensuring an assertive diagnosis of spindle condition. It is known that different vibration signals are associated with different problems. Besides they often present different frequency characteristics. By knowing sensors' working principles, capabilities and limitations, a better decision can be taken when selecting them for the appropriate spindle defect to detect.

There is a wide range of transducers used for measuring vibrations. Below, the most common used in the industry for condition monitoring are described. The classification given is based on which physical unit is being measured (displacement, velocity or acceleration). However the main physical property measured can be later converted to the other units by simple signal processing. For instead, vibration collected using accelerometer will be originated in acceleration units (ex g or m/s^2), but this data can be converted lately into velocity units.

3.2.1 Displacement transducers

The most commonly used displacement transducers are of non-contact type and most of them operate on eddy current principle (Motalvao e Silva, 1990). This is illustrated in Figure 67. Eddy-current probe consist on a probe, an extension cable and oscillator/demodulator. A high frequency radio signal of 1-2 MHz is generated by de oscillator/demodulator. This is transmitted by the cable to the probe coil and generates a magnetic field, which radiate from the probe tip. When the probe tip is close proximity to an electrical conductive material (target), eddy current are produced in the target. This generates a decrease in amplitude of the probe excitation, which is related to specific DC voltage at the output of the oscillator demodulator (Adams, 2001, p. 233).

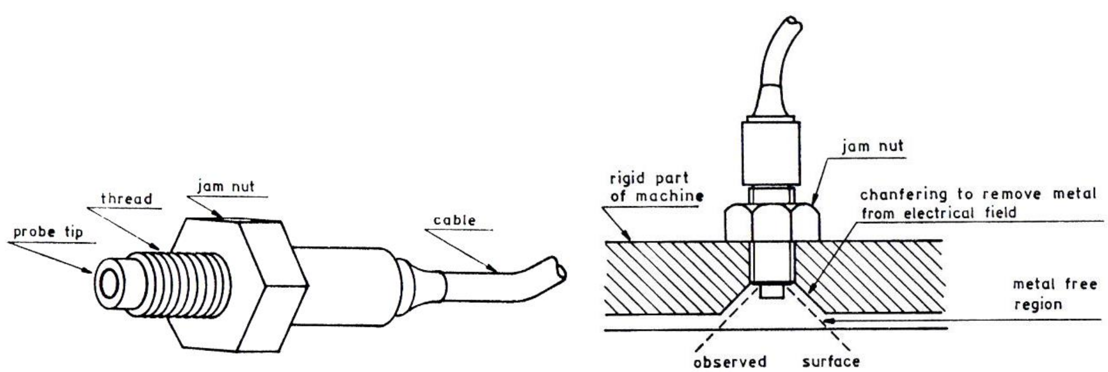


Figure 67 Typical eddy-current probe and its mounting (Motalvao e Silva, 1990)

By this way the distance between the probe tip and the target are related proportionally. According to (Motalvao e Silva, 1990) the resolution of this probes are usually 4mV/ µm or

8mV/ µm in the linear range and measuring range from 250-2250 µm. He also indicates that the length and diameter of the probe tip are in proportional to the measuring range.

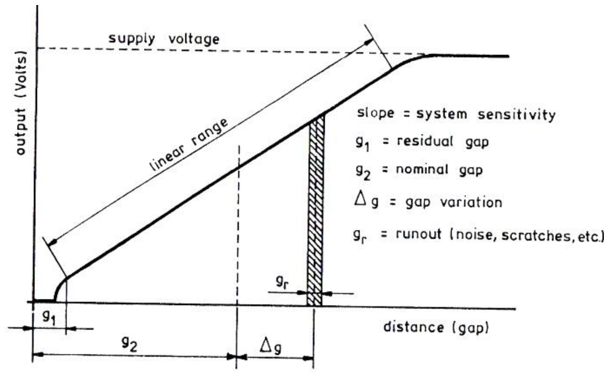


Figure 68Typical Eddy current probe sensitivity calibration curve (Motalvao e Silva, 1990)

Another displacement transducer is the capacitive-based. The main difference is that capacitive sensor provided comparatively higher resolution than eddy-current sensor. However these are not suitable for dirty or wet environment in which eddy-current are preferred. Besides, capacitive sensors are not suitable when large gap between sensor and target is required. In this case, laser sensor could be a better option (Lion Precision, 2015).

As its name indicates, capacitive-displacement transducers work with capacitive principle. This means that displacement D is indirectly determined by measuring the capacitance C between the target and the probe (sensor). See Figure 69. The capacitance is an electric property affected by the area, dielectric and the distance between the two objects. In particular, capacitive probes work with the assumption that only the distance is changing between the target and probe. The distance is inverse proportional to the capacitance. The larger the distance between sensor and target the lower the capacitance will be.

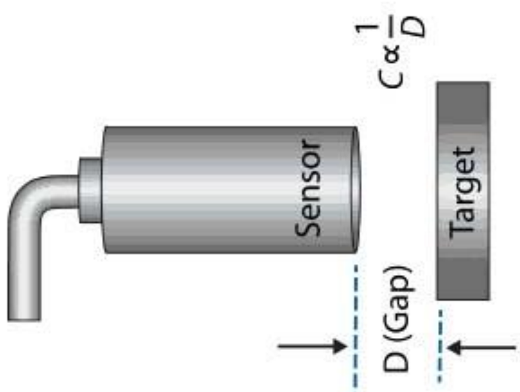


Figure 69 Capacitive sensor principle

In order to operate, a voltage is applied to the capacitive probe which creates and electric field, the extension of this electric field will determine the measuring range of the probe. This electric field is reshaped by a guard to only allow the front part of the probe to be sensitive.

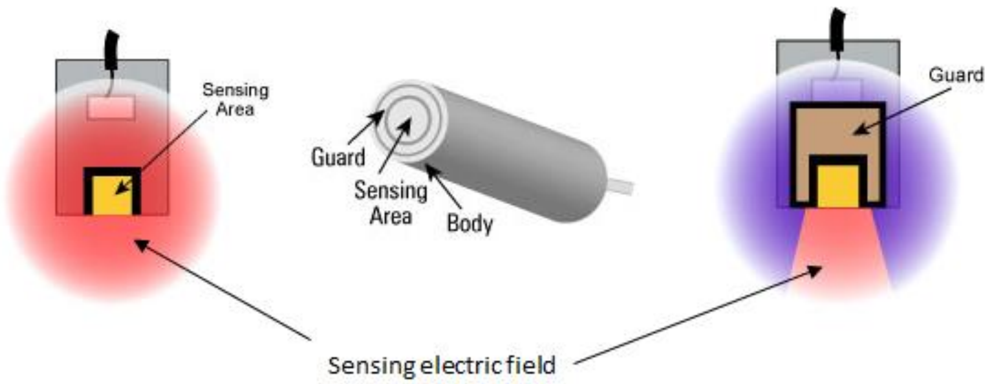


Figure 70 Capacitive probe. (a) Without guard (b) parts (c) with guard . Adapted from [Lion Precision]

The characteristic of the sensor (measuring range, sensitivity, resolution etc.) will depend of the dimensions of the probe. The range for example will depend mainly on the size on the sensing area. In other words, the bigger the probe, the larger displacement could be measured. As a rule of thumb, the sensing range is approximately 40% of the diameter of the sensing area.

The performance of capacitive probes can be significantly improved by calibration with the driver system. The driver system (also called module) refers to the electronic module that gives excitation voltage to the probe. And receive the output voltage. The driver system is connected with the probe by cable (Lion Precision, 2015).

3.2.2 Velocity transducers

Velocity transducers are based on electromagnetic principles. They consist in a mass that is a permanent magnet and it is suspended by one or two springs by its ends. This mass is immerse in a fluid and surrounded by an electrical coil. This fluid provides critical damping to the system and prevents the mass from moving when operating. In contrast to the mass, the transducer's case move back and forth producing a relative movement of the magnet. This induce a voltage in the coil with it proportional to the velocity of the case which the transducer is firmly attached to (Adams, 2001, p. 231).

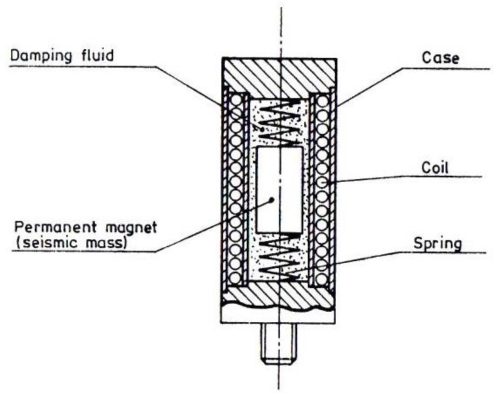


Figure 71 schematic of a velocity transducer

These types of transducer operate typically in the range of 10-1500 Hz, which correspond to a range above its natural frequency (Adams, 2001, p. 232) as Figure 72 illustrates. When it comes to sensitivity, this is often expressed in mV/mm/s. standard values are in the range of 20-30 mV/mm/s according to (Scheffer & Girdhar, 2004, p. 31)

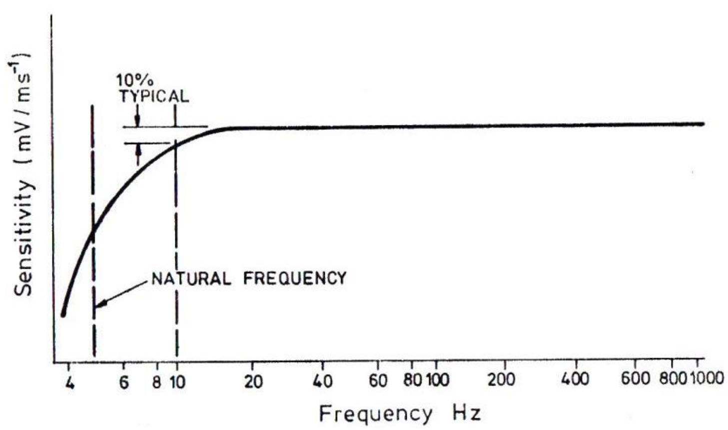


Figure 72 Typical sensitive curve for a velocity transducer (Motalvao e Silva, 1990)

3.3.3 Acceleration transducers

There are different types of accelerometers depending on its working principle. For instead, there are Piezoelectric, Capacitive, Piezoresistive, Hall Effect, Magnetoresistive and Heat transfer based (Britton, 2005, p. 19). However piezoelectric accelerometers are the most common used in the industry because their relatively good performance-cost rate.

Piezoelectric-based accelerometers internal mechanism consist in an internal mass compressed (by a spring or ring) in contact with a load cell (piezoelectric crystal) as Figure 73 illustrates. Once mounted, the base of the accelerometers is in contact with the vibrating surface. When the mass is subjected to vibration, it compress the piezoelectric crystal which produce a varying charge, proportional to this compressive force. This charge is then transformed into voltage output and amplified, usually thanks to electronics embedded in the accelerometers. When this is the case, these accelerometers denominated as Integrated Circuit Piezoelectric or simply ICP (Smith, 2013).

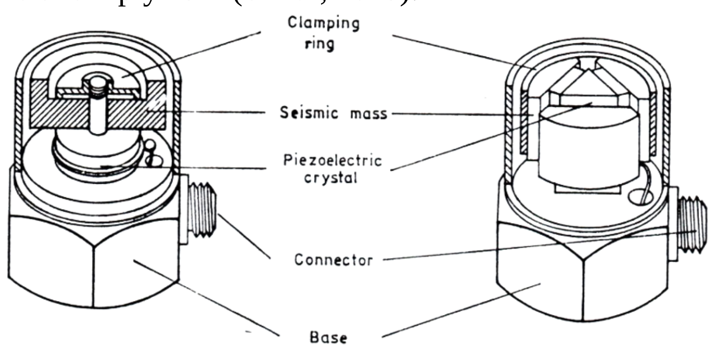


Figure 73 Accelerometer main components. A) Compression type and b) Shear type (Motalvao e Silva, 1990)

The frequency range of an accelerometer is limited by its resonance frequency when mounted as it is shown in Figure 74. However as it (Scheffer & Girdhar, 2004) points out, accelerometers have typically a frequency range from 1-2 Hz to 8 or even 10 kHz. And a sensibility of 100 mV/g. However special accelerometers with higher frequency range and sensibility are also available.

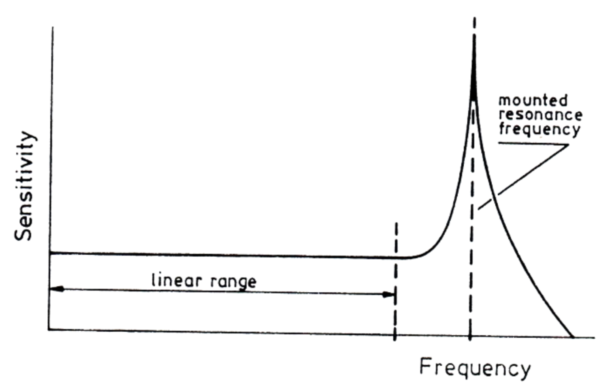


Figure 74 Typical accelerometer sensitive curve

In order to ensure linearity in the electrical output with the vibration measured (Adams, 2001) recommends that internal natural frequency of the accelerometers, should be at least five times higher than the maximum end of its intended frequency range of use. However, this could be considered very conservative. Other estate that the upper limit of the usable range can be estimated roughly as 1/3 of the natural frequency of the accelerometer (Dytran Instruments Inc., s.f.). Adams also explains that in order to increase the frequency range of an accelerometer, the accelerometer's internal mass should be decreased. But by decreasing the mass, lower peaks load-cell are obtained and therefore the resolution of the accelerometers is compromised. Despite these issues, he highlights that still accelerometers are the best option when measuring high frequencies signals.

In the table the main characteristics must be considered when selecting accelerometers

Table 1 Characteristic factors to be considered when selecting and accelerometers. Based on (Albarbar, et al., 2008, pp. 3-4

Characteristic Factors	Units	Typical values[5]	Interpretation/Comments
Sensitivity	[mV/g]	100	Represent the ratio of its electrical output to its mechanical input.
Amplitude limit	[g]	+- 50	Refers to the maximum range of measurable acceleration.
Shock limit	[g pk]	5000 g	Indicates the acceleration can be withstood without being damaged. Important if accelerometer is drooped!!!
Natural Frequency	[kHz]	25	It determines the useful range of frequency of the accelerometer.
Resolution	[mg]	3	How the smallest change in mechanical input reflect in electrical output. Its depend not only on accelerometer, but the measuring arrangement[6].
Amplitude linearity	%	+- 1	It reflects the degree of accuracy that accelerometer report within its frequency range.
Frequency range	[Hz]	0.5-10000	Limits in which the accelerometer presents the sensitivity specified at most. Also depend on measuring arrangement.
Phase Shift	Ms?		Time delay between the mechanical input and the corresponding electrical output signal of the instrumentation system.

[5] Just referenced values for a typical industrial ICP accelerometer used in this work. This values can varied significantly depending on the model of the accelerometer.
[6] With measuring arrangement the author means, mounting method, auxiliary equipment, background noise etc.

Other factors mentioned by (Albarbar, et al., 2008) and accelerometers suppliers includes:

- Environmental factors
- Sensor mounting options
- Mounted resonant frequency
- Grounding (isolated non isolated)
- Transverse sensitivity
- Mechanical resistance to wear, moisture
- Dimensions
- Weight
- Electrical connection position
- Spectral noise (at different frequency bands)

3.3.4 Micro electromechanical systems MEMS

In the last decades, advances of nanotechnology and electronics have result in the development of comparably small sensors with promising characteristics for condition monitoring of equipment. One of these sensors is the so called MEMS or microelectromechanical systems. Depending of their configuration, these devices are capable of measuring displacement, velocity, acceleration and many others parameters.

Two types of MEMS accelerometers are usually identified: Piezoresistive and capacitive-based. See Figure 75.(Albarbar, et al., 2008) explains: 'The MEMS accelerometer has a cantilever beam with a proof mass at the beam tip and a Piezoresistive patch on the beam web. The inertia of the mass causes a change in the gap between the mass and the bulk of the device made of the silicon wafer when the device is subjected to acceleration. The mass may move out of the plane of the silicon wafer or in the plane…The electric signal generated from the Piezoresistive patch and the bulk device due to vibration is proportional to the acceleration of the vibrating object'. While 'Capacitive based MEMS accelerometers measure changes of the capacitance between a proof mass and a fixed conductive electrode separated by a narrow gap'

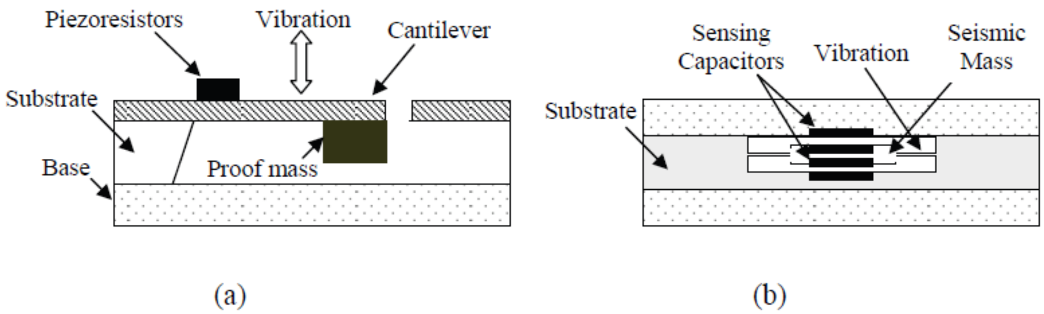


Figure 75 Typical MEMS accelerometers construction.. (a)Piezoresistive (cantilever design). (b)capacitive (membrane design) (Plaza et al. cited in (Albarbar, et al., 2008))

Some studies as in (Albarbar, et al., 2008) have explored the potential of MEMS for condition monitoring of machine tools with satisfactory results. Other authors (Ramachandran, et al., 2010) point out the possibilities of MEMS for multisensory condition monitoring, this means measuring different variables using the same sensor (eg. temperature, pressure, vibration etc..). They add that MEMS can be an effective solution for wireless and online monitoring. (Looney, 2014) emphasizes that in comparison with conventional piezoelectric sensors, MEMS are less expensive and integrated with additional devices can be programmed to automatically take measurements. Besides they can transmit process data using wireless signals.

3.3 Mounting method for accelerometers

The mounting of the sensor will also influence the upper frequency range of the accelerometer. As it shown in the Figure 76 stud mounting provide the widest frequency response and most reliable attachment. However this type of mounting may not be always possible and therefore other options would be more suitable.

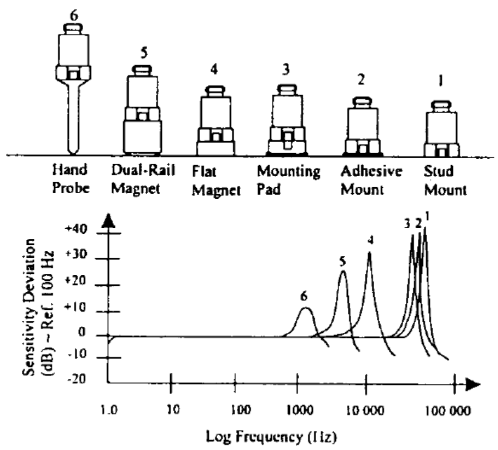


Figure 76 Different mounting methods for accelerometers and their effect on high frequency (IMI SENSORS, s.f.)

3.3.1 Not mount

It refers to the possibility of holding the accelerometer with the hand, when measuring. This is a very practical and easy method but the frequency range of the accelerometer is compromised. With this method the usable upper frequency range of the accelerometer can be drastically reduced. For this reason, it may not suitable when measuring high frequency vibrations, but it could be reasonable for detecting problems as unbalance, misalignment which are detectable in the low part of the frequency spectrum of the vibration signature.

3.3.2 Magnetic mounting

This method is preferable when carrying out portable measurements. The base of the accelerometers can have different shape to better accommodate to the surface. For example it can be flat or dual-rail. Flat magnet will better accommodate in flat surfaces, while dual-rail magnet are suitable for curved surfaces as Figure 77 illustrates. Of course the mentioned variants will require that the surface is magnetic. If this is not the case, the sensor can be mounted in a steel mounting pad. This pad can then be attached to the surface by other means (epoxy, welding, etc.)

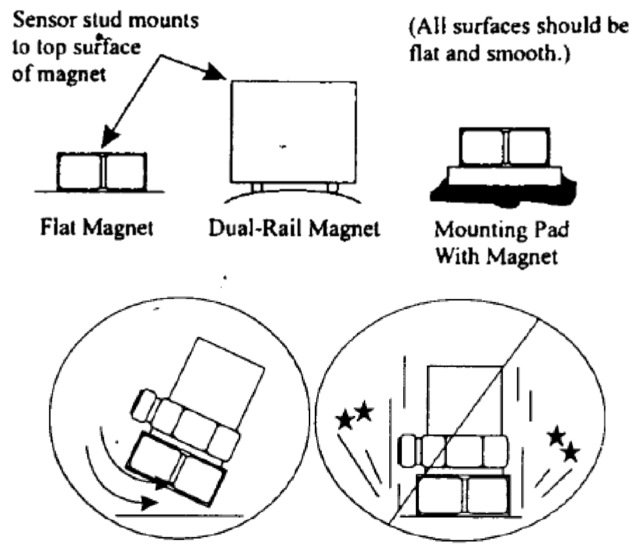


Figure 77 types of magnetic mounting and adequate mounting procedure (IMI SENSORS, s.f.)

3.3.3 Adhesive mounting

This method has the advantage of ensuring a secure attachment. The adhesive can be epoxy, glue or wax. Glue adhesive requires the mounting surface be very clean and good finishing. On the other hand, it can reduce the accuracy and maximum and the operational frequency response range of the accelerometer. Besides compared to the other methods, the replacement of removal of the accelerometer is specially time consuming (Scheffer & Girdhar, 2004, p. 34). Another alternative is wax, which is comparison with glue, allows easier removal of the accelerometer.

Figure 78 shows possibilities when installing accelerometers using adhesive methods. In the (a) example the accelerometers is mounted directly using adhesive. In (b) an adapter between the accelerometer and the surface is used instead. (c) Illustrate a common mistake when using this technique: excessive thickness of the glue layer, this create and own mechanical system illustrated in (d). The same effect can occur when the surfaces (either base of accelerometer or mounting surface) are not flat (Dytran Instruments Inc., s.f.).

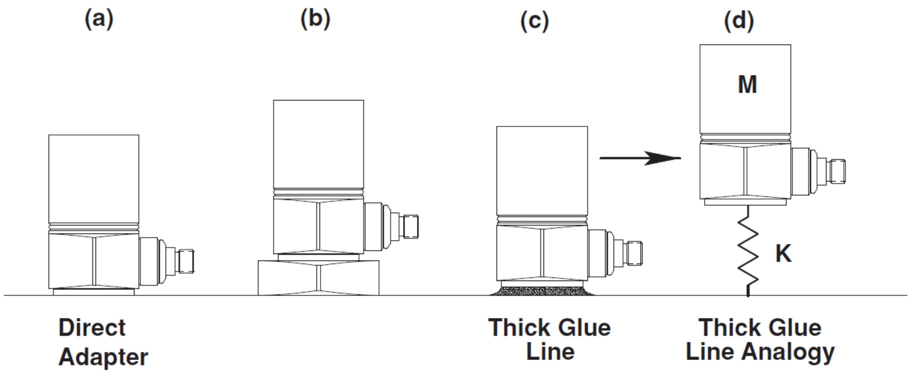


Figure 78 Typical options when using adhesive mounting (Dytran Instruments Inc., s.f.)

3.3.4 Stud mounting

This is used for permanent mounting application because it required at thread hole where the accelerometer can be screwed into. For this proposed the accelerometer can be equipped with a permanent or removable stud depending on the model See Figure 79. With this mounting technique, the surface is required to be flat, smooth and even for securing sufficient surface contact and prevent misalignment (IMI SENSORS, s.f.). When using this technique, excellent surface quality must be ensured. Besides, the accelerometer must be adjusted with the specific mounting torque by means of a torque wrench.

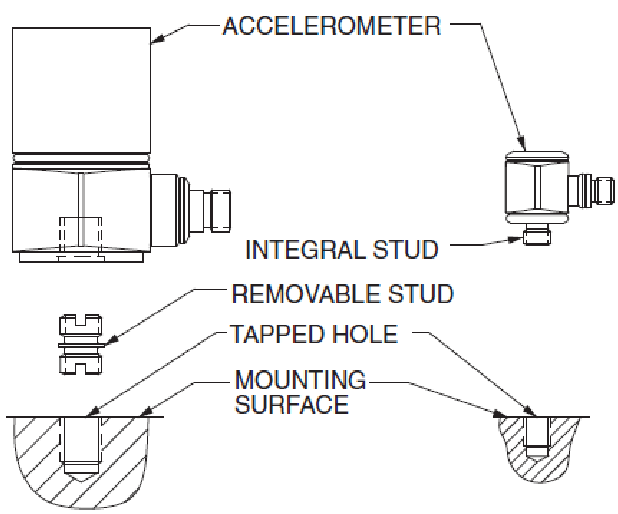


Figure 79 Possible stud configuration for accelerometers

3.4 Basic concepts of signal analysis

3.4.1 Anatomy of a signal

A periodic signal has different features which are convenient to introduce at this point. These features are not easily visible in vibration signal collected within condition monitoring, mainly because is composed a variety of signals with different frequency. For this reason it may not helpful to explain these features by showing a real signal. However a sinusoidal signal can illustrate better different features associated with the signal as it shown in the Figure 80.The figure shows a sinusoidal signal during an interval of time. The amplitude of the signal refers to magnitude of the vibration which because its periodicity, it will change over time. The maximum absolute value of amplitude detected is called **peak**,. The difference between the maximum and minimum is called **peak to peak** value.

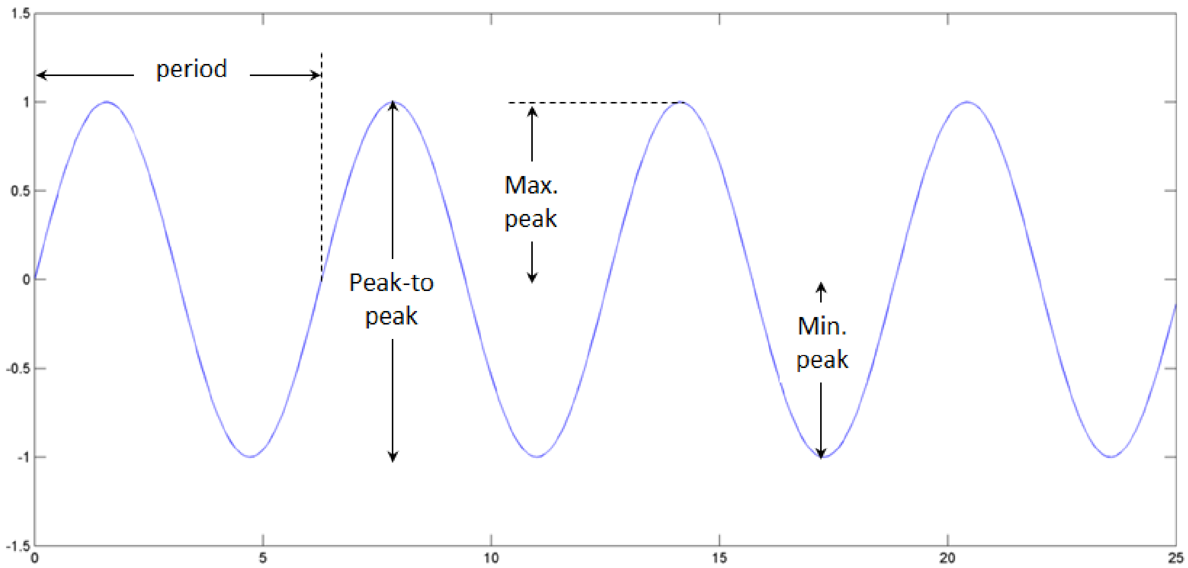


Figure 80 Example of periodic signals. Sinusoidal signal

3.4.2 Filtering

Filters are signal processing tools used to limit the frequency content of the signal. By using filters, only signals within the frequency range in interested are including on the further processing.

Filters are often classified depending of the relative location of the pass band in the frequency spectrum. Pass band, is the frequency range the filter will let the signal pass through. Based in this classification the filters are: Low pass (LP). High pass (HP), Band pass (BP) and Band stop(BS) as it shown in Figure 81.

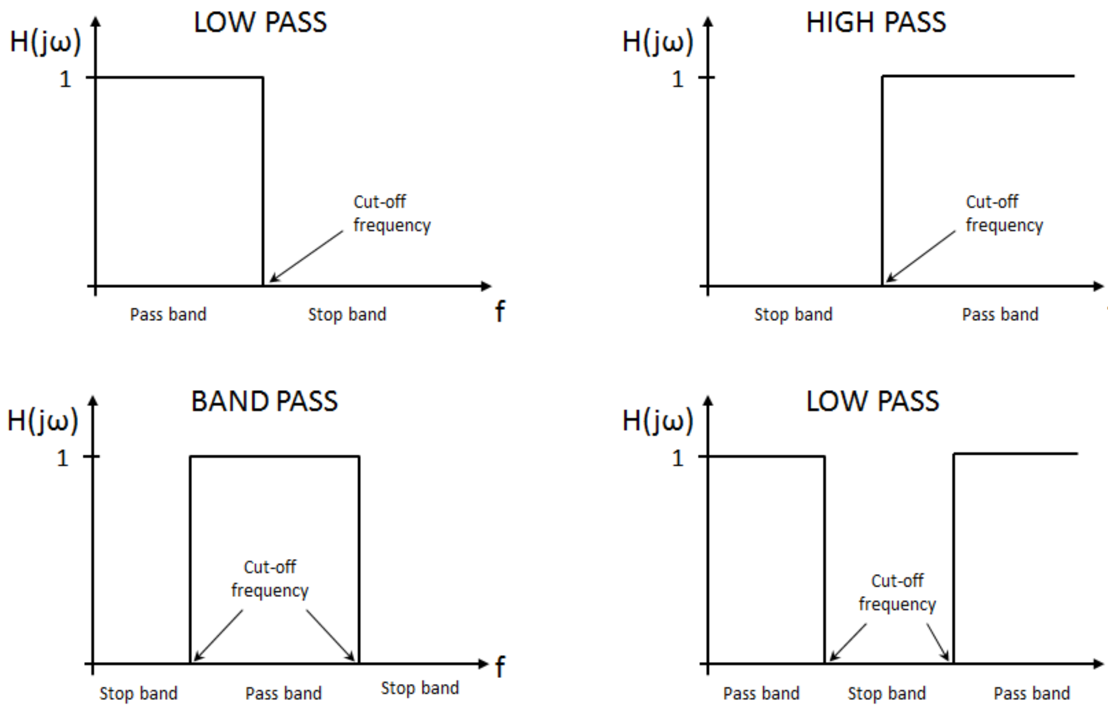


Figure 81 Magnitude of the frequency response for ideal filters (Bóden, et al., 2014, p. 145)

The above representation of filters corresponds to ideal filters. But in reality filters may let some part of the signal in the stop band pass through, given by the **attenuation [dB]** of the filter. Besides a real filter will have a **transition region** between the pass band and the stop band. This transition region will be characterized by the **slope** in dB/octave. Finally in the pass band region filters will be characterized by the **ripple**[dB]. See Figure 82

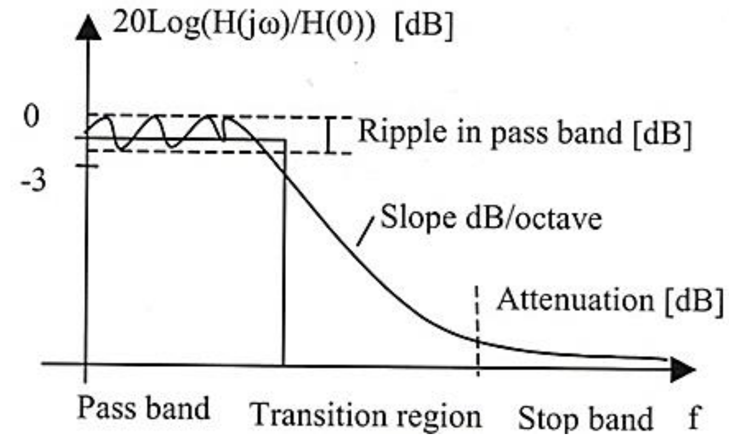


Figure 82 Magnitude of the frequency response function for a realistic low pass filter (Bóden, et al., 2014, p. 146)

There are analog and digital filters. Digital filters are filters used after the signal have been acquired and transform from analog to digital information.

Some advantage of digital filters point out by Boden et al. includes:

- Digital filter are insensitive to environmental factors
- Some mathematical parameter of the filter can be easily changed to increase accuracy
- They are reproducible

In the other hand, analog filters are used before discretizing the signal. They are mainly used to avoid acquiring part of the signal with frequency content higher than interested. By this way aliasing is avoided. For this reason these analog filters are often referred as antialiasing filters.

3.4.2 Aliasing and sampling frequency

When vibration data is collected in form of points (amplitude, time) it is important to know that this is just a representation of the “real” signal. This is because the analogue signal is sampled at a certain equal intervals of time. Thus using a certain sampling frequencies see Figure 83.

The higher the sampling frequency, a more close copy of the real signal will be obtained. However this is often limited by capacity in the instruments used to acquire data. On the contrary, if the sampling frequency is too low, it may not represent the signal sufficient with accurately as it is illustrated in Figure 83 (b). In fact it can result in a false representation of the signal with totally different period and characteristics. This problem is often referred as aliasing.

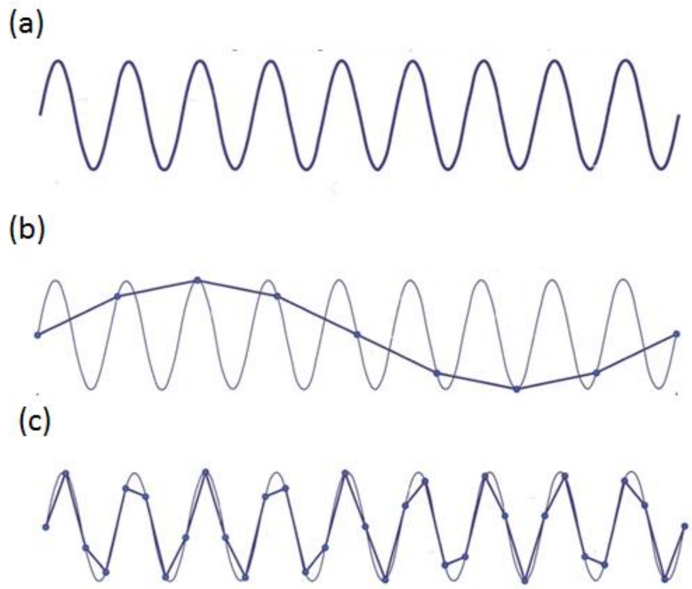


Figure 83(a) real analogue signal. (b) real signal sampled with too low sampled frequency. (c) singal sampled with Nyqvist frequency

Hopefully aliasing was observed and studied by the Swedish-American Harry Nyqvist who determinated empirically in the 20's, that the sampling frequency should be at least twice than the highest frequency aimed to study. Later Claude E. Shannon proved mathematically Nyqvist's finding. For this reason, the minimal reliable sampling frequency is often referred as Nyqvist-Shannon sampling theorem or simply Nyqvist frequency (Bóden, et al., 2014, p. 41). However it is common practice to use as sampling frequency 2.56 times the highs frequency aimed to measure. Most of the modern acquisition systems guide the user to decide an appropriate sampling frequency based on these principles.

To exemplify the use Nyqvist-Shannon sampling theorem, consider this case: if there the objective it is study bearing damage, it is know that they are often associated with vibration with high frequency signals. In a early stage these signals appear in a range of 500 Hz-2kHz. Therefore the sampling frequency should be at least 4 kHz. This is equivalent to 4000 samples per second. If data storage is not a limitation, oversampling is also a good option for ensuring robust data acquisition.

3.4.3 Leakage problem and windowing

When sampling vibration signals, this is carried out during a certain period of time, referred as a sampling period. The sampling period usually may not be a multiple of the period of the containing signals and therefore the signal can be become truncated when is sampled. See Figure 84. This will affect the resolution on the frequency spectrum, when using FFT algorithm, resulting in a broader peaks than they actually is. This issue is referred as leakage, because somehow the amplitude of the peak, leakage to closer frequencies.

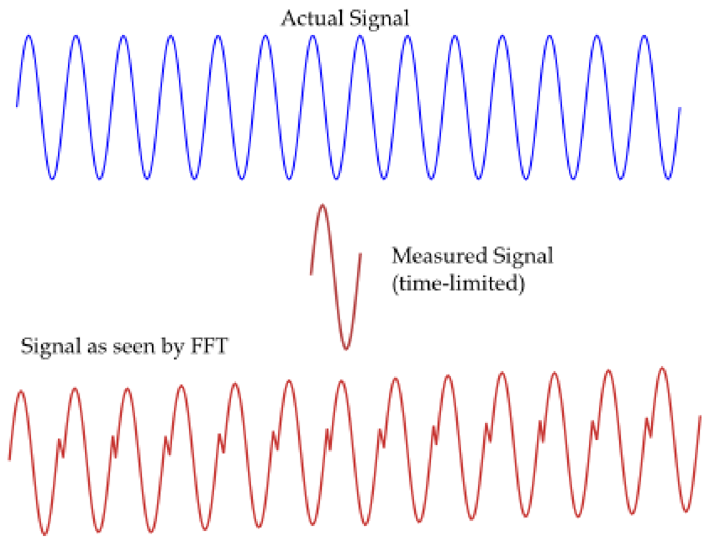


Figure 84 Cause of leakage when sampling a signal

In order to minimize the leakage effect, a technique used windowing is used. This technique consists in multiplying the signal sample by a window function of the same length. By this way, the sampled signal is force to adopt zero value at its beginning and its end. This will result in that the signal will seems periodic without discontinuities for the FFT algorithm.

There are different types of window function as: rectangular, flat top, Hanning, Hamming, Kaiser Besserl, Blackman, Barlett

3.5 Signal processing/analysis techniques

There are many techniques for processing and analyzing the data obtained. These can be combined depending on the parameter of interest. Most of the techniques described here are used when assuming the signal is sampled in a steady-estate, while others are used transient signals as in the case of free vibration produced due to impact.

3.5.1 Fast Fourier Transform

Fourier transform is a mathematical algorithm to convert a periodic signal from the time domain (amplitude v/s time) to frequency domain (amplitude v/s frequency) as it illustrated in Figure 85. This is possible thanks to Fourier Analysis theory developed at the beginning of XIX century by the French mathematician Joseph Jean-Baptiste Fourier. He developed this method to approximate any periodic signal by a sum of sin and cosines.

The theory behind Fourier analysis is expressed at a high mathematical level and therefore may be difficult to interpret. However an intuitive way for understanding it is shown in Figure 85. The sampled signal is composed of several single signals which have different frequencies. By looking at the time domain (B), the sum of this signal is observed. However when looking the signal from the frequency domain, for every frequency component correspond a specific amplitude (C).

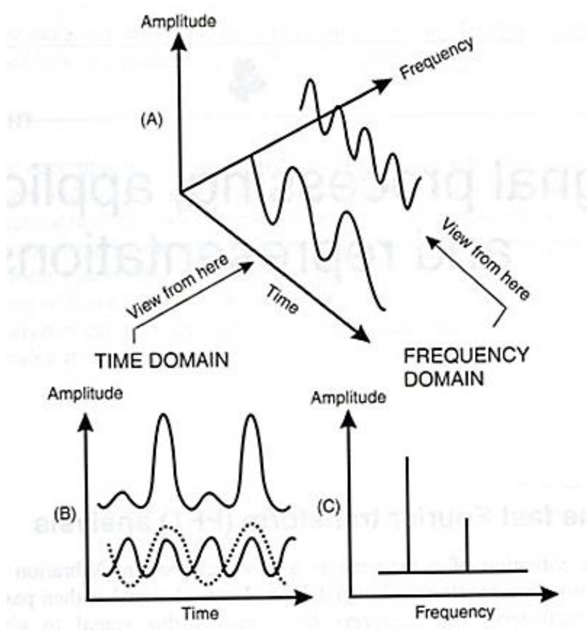


Figure 85 Interpretation of the Fourier Transform of a signal (Scheffer & Girdhar, 2004, p. 56)

The Fourier transform was firstly considered for continuous functions, but is equally applicable for sampled function when sampling vibration signals. For the latter case the Discrete Fourier Transform is used, referred simply as DFT.

It is common to refer to DFT as it were FFT (Fast Fourier Transform). However they are not exactly the same. FFT refers to an algorithm to calculate DFT. This algorithm utilized some properties of the Fourier Transform to decrease the number of calculation necessary to obtain DFT. By this way calculation can be carried out easily with modern computers.
The FFT may be the most used technique for condition monitoring because it rapidly show the peaks for vast range of frequencies.

3.5.3 Envelope

Envelope techniques are aimed at separating two high frequency signals from each other. By this way, the problem in the machine can be better diagnosed. For example, ehen defects on bearing (or gears) appear on the vibration spectrum, they are often undistinguishable and can be misinterpreted by using FFT. When this is the case, this signal is referred as modulated signal.

Consider the case of a defective bearing that has a cavity in the outer race. Every time a ball passes the cavity it will generate an impact, which will be recorded in the vibration signature. However the impact will also generate a high frequency vibration because every time the ball hits the defective race provoke the bearing to resonate. These two signals (the impact and the high frequency vibration produce by the bearing resonance) will result in a vibration signal similar to the one show in Figure 86.

By demodulating the signal, means to extract the envelop (modulation signal) from the total signal as illustrated in the bottom of Figure 86. This is done by identifying the absolute peaks of the signal absolute value. This is achieved by the use of algorithms as the Hilbert transform (Bóden, et al., 2014).

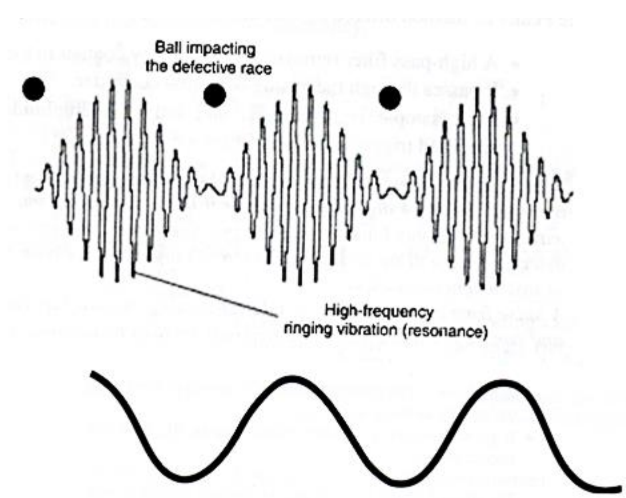


Figure 86 Vibration-balls over the a spall defect and its envelope (Scheffer & Girdhar, 2004)

3.5.3 Cepstrum

It is used to measure the existence of harmonic components in a signal. One definition of Cepstrum given in (Bóden, et al., 2014)

$$g_{cepst}(\tau) = \frac{1}{\sqrt{2\pi}} \int_{-\infty}^{\infty} \log|G(\omega)| e^{i\omega t} d\omega$$

Where G is the power spectrum of the signal.

The application of Cepstrum can be illustrated in Figure 87, which shows the spectrum of a grinding machine running at 1968 rpm or (32.8 Hz). Because of the power is transmitted to axle by a gear with a transmission ratio 31:15, harmonics of 67.8 Hz are expected due to gear meshing. This harmonics are significantly more clear as peaks in the Centrum spectrum (right) than in the frequency spectrum (left).

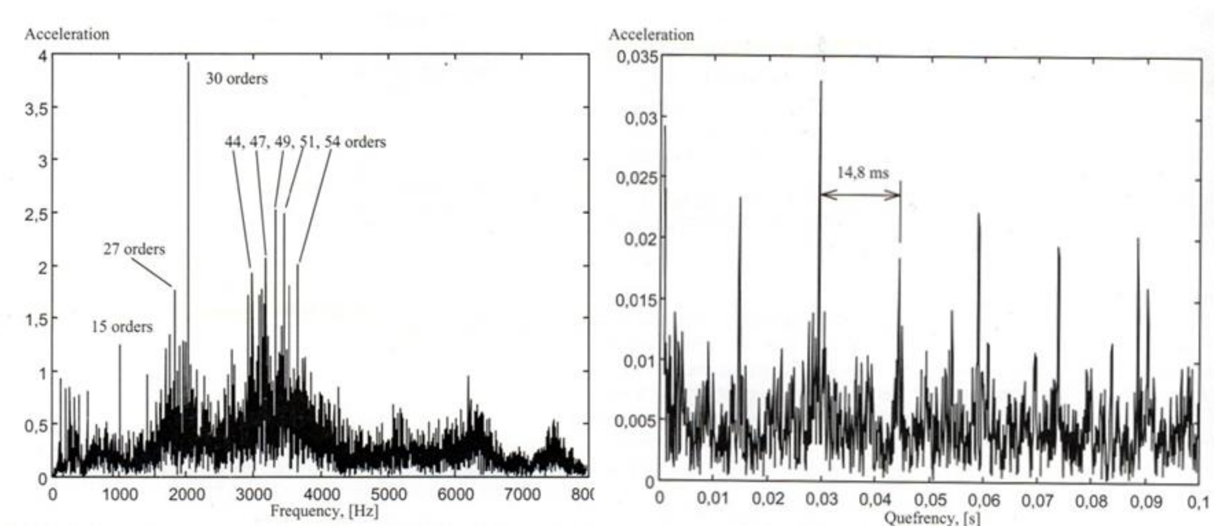


Figure 87 Vibration signature of a grinding machine using FFT and Cepstrum (Bóden, et al., 2014)

3.6 Indicators for machine condition

Once the signal has been filtered or processed with one or several of the techniques explained earlier, the next step when carrying out condition monitoring is to choose and indicator that will reflect the condition of the spindle. Here are described the most common indicator used for rotatory machinery.

3.6.1 Rms

The root mean squared or rms. is a measure of the energy content of the signal. The amplitudes are averaged in a way only the magnitude of positive and negative points are considered. This is done by squaring every point. This indicator is often used once the signal has been filtered to the frequency of interest.

$$R.M.S = \sqrt{\frac{1}{N}\sum_{n=0}^{N-1}(x_n - \bar{x})^2}$$

3.6.2 Peak to peak

This indicator is represent the maximum amplitude of a signal during the time is being measured. So the highest the peak, the condition of the spindle will be more severe.

$$Peak\ to\ peak = |Max(x_n) - Min(x_n)|$$

3.6.5 Crest factor

Crest factor is a combination of peak-to-peak and rms. It is intended to give a sense of how extreme the peaks are in the signal. It is the quotient between the peak and he rems value.

One of the definitions of crest factor is :

$$C_F = \frac{peak - to\ peak}{2R.M.S}$$

Boden warns in (Bóden, et al., 2014) that the definition of crest factor is not unified and can differ from other authors. In that sense, one should be careful in specifying the definition when referring to this indicator.

3.6.3 Kurtosis

Kurtosis is often used in statistics for describing the shape of the distribution of a certain population. In a similar manner is used within maintenance to describe the shape of the signal. Certain values of kurtosis are associated with bearing damage. The kurtosis for a signal s is defines as:

$$K = \frac{\frac{1}{N}\sum_{n=1}^{N}(x_n - \bar{x})^4}{\left(\frac{1}{N}\sum_{n=1}^{N}(x_n - \bar{x})^2\right)^2}$$

3.6.4 Bearing frequency equations

Bearing frequency equations are used to identify where the problems on the bearings are. Once these frequencies are calculated, they may appear in frequency spectrum of the measured signal, when bearing damage has occurred. By looking at the amplitude of these frequencies, peaks may indicate damage in the components. In order to obtain these equations, geometrical parameters of the bearing are necessary. It is important to notice that some these geometrical parameters are not available in bearing catalogs.

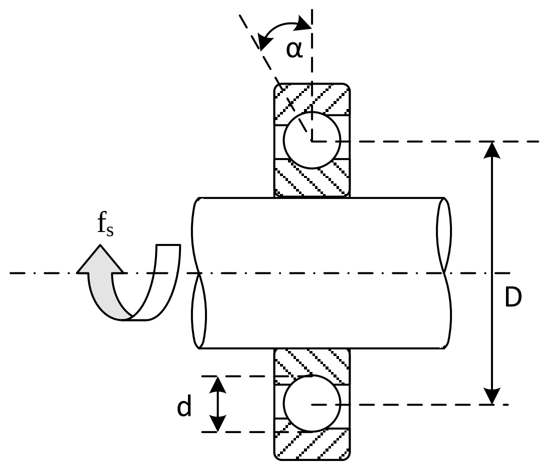


Figure 88 Geometrical parameters for calculating bearing frequency equations

Where:

f_s	: Shaft speed in [rev/s]
d	: Bearing ball diameter [m]
D	: Pitch diameter of the bearing [m]
α	: Ball contact angle
N	: Number of rolling elements

Cage rotational frequency or also knows as fundamental train frequency (FTF)

$$FTF = \frac{f_s}{2}\left(1 - \frac{d}{D}\cos\alpha\right) \text{ (1)}$$

Ball spin frequency (BSF)

$$BSF = \frac{Df_s}{2D} \text{ (2)}$$

Rolling element pass frequency with respect to the outer race (BPFO)

$$BPFO = N\frac{f_s}{2}\left(1 - \frac{d}{D}\cos\alpha\right) = N \times FTF \text{ (3)}$$

Rolling elements pass frequency with respect to the inner race (BPFI)

$$BPFI = \frac{f_s}{2}\left(1 + \frac{d}{D}\cos\alpha\right) \text{ (4)}$$

Notice that these frequencies will depend on the rotational speed of the shaft. In the case of fundamental train frequency FTF this will be below shaft rotational frequency, while for BPFI, BPFO and BSF above.

Based on literature research conducted, some limitations have to be taken into consideration when using bearing frequency equations:

- Bearing geometrical dimension must be available. Some large bearings manufacturer provides these frequencies as a factor of the speed, while others provide some general geometrical dimensions of the bearing which may not be sufficient to calculate these frequencies. One approach to overcome this problem is to indirectly calculate the necessary dimensions based on CAD models of the bearing provided by manufactures or by other means.
- This is only for angular contact ball bearings, for other bearings, the formulas may vary.
- This formula do no perform well at very high spindle speed, because other effects occur on the spindle (centrifugal forces causes rolling and slipping) (Smith, 2014)
- These formulas do no perform well when the damage on the bearings is considerable (Castelbajac, 2016). This is because the damage on rolling elements can be so extended that is will produce noisy signals spread in the frequency spectrum.

4.EXPERIMENTAL SETUP

4.2 Equipment

The equipment used in the experimental part of this work is described below. It consists basically in three machine tool spindles, accelerometers used to collect vibration and auxiliary equipment for measuring vibration.

4.2.1 Machine tool spindles

Vibration measurements were carried out in spindle of tree milling centers, located in the laboratory of production engineering of KTH. These milling centers are mainly used for research, and therefore have operated for a minimal amount of hours compared with a milling center used in industry. These milling center are designated with the letters A, B anc C, as Table 2 displays. Important to highlight is that B corresponds to a recently purchased machine tool (2015) while A was equipped with a new spindle during 2014.The basic information of these three milling centers is shown in **¡Error! No se encuentra el origen de la referencia.**.

Table 2 Basic information of the three spindles studied

Designation	A	B	C
Number of machine axis	5	3	3
Spindle			
-Transmission	Integral	Belt driven	Gear driven
-Max. Speed	12 000 rpm	8 000 rpm	7500 rpm
-Torque S1/S6	25/ Nm		
-Power S1/S6	31/52 kW	11 kW	22 kW
-Tool interface	HSK A-100	ISO/BT 40	ISO/BT 40

Interesting to notice is the fact that these three milling centers have different type driving system for their spindles.

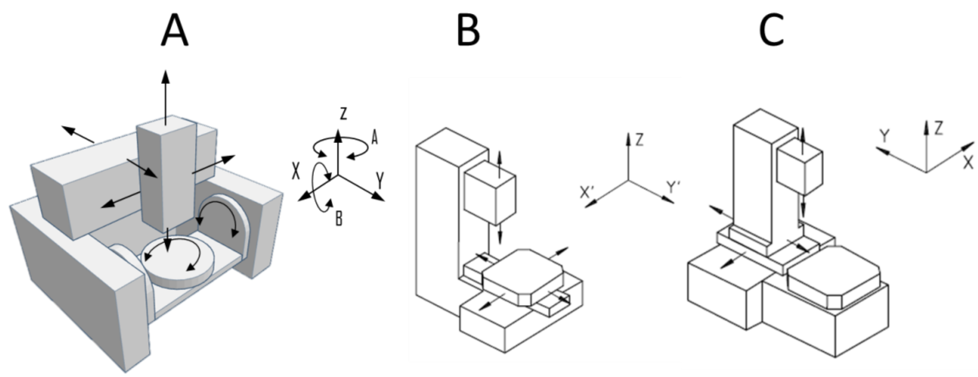


Figure 89 Configuration of the milling centers investigated.

4.2.2 Vibration transducers

Several vibration transducer models were used. Two of them were accelerometers piezoelectric-based, while the third model was a capacitive-based displacement transducer. Their characteristics are shown in the table below.

	ICP 603C01	3225F	C8-32.20
Manufacturer	IMI Sensors	Dytran	Lion Precision
Measurement unit	Acceleration (uniaxial)	Acceleration (uniaxial)	Displacement
Principle	Piezoelectric (shear)	Piezoelectric	Capacitive
Amplitude range	±50 g	500 g	50µm
Sensitivity	---	10 mV/g	---
Frequency range	0.5 Hz to 10kHz	1.6 Hz to 10 kHz	---
Natural Frequency	24 kHz	---	---
Resolution	350 µg	---	---
Mounting method	Magnetic (dual rail)	Wax mount	---
Comments	Used for setups 1 and 2	Used Setups 3,4 and 5	Used for Setup 4 and 5

Table 3Vibration transducers used in the different setups. Some information was not available

The capacitive transducers used (C8-32.20) were mounted in Contactless Excitation Response System CERS. CERS is a device developed at KTH for research purposes in machine tools. This device is mounted in the machine tool table. Once is mounted CERS is capable of applying a radial force in custom tool (referred as dummy tool) mounted on the tool holder. This force is generated by an induced magnetic field powered externally. Besides, the displacement of the dummy tool can be accurately sensed by two capacitive probes located perpendicular to each other in the CERC unit as Figure 90 shows.

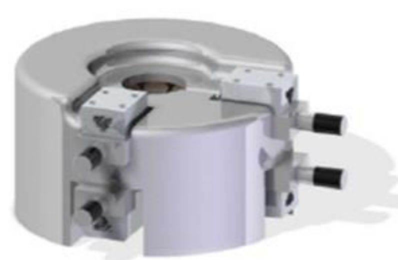


Figure 90 CERS unit

4.2.3 Data Acquisition equipment

Two data acquisition equipment were used. The first was SEMA 600 (SEMA-TEC) which is a portable collector and analyzer of vibration measurement used in industry. This equipment is commonly used for maintenance proposes because gives a friendly user's interface. This collector/analyzer have a custom software named SEMALYST where allow to obtain rapidly key parameters when measuring vibration. It also has 6 channels in which four of them correspond to ICP cards for connecting accelerometers. ICP cards operate in a frequency range of 0.1 Hz to 25 kHz. Besides the cards allow a sampling frequency of up to 500 kS/s. this sampling frequency which is reduced to 65kS/s when the 6 channels collect in parallel.

The second acquisition equipment used was SCADAS mobile unit SCM01 provided by LMS. In this case it corresponds only to a recorder unit. Therefore the data processing has to be carried out by an external Computer. SCM01 is used for laboratory but also industrial application because its compact dimension and low weight (2.5 kg). In this work this equipment was used for obtaining data from CERS (setup 4 and 5).

Figure 91 Data acquisition system used. SEMA600 by SEMATEC and SCM01by LMS

4.3 Experimental setups

Five different setup where prepared during this work. Each of which had a different focus. Setup 1, Setup 2 and Setup 3 were aimed at comprehend better how different variables must be taken into account when evaluation spindle condition based on vibration measurements. Setup 4 and 5 was aimed at evaluating CERS as a possible tool for assessing bearing condition in machine tool spindles. This, by measuring vibration displacement on the tool and analyzing the frequency spectrum of the signals collected.

4.3.1 Setup1: Speed sweep and evaluation of damage parameters

In this setup vibration was measured at two perpendicular radial positions (x and y) on the front spindle housing for the three spindles early mentioned (for milling centers A,B and C) as it is illustrated in Figure 92.

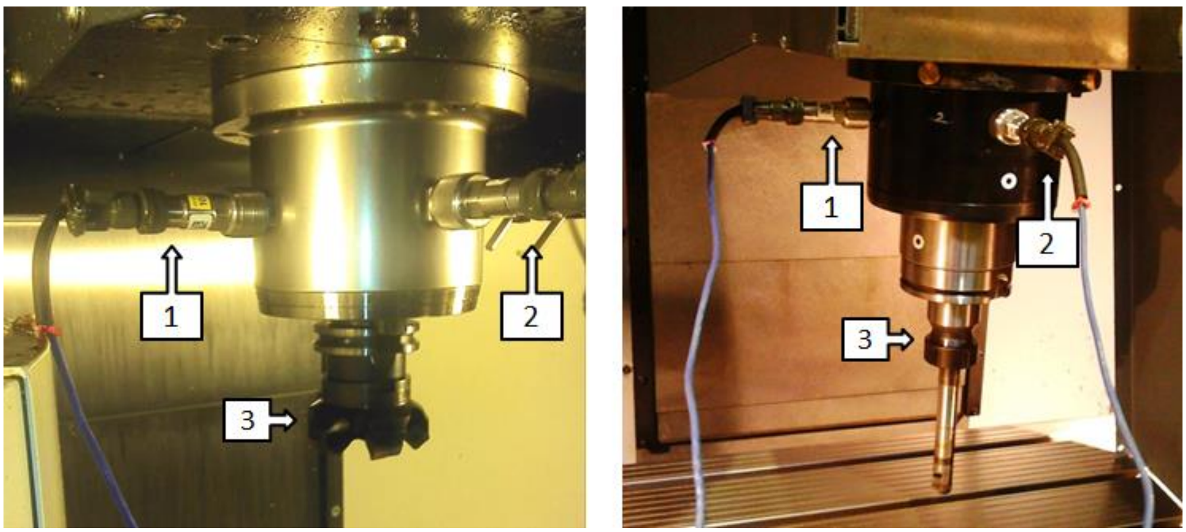


Figure 92 Vibration measurement for Setup1 in C and B. (1) ICP 603C01 accelerometer in X axis (2) ICP 603C01 accelerometer in Y axis. (3) Milling tool

This was carried out on fixed speed steps of 1000 rpm until reaching maximum operating speed of the spindle. The sampling time was four seconds and the sampling frequency of 65.6 kHz. The measurements were carried out with and without a milling tool mounted. The milling tool was different for every spindle. In the case of A, the control panel of the machine tool prevented to rotate the spindle without tool. For this reason it was not possible to take vibration measurements of the spindle without tool in A’s spindle.

The intention of this setup was to observe how different damage severity parameters vary with speed. A second intention with this setup was to acquire sense of how vibration levels differ for spindles with different driving systems.

4.3.2 Setup 2: Effect of acceleration mounting position on vibration level

In order to study the influence of the accelerometer position in vibration measurements carried out in spindle housing, the machine B's spindle was chosen. This is because the front part of the spindle housing was accessible along its entire housing perimeter. This enabled the mounting of the accelerometer (ICP 603 C1) in different locations along the housing contour. The position of the accelerometer was varied in 15° along the front spindle bearing housing. This resulted in 24 positions illustrated in Figure 93. The spindle was then run without tool in two different speed 400 rpm and maximum speed 8000 rpm.

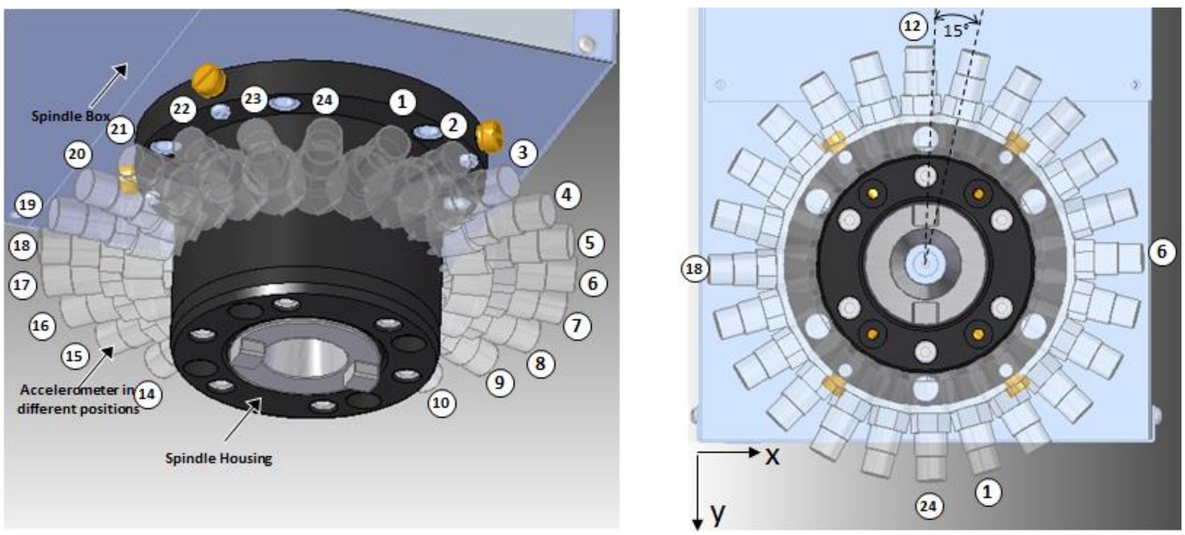


Figure 93 Setup 2 varying the position of the accelerometer radially along the spindle housing of machine B

Two measurements were taken randomly for every position of the accelerometer with a sampling time of four seconds and sampling frequency of 65.6 kHz. Then, for every measurement, the rms. for the frequency band 10-2000 Hz and 2-20 kHz was obtained using SEMA600's software SEMALYST. The two values for every position were then averaged.

In typical vibration measurements the vibration is measured along X and Y axis of the machine workspace. The goal with the setup was to study was the relation between accelerometer position and vibration level.

4.3.3 Setup 3: Impact test on the spindle

In setup 3 an impact test was carried out in machine A because its configuration: In this machine the spindle can move along three different axis X, Y and Z. This was interesting in the sense of trying to study a relation between the spindle position in the workspace and the natural frequencies observed when measuring vibration on the tool holder and housing. The measurements were carried out in 12 different spindle positions within the machine tool's workspace.

The procedure consisted in impacting the tool holder with a hammer. This is to generate free vibration in the spindle structure. A minimum of five impacts were necessary for every measurement. The force generated by the hammer is registered and also the response (in form of acceleration by three accelerometers Dytran F3225F: one on the tool holder y direction (same direction the impact was directed). As well as two on the spindle housing (x and y).

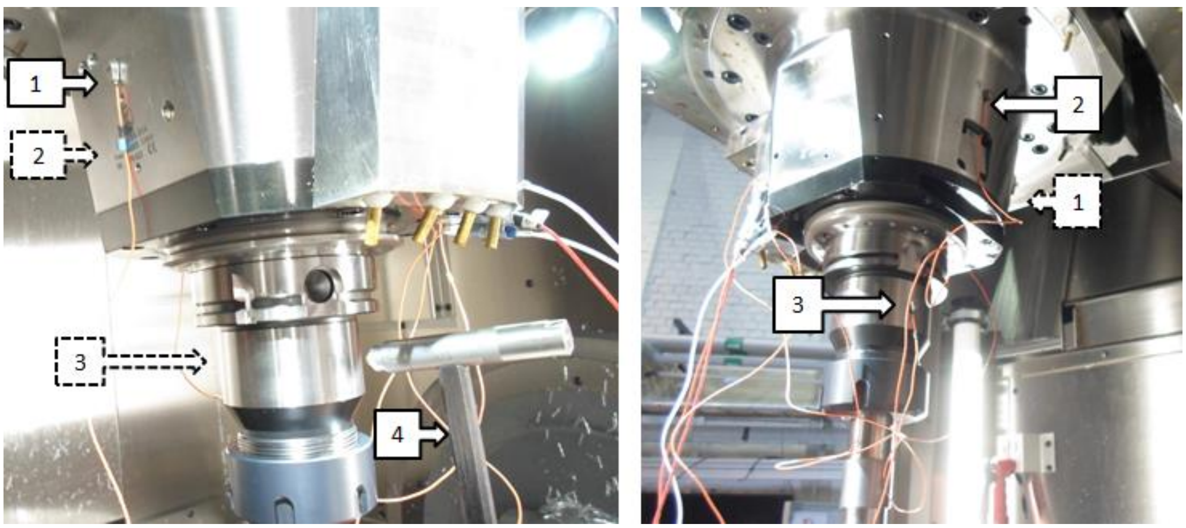


Figure 94 Setup 3 (1) Accelerometer in spindle housing x axis. (2) Accelerometer in spindle housing y axis. (3) Accelerometer in tool holder y axis. (4) Impact hammer.

4.3.4 Setup 4: Tool vibration measured by CERS (not forced applied)

As mentioned before, CERS is a device used for research proposes at KTH's department of industrial production. It consists in a cylindrical aluminum housing that contains a "magnetic bearing". The CERS unit is mounted on the machine tool table (3) as Figure 95 illustrates. This unit can hold two capacitive probes mounted perpendicular to each other (4) in the upper side of the CERS unit. Besides, CERS has an excitation unit similar to a magnetic bearing. This excitation unit is capable of generating constant or variable axial force on the dummy tool (2). This is done by generating a magnetic field powered externally.

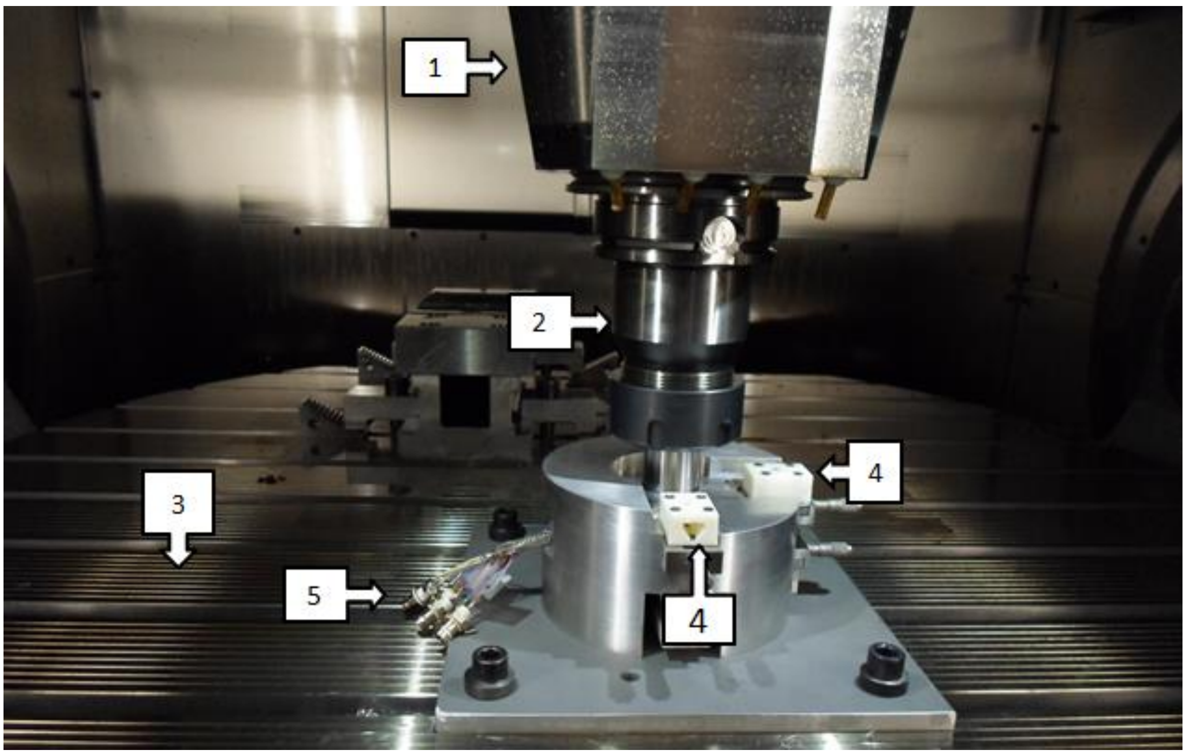


Figure 95 CERS unit mounted in machine tool A. (1) Spindle housing. (2) Dummy tool (3) Machine tool table. (4) Capacitive probe housing (5) Connectors for excitation unit.

The purpose of setup 4 and 5 is to study if CERS has the potential of detecting defects on spindle bearings and thus assessing spindle condition. Assuming that vibration generated by defects on the bearings transmits through the spindle, it would be reasonable to think that these high frequency signals can be detected on the tool. As mentioned earlier, high frequency signals have the characteristic of having very small amplitude, which make difficult to measure them. However capacitive sensors can reach a resolution of nanometers with proper calibration and drive system for the probes. The frequency spectrum collected will be later compared with the frequencies associated with bearing damage. Even if the bearing is in good condition, it may be possible to observe bearing frequencies as in (Vafaei, et al., 2002).

In setup 4, vibration were measured in four position (two on the housing and two in the tool tip), for two different speeds (4000 and 12000 rpm). Three measurements were carried out for every speed with a sampling frequency of 25.6 kHz for accelerometers and capacitive probes. Figure 96 shows the disposition of the accelerometers and capacitive probes used for collecting vibration signal from spindle on the housing and on the tool.

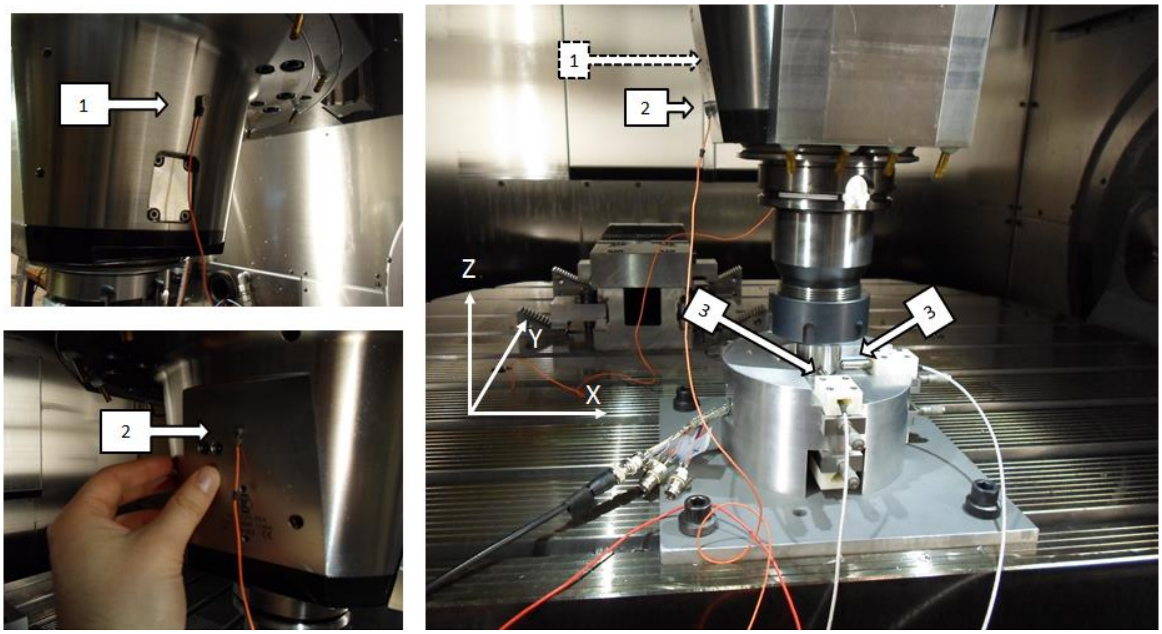


Figure 96 Setup 4 and 5.(1) Accelerometer in spindle housing Y axis. (2) Accelerometer on the spindle housing X axis. (3) Capacitive probes mounted on CERS unit.

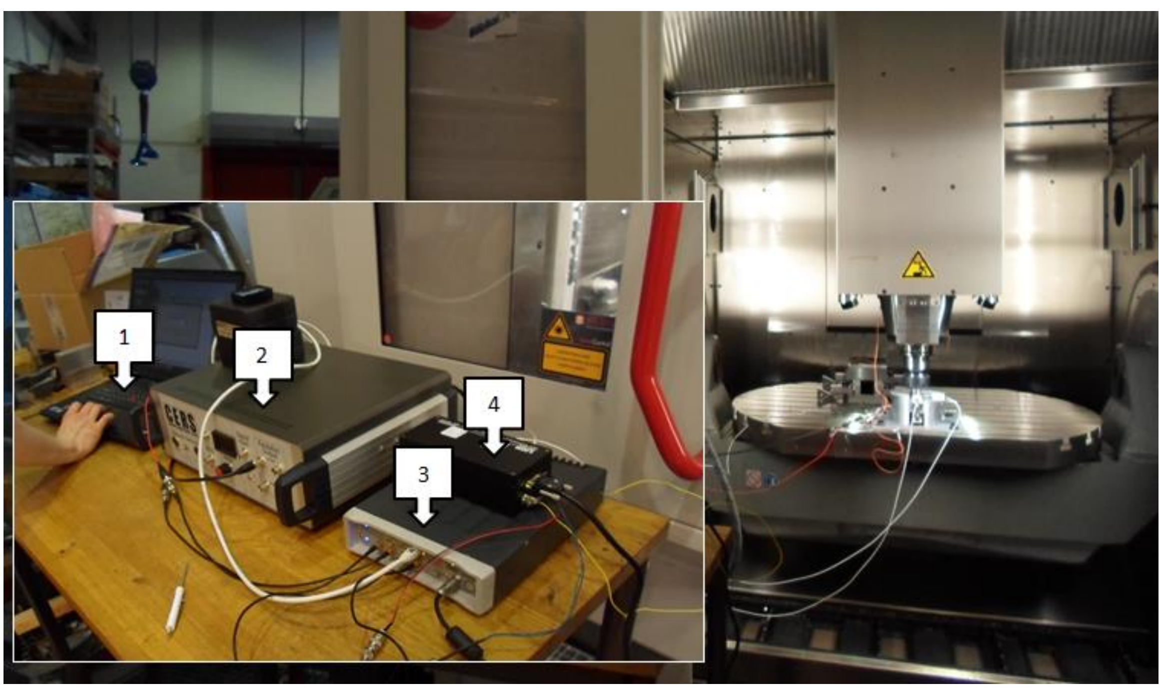


Figure 97 Setup4 and 5. (1) Laptop for data collection. (2)CERS' external power unit (3) SCM01 (4) Lion Precision Driver for the capacitive probes

4.3.5 Setup 5: Tool vibration measured by CERS (load applied)

This setup is the same as setup 4 but with the excitation unit with maximum current. It has been estimated in previous experiments with CERS that is equivalent to a radial force of approximately 10 N. This force was set to it generated as constant load in X direction. Even though the value of the force applied is not exact, the most relevant to study is the influence of the force on the frequency of interest in the vibration spectrum.

5.RESULTS AND ANAYSIS

6.1 Setup 1

Once the speeds-vibration data points were plotted, a smoothing curve was generated to connect them. This curve was created in Matlab using smooth spline algorithms (smoothing parameter 0.9998). The curve is believed to represent approximately if the data were taken using narrower speed steps.

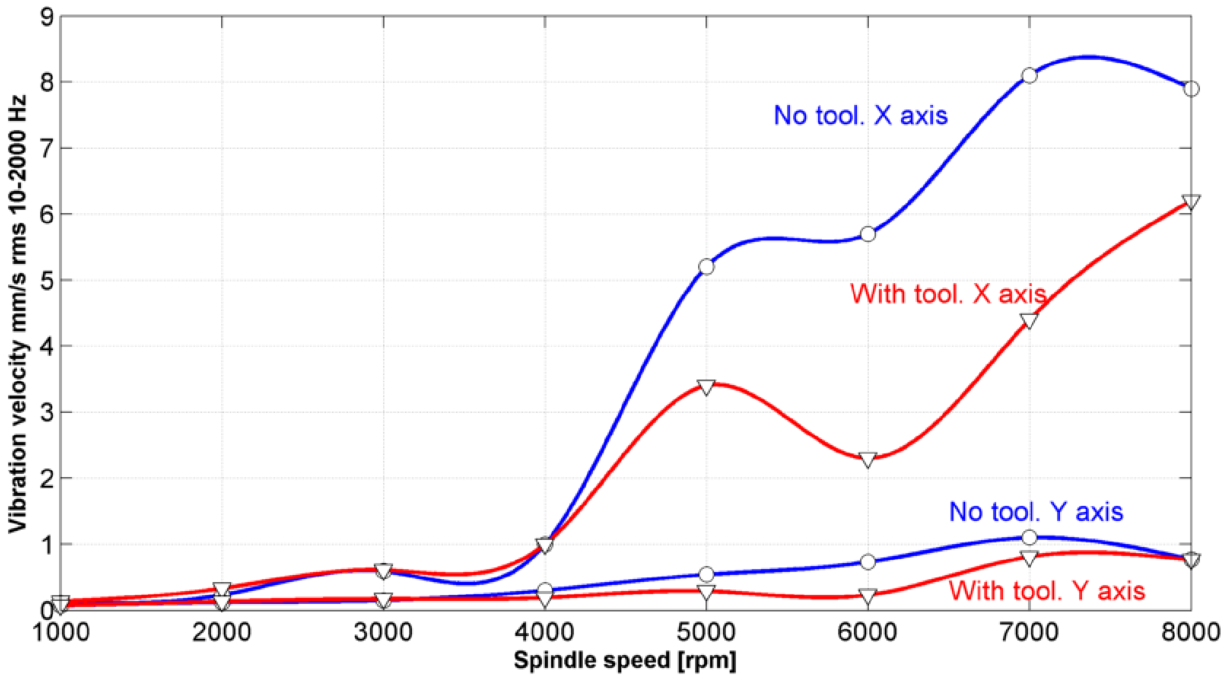


Figure 98 Vibration levels in machine tool B's spindle nose. X and Y axis. With and without clamped tool

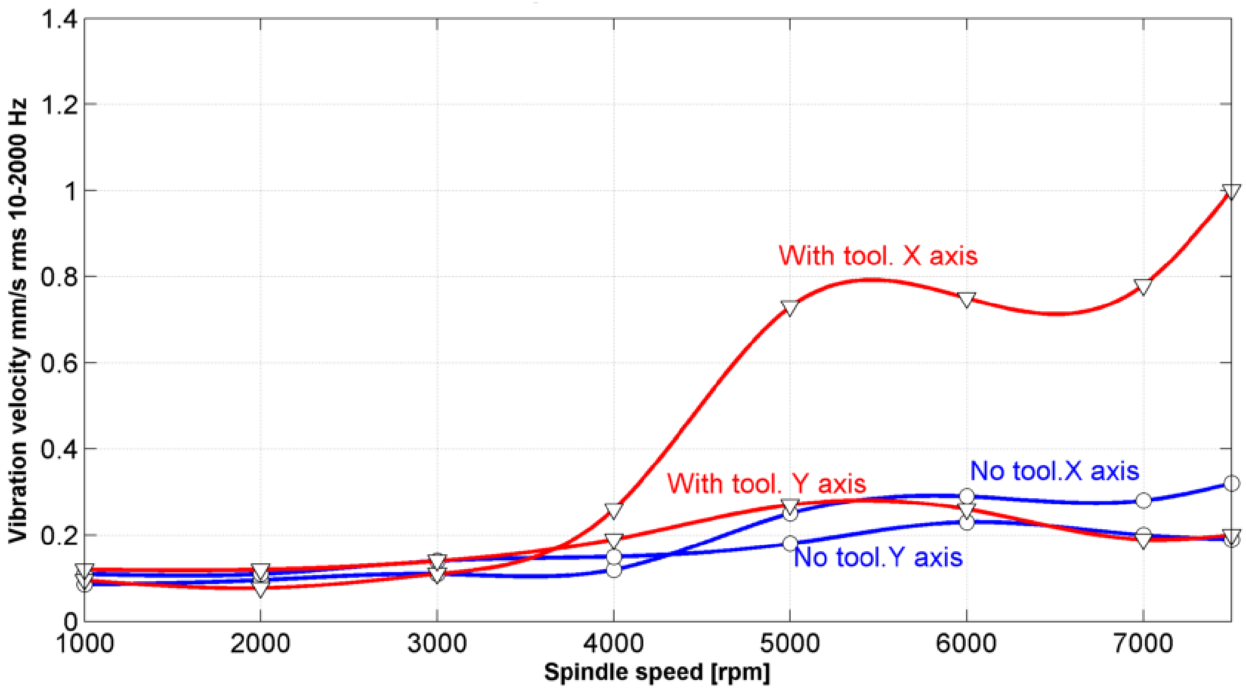


Figure 99 Vibration levels in machine tool C's spindle nose. X and Y. With and without tool.

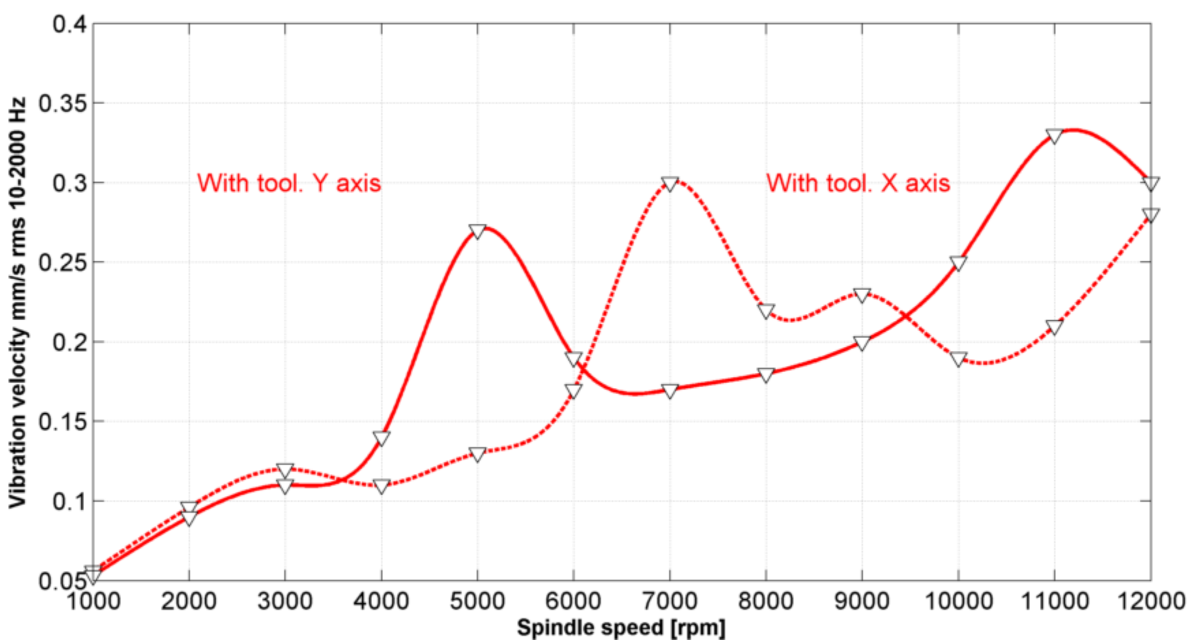


Figure 100 Vibration levels in machine tool A's spindle nose. X and Y axis. With tool clamped

The overall increase of the vibration level with the speed observed in the three spindles can be attributed to unbalance. However some local peaks are observed (as in B around 5000 rpm, C at 5500 rpm or in A at 5000,7000 and 11000 rpm. These are believed to be result from critical speeds in the spindle components. These critical speeds may be attributable to tool, spindle components or even to components of the driving system.

Overall vibration levels differ greatly among the three spindles studied. The different vibration levels were expected due to every spindle use different driving systems, which has a significant influence in the vibration level. In the case of machine tool B for instead, the maximum vibration level is above 8 mm/s rms in the X axis near 7000 rpm, when running without a tool. While (without a tool) reach just a vibration level above 0.3 mm/s near its maximum speed, this is 7500 rpm. On the other side, overall vibration levels in A are much lower compared with B and C at a similar speed.

Differences in vibration levels when tool clamped and unclamped within a specific spindle are also significant. Especially in the X axis when the highest vibration levels were found for machine tools B and C. Because the unbalance of the tools used for every spindle was unknown, the results can only be understood quantitatively in this case. It is interesting to observe that a tool is considered to be a source of imbalance, and therefore measurement with tool are expected to result in higher vibration levels measured in the housing. However in the case of machine tool B, the opposite occurred. In the X and Y directions the vibration levels were lower when clamped tool compared when non tool case. An explanation to this is that the position of imbalance mass of the tool may have been somehow opposite to the imbalance mass of the spindle, which may have result in a lower resulting imbalance (spindle with tool)., generating lower vibration levels.

In the next set of graphs four different vibration severity indicators were evaluated for the three spindles along their operating speed. These indicators are peak, peak-to-peak, crest factor and kurtosis. The equation for calculating these indicators are indicated in section 3.6. The indicators were evaluated for vibration taken in the x axis at the spindle nose. Vibration was measured for the spindles of A and B without tool clamped.

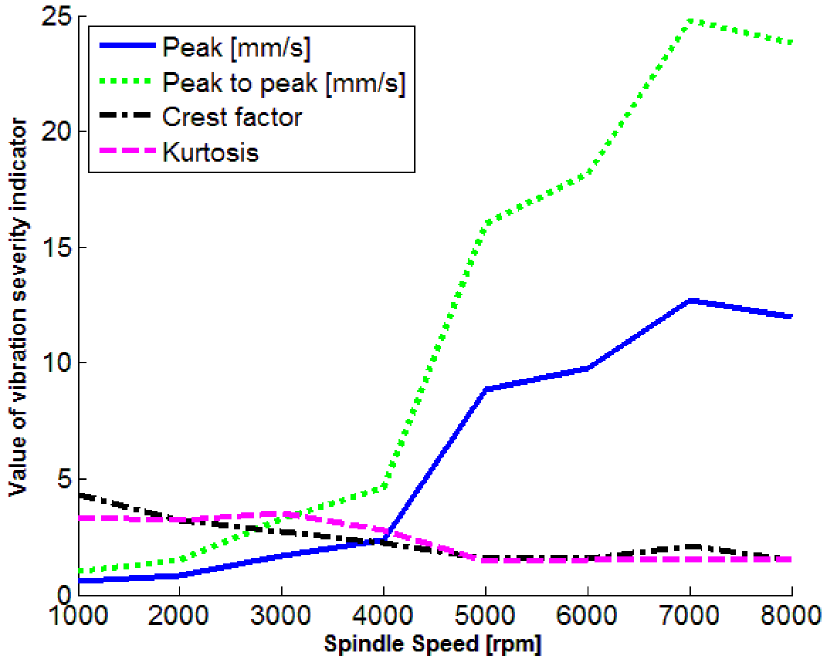


Figure 101 Evaluation of vibration indicators among spindle speed. Machne tool B

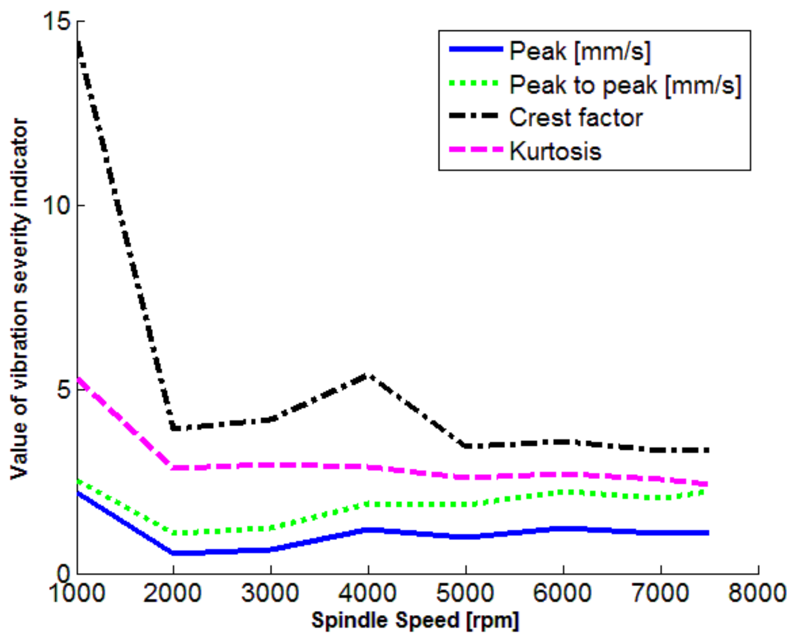


Figure 102 Evaluation of vibration indicators along spindle speed. Machine tool C.

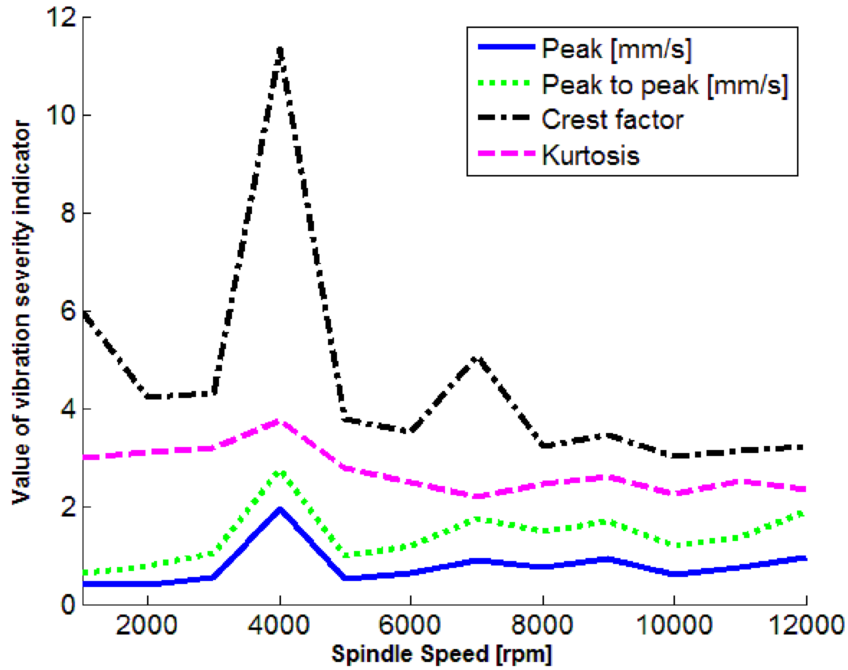


Figure 103 Evaluation of vibration indicators along spindle speed. Machine tool A

Note that for the three spindles studied, each of the vibration severity indicators varies significantly along the spindle speed. The indicators peak and peak-to-peak trends to increase with speed in a similar manner, while peak-to-peak being approximately two times higher than peak indicator. Crest factor and kurtosis on the contrary, tend to decrease in value with higher speed the spindle when they are compared with their own initial values.

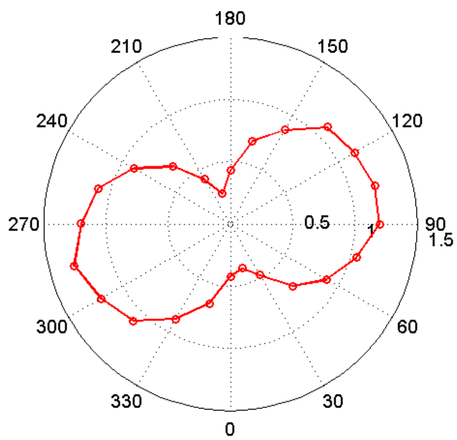
Some peaks identify along spindle speed for these indicators, not necessarily occur at critical speeds found in RMS-speed diagrams. These peaks are explained by high vibration amplitude at relatively low frequency (low energy).

Because the fluctuations of values among these severity indicators along the spindle speed, they must be used carefully when using them for evaluating spindle condition.

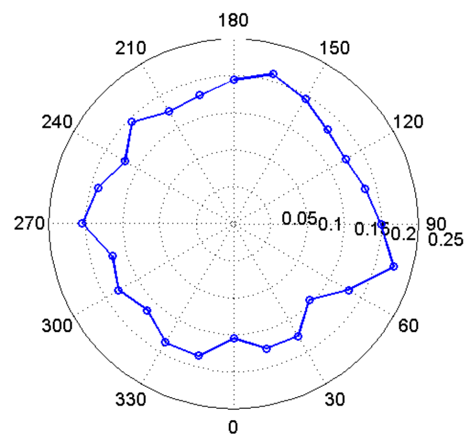
In the appendix A, two guidelines for vibration severity evaluation can be found. In both of them peak in mm/s is used. In machine tool B, this indicator reaches its maximum of 13 mm/s (0.512 in/s) at around 7000 rpm, which is considered “FAIR” at that speed for a machine tool using “Machine-tool vibration tolerance table” provided by Entek IRD. However according to the overall alarms for spindles suggested by Technical Associates of Charlotte, Inc. , this vibration level is long over “ALARM” value of 3 mm/s peak for roughing operations. In a similar manner, spindle in C, has it maximum of 2 mm/s (0.079 in/s) at 1000 rpm which is “GOOD” for roughing operating but set the ALARM for machine finishing.at the contrary is considered between “VERY SMOOTH” and “EXTREMMY SMOOTH” by Entek chart.

6.2 Setup 2

By looking at the diagrams it is interesting to observe the different vibration levels depending on the mounting position of the accelerometer in the spindle housing for a fixed speed. In the figure the angle 0° correspond to the Y axis of the spindle. It is observed that the maximum, do not necessarily occur at 90° or 270°. In fact it seems to be near 105° and 285°.

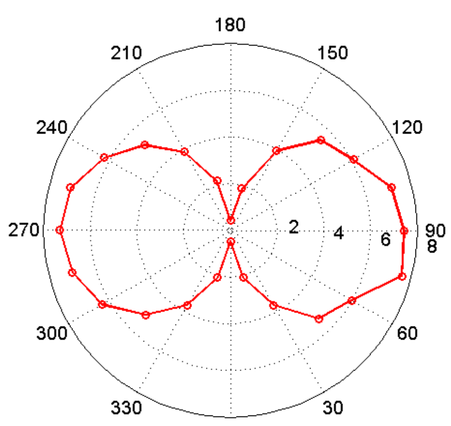


Vibration velocity[mm/s] rms 10-2000 Hz. at 4000 rpm

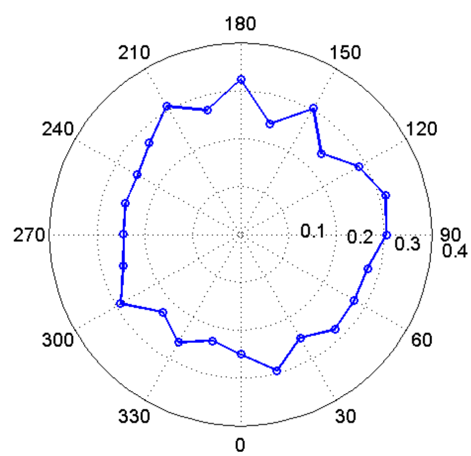


Vibration acceleration [g] rms 2-20 kHz at 4000 rpm

At the speed of 8000 rpm the differences in vibration around the spindle nose are more evident. The different between the maximum and minimum vibration level for a fixed speed of 8000 is greater than in the case of 4000 rpm. For the velocity diagram, in this case, the maximum seems to occur at 90° and 270°. This means in the X axis of the spindle with is usually preferred as one of the measuring spot when conducting condition monitoring. In the case of the acceleration vibration levels also vary but not a clear pattern is observed as in the case of vibration velocity. This variation may be of random nature.

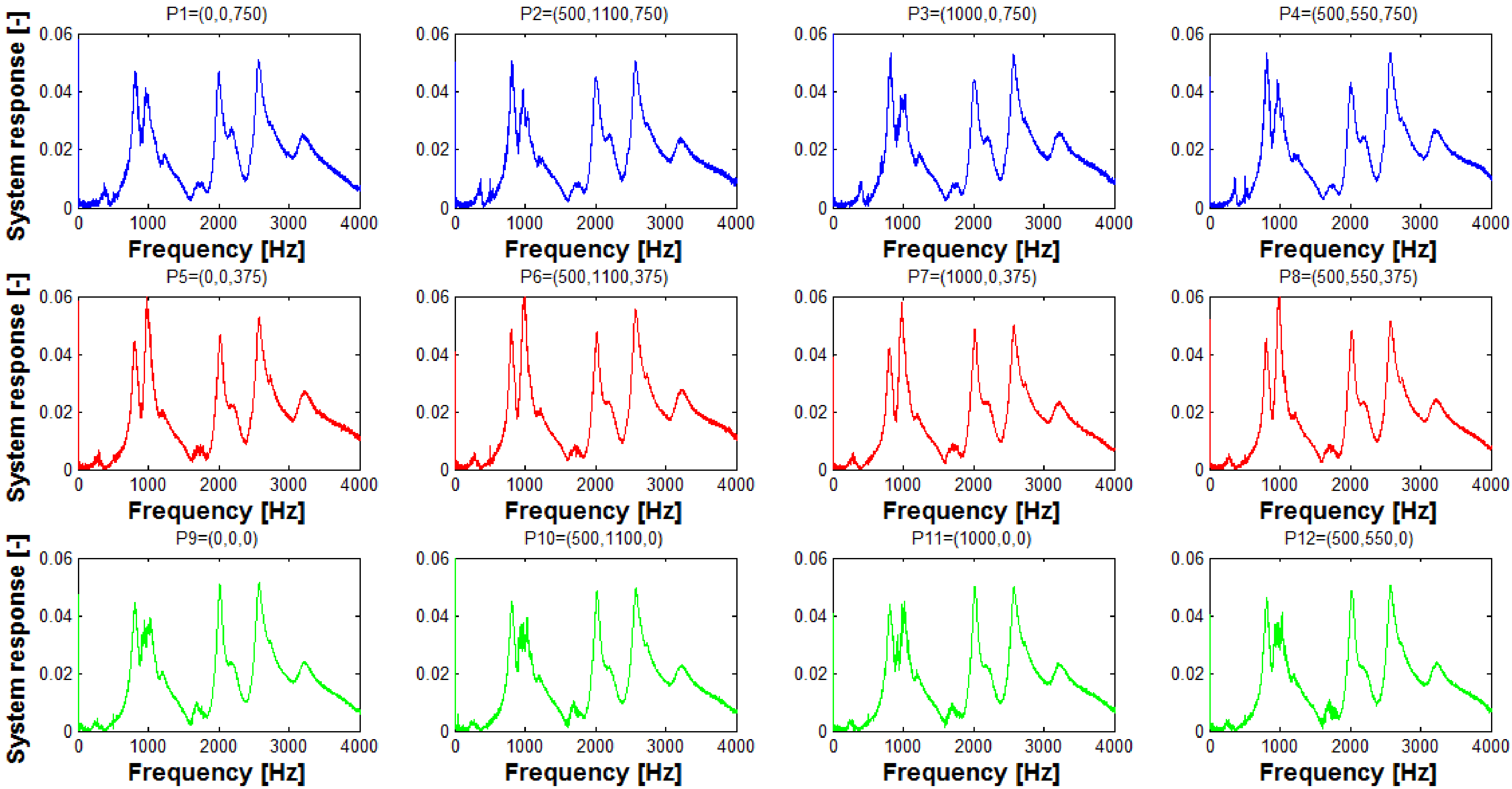


Vibration velocity[mm/s] rms 10-2000 Hz. at 8000 rpm



Vibration acceleration [g] rms 2-20 kHz at 8000 rpm

6.3 Setup 3



P1=(0,0,750)
P2=(500,1100,750)
P3=(1000,0,750)
P4=(500,550,750)
P5=(0,0,375)
P6=(500,1100,375)
P7=(1000,0,375)
P8=(500,550,375)
P9=(0,0,0)
P10=(500,1100,0)
P11=(1000,0,0)
P12=(500,550,0)
System response [-]
Frequency [Hz]
0.06
0.04
0.02
0
1000
2000
3000
4000

Several peaks are observed (396 Hz, 891 Hz, 965 Hz, 1207 Hz , 1661 Hz , 1993 Hz, 2192 Hz, 2550 Hz and 3065 Hz). They indicate critical frequencies of the tool holder associated with vibration modes. These critical speeds appear not to be sensitive the position .

No one of the peaks is within the frequency range of the spindle, and therefore is not possible to stablish a relation between these peaks and the one observed in the speed sweep carried out in machine tool A in setup 1 Y direction. From the impact test, the minimum critical speed is 396 Hz. This could be excited by the unbalance at a rotational speed of 23760 rpm. However this speed is beyond the operational speed of the spindle which is 12000 rpm.

A suggested test is to carry out the impact test at the spindle bearing housing. By this way may be possible to identify clearly the critical frequencies associated with spindle structure. However the equipment available was considered not appropriated for this test due to the significant mass of the spindle compared to the available impact hammer in the department.

6.4 Setup 4 and Setup 5

In order to compare if the frequency spectrum presented peaks in frequencies associated with bearing damage, it was necessary to calculate the frequencies related to bearings (See section 3.6.4).

The product number corresponding to bearing model could be found in the manual of the spindle manufacturer which corresponds to HCB 71920C and manufactured INA/FAG. The model correspond to hybrid angular contact ball bearings with 100 mm inner-diameter and contact angle of . 15°

Fortunately INA/FAG provides a database with bearing frequencies factors for most of their models. Bearing factors are factors which must be multiplied by the shaft speed in hertz in order to obtain the corresponding frequency. These factors are presented in Table 4. The designation used for these factor may vary from the on the section 3.6.4 but the concept is basically the same. For example INA/FAG used for BPFO the designation BPFFO.

Designation		**Frequency factor**
Overrolling frequency factor on outer ring	BPFFO	11.2327
Overrolling frequency factor on inner ring	BPFFI	13.7673
Overroling frequency factor on rolling element	BSFF	4.7148
Ring pass frequency factor on rolling element	RPFFB	9.4296
Speed factor of rolling element set for rotating inner ring	FTFF_i	0.4493
Speed factor of rolling element set for rotating outer ring	FTFF_o	0.5507

Table 4 Frequency factors machine tool A's spindle bearing HCB71920C. Source INA/FAG

The following diagrams show the frequency spectrum for vibration measure on: the dummy tool using CERS's capacitive probes and on the spindle housing obtained by accelerometers. The data was filtered using a digital high pass filter. The filter was designed using Matlab vibration analysis tool box and was not only aimed at removing the noise in the low part of the frequency spectrum but also the contribution of imbalance at synchronous speed. Because the vibration data was filtered up to spindle speed frequency, the frequency factors lowe than spindle frequency were not considered for analysis, this is FTFF_i and FTFF_o shown on Table 4.

The frequency axis FFT diagrams presented were delimited to the frequency range of interest. This means just above the "Overrolling frequency factor on inner ring" denominated BPFFI on Table 4.

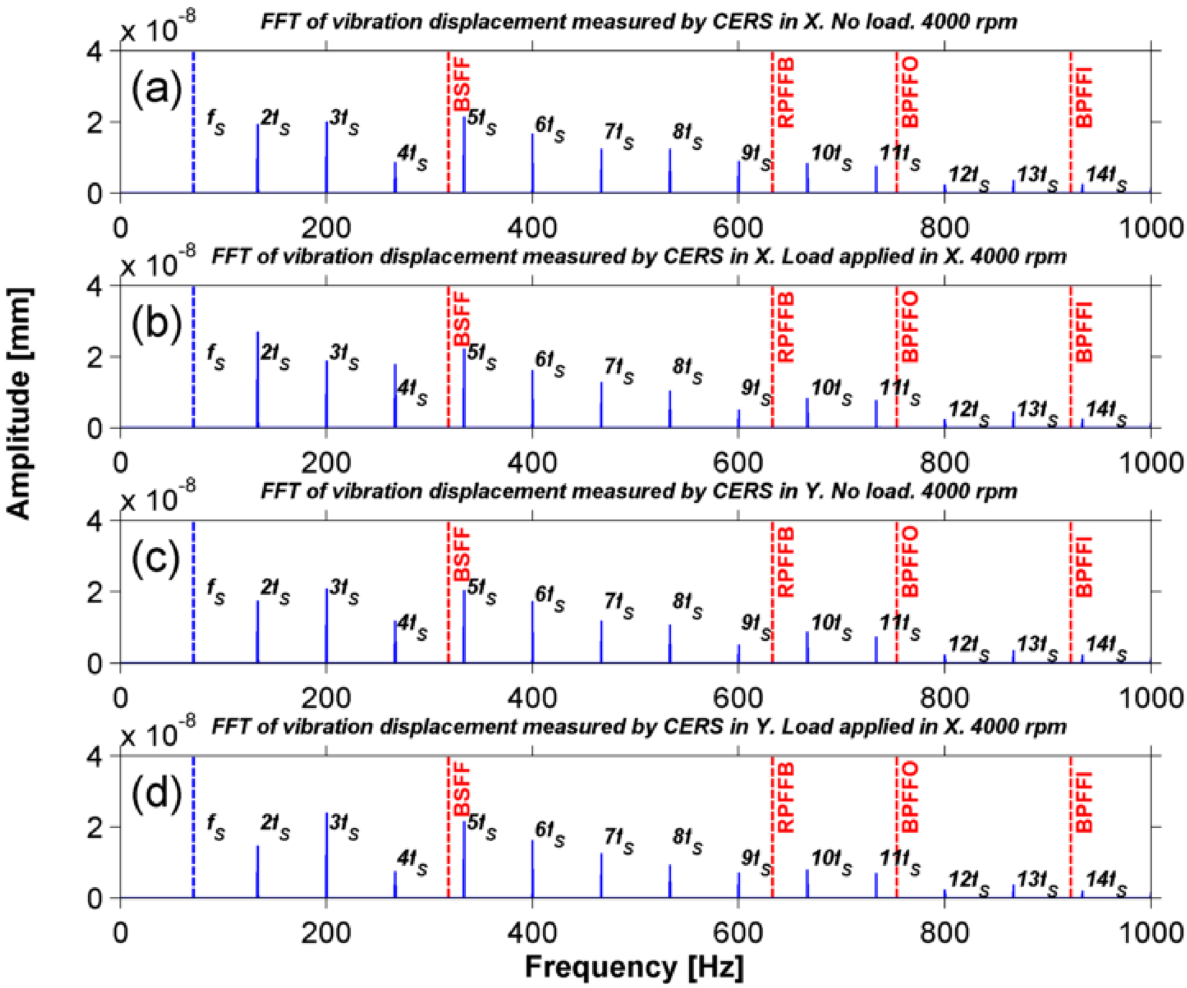


Figure 104 Vibration analysis of data acquired by CERS at 4000 rpm

At 4000 rpm, when measuring vibration displacement using CERS´s capacitive probes, multipliers of the spindle speed frequency (66.6 Hz) are found along the frequency spectrum as Figure 104 shows. These components show higher amplitudes close to spindle frequency Fs (which is represented by a blue dashed line) and progressively decreasing along the frequency spectrum.

When forced is applied in the X direction with CERS, $2F_S$ component of the spectrum appear to be increase in amplitude in the x direction Figure 104 (b). In contrast, a decrease in amplitude is observed in the Y direction for the component $2F_S$ as Figure 104 (c) and (d) illustrate.

The red dashed lines are markers indicating bearing frequencies for the specific spindle speed. However it is observed no part of the spectrum coincide with these markers. This fact implies that frequency no components of the spectrum measured using CERS associated with bearing frequencies were found when spindle is running at 4000 rpm.

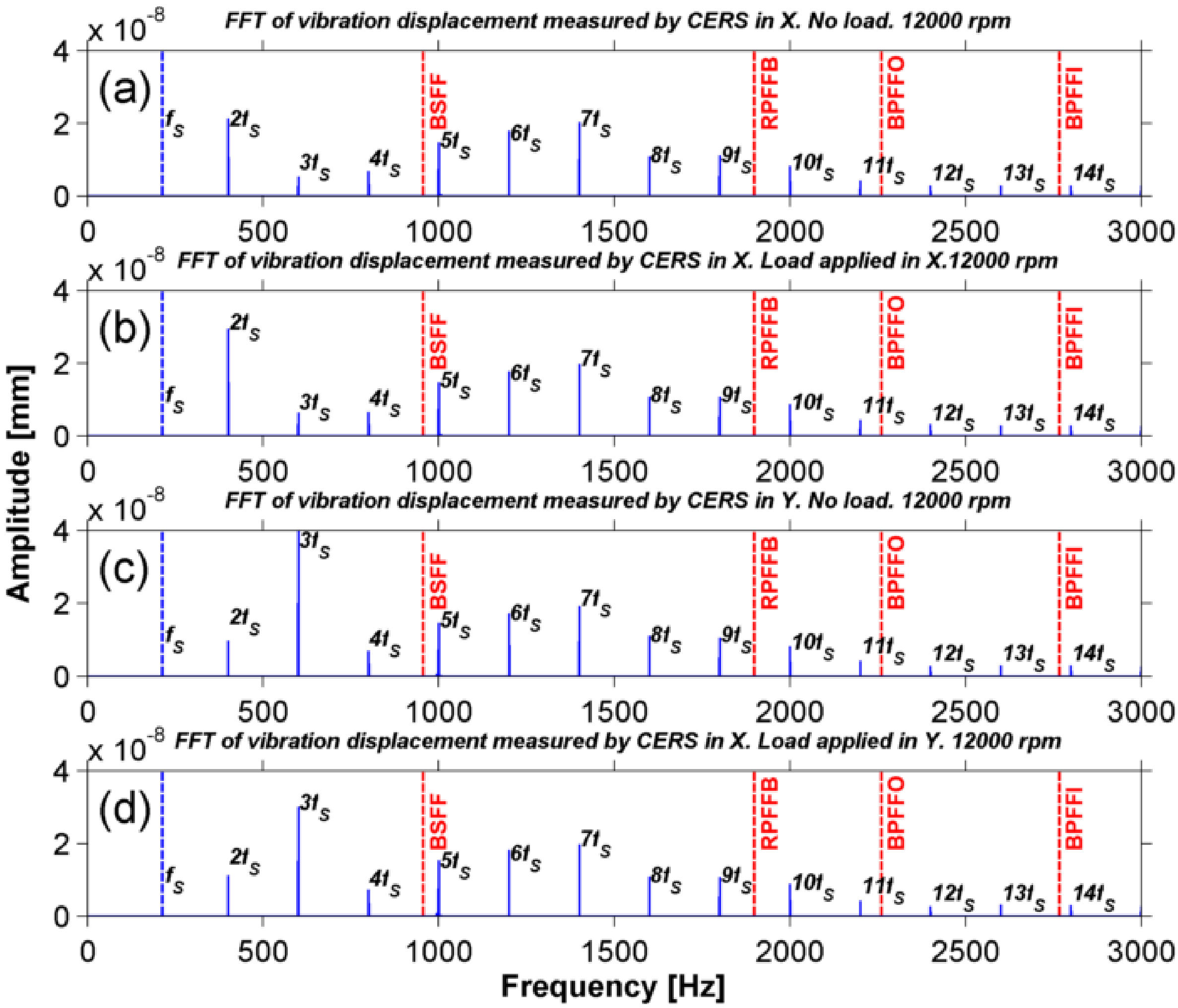


Figure 105 Vibration analysis of data acquired by CERS at 12000 rpm

When spindle running at 12000 rpm similar results as with a rotational speed of 4000 rpm, are obtained as Figure 105 evidences. Multipliers of the spindle frequency (200 Hz) are observed. For this case the frequency component at $2F_S$ in X increases in amplitude when load is applied on the dummy tool by CERS Figure 105 (b). In Y direction however, this frequency component remains unaffected as is observed in (c) and (d). But 3Fs decrease in magnitude when load is applied. Amplitudes in frequency components higher than 3Fs remain virtually the same for both unload and load case.

Similarly, bearing frequencies which match the bearing frequency markers are not observed in the spectrum.

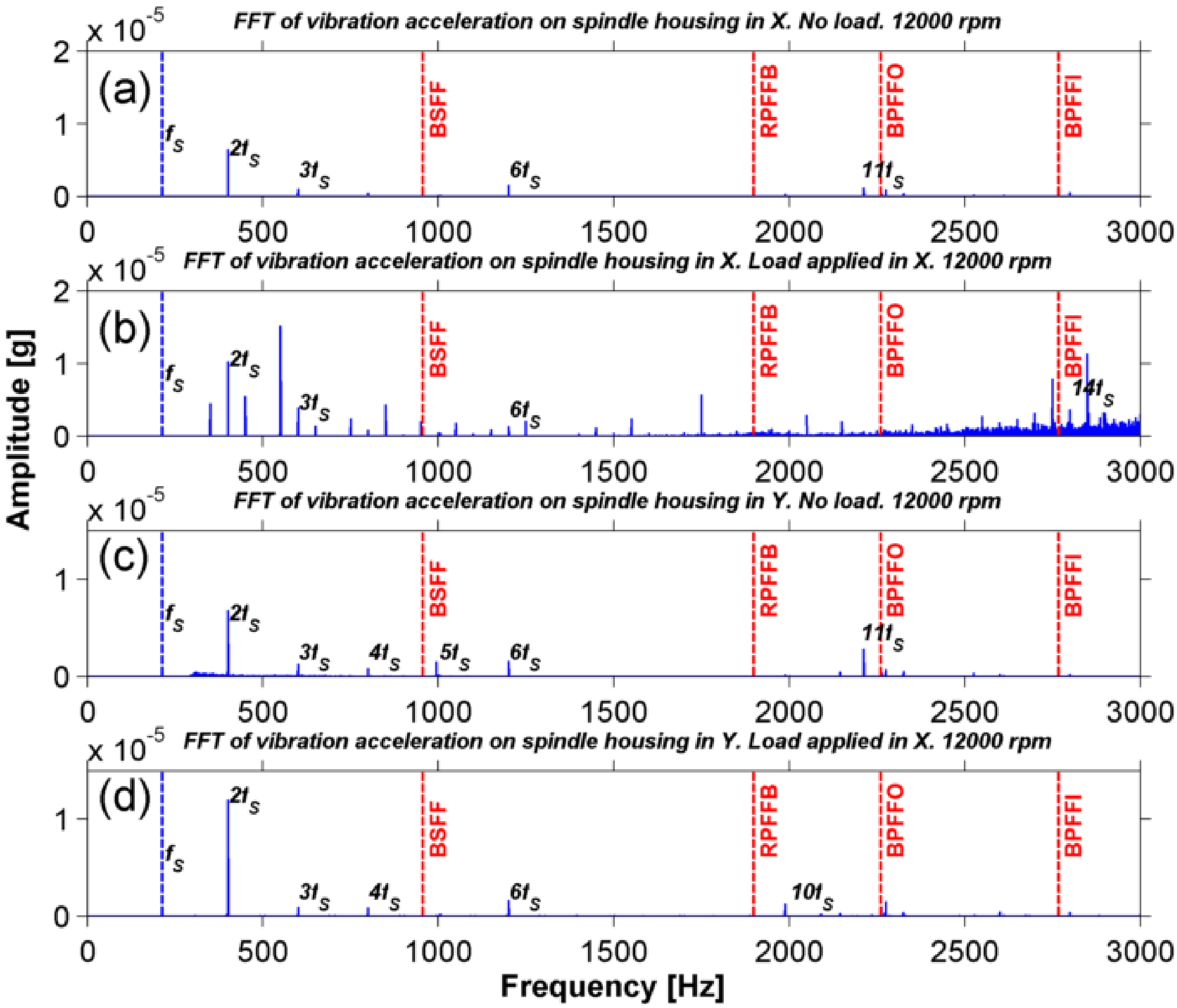


Figure 106 Vibration analysis data collected in spindle housing at 12000 rpm

The results of vibration measurement carried out on the spindle housing were compared with vibration measured on CERS. Some of the frequency components collected by the capacitive probes installed on CERS, were also visible in the frequency spectrum collected from accelerometers mounted on the spindle housing. For instead 2Fs, 3Fs, 6Fs. However most of the frequency components observed in CERS were not present at spindle housing's collected data.

When load is applied in X direction, different frequency components appears in this direction with some noise in the higher part of the frequency spectrum as it can be observed in Figure 106 (b). At the same time, frequencies which are very close BPFFI and BSFF appear in the X measurement direction of the spindle housing. If these frequencies correspond to the bearing frequencies calculated, they may have increased in amplitude due to the load applied by CERS on the dummy tool. The load from the tool is transferred to the bearing, affecting the amplitude in which bearing frequency components.

On Y direction is it possible to identify a frequency component close to BPFFO. This is present in both cases, with and without load.

*6.*CONCLUSIONS

RQ1 Is there any relation between vibration fingerprint of a machine tool spindles and their lifetime? In other words, can high initial values of vibration be an indicator that a spindle wears out rapidly?

High vibrations on spindles are frequently a symptom of unbalance or bearing damage. If the spindle has been recently acquired, bearing damage may appear as an unlikely source of high vibration, assuming appropriate manufacturing and mounting of these components. However, bearings are precision assemblies which can be negatively affected by vibration-related issues. For example, excessive imbalance of the rotor can reduce significantly bearing life, which leads to reduction in spindle life. The main difficulty though is to know with exactitude what should be considered “high vibration”.

RQ2 What are the main sources of vibration in a machine tool spindles? Besides which of these sources are related with condition monitoring of the spindle?

In this work several source of vibration in spindles were identified, of which broken tool, transmission problems, motor damage, chatter, critical speeds among others. However those more recurrent and interesting from the maintenance point of view are critical speed, imbalance and bearing damage. Critical speeds for a certain spindle should be identified in order to avoid operating and testing the spindle in those speeds, which in some cases can be harmful for spindles and can lead to misleading vibration data. Imbalance is also important because is a property that is susceptible to vary during the operation life of the spindle and should be monitored.

RQ3 At which extent vibration measurements at the spindle housing represent the truth condition of the spindle?

It is difficult to give an answer to this question for a number of reasons. In order to relate vibration levels with spindle condition an ideal scenario is to have reliable historical data of an appropriate population of similar machine tools where the evolution of the vibration can studied in detail. When total failure occurs, the vibration levels can be later validated with a visual inspection of the spindle components as spindle bearings. This is obviously expensive and requires close cooperation between industry and universities.

RQ4 Which factors may affect the reliability of vibration measurements on the spindle housing?

Based on the results from the experimental part, the angular mounting position of the accelerometer is strongly related with vibration levels measured. Therefore, careless installation of accelerometers could lead to misleading results when evaluating the evolution of spindle condition. However if the measurements are carried out following specific guidelines as SS 728000-1 standard, which recommends measuring in fixed positions X and Y, the risk for low repeatability when measuring can be reduced.

RQ5 Is it possible to establish vibration limits for machine-tool for condition monitoring by measuring vibration in spindle housing?

Determine vibration limits for spindles using theoretical models can be extremely difficult and impractical. Mainly because the large number of variables which can potentially affect vibration amplitude. These include bearing preload, housing rigidity, thermal and dynamic effects to name a few. A better approach to stablish conformity limits is to gather data from a similar population of spindles as in standard SS 728000-1. Because the low vibration levels present in integral spindles vibration data acquisition must be carried following very specific guidelines as the mentioned standard describes.

RQ6 Do current standards provide and effective tool for condition monitoring of integral spindles in machine tools? Could they be improved further to meet this point?

Regarding SS 728000-1 and as mentioned earlier, this standard has two main goals: being quality tool for new spindles and to serve as a vibration severity guide for operating spindles. When following the mentioned standard, a large amount of data that has to be managed due to close measuring points along the spindle speed range which is then multiplied by five measuring positions (X and Y direction in the back and front of the spindle housing, besides one Z position). In addition, in the experimental part of this work it was found that the back part of the spindle may be hardly or impossible to reach in some machine tool models. For these reasons, the mentioned standard requirements could be laborious for condition monitoring and may be better justified for one-time quality control check. A suggestion to improve this aspect is make a more clear differentiation between these two purposes in the standard. This could be achieved for example by simplifying the measurement procedure for condition monitoring.

The evaluation of spindle condition by the vibration severity indicators as rms, peak-to-peak, kurtosis and crest factor have to be made taking into account spindle running speed. As it shown in the results of setup 1, these indicators can vary significantly within the operating speed of the spindle. Because spindles are expected to operate in their entire speed range, when evaluating these indicators, the running speed should be also reported.

Tests with CERS showed that it is difficult to capture spindle bearing frequencies by using the two capacitive probes which this unit is equipped with. Nevertheless the spindle tested (Machine tool A) is relatively new with a very discrete history of use. As explained earlier, the spindle is almost entirely dedicated to research purposes. Therefore no extended damage on the spindle bearings was expected on it, which may explain the absence of spindle bearing frequencies on the frequency spectrum. Despite this, CERS seems promising for enhancing frequency components related to spindle bearing when vibration is measured on the spindle housing. Results obtained in setup 4 and 5 suggest that by applying a constant load on the dummy tool in a certain direction, frequency components as BSF and BFI can increase in amplitude, resulting more visible in the resulting frequency spectrum. The amplitude increase appears in the same direction of the load application. Based on these results, this technique could be used to detect onset damage on spindle bearings by studying the development of these “enhanced” bearing-related frequencies, which become only detectable in an early stage with help of CERS.

Future Work

Further research in needed to evaluate the full potential of CERS for bearing damage detection. In this scenario an experimental setup which includes damaged bearing could facilitate the validation of the results, which unfortunately was not possible in the present work due to technical challenges that this involves. Because it nearly impossible or impractical to disassemble a spindle for this purpose, an external “spindle housing” prototype could be designed. In this, bearings artificially damaged could be mounted and their vibration signature captured by CERS.

Another interesting topic worth to investigate is the effect of imbalance of the tool in vibration measurement carried out in spindle housing. In order to carry out a study of this type, balanced tooling system is needed. An appropriate number of tooling system with different balance grade and mass could be used. Alternatively a balance-adaptable dummy tooling system

Appendix

Appendix A: severity charts

Entek IRD

Entek IRD international, acquired by Rockwell Automation in 2000, has supplied globally machine monitoring equipment since many decades. The company has developed severity charts for monitoring different types of equipment based on large data collection of different machines (Girdhar & Scheffe, 2004). These guidelines do not define absolute values for vibrations but still give general criteria to warn when corrective actions must be taken during a maintenance program [7] as the following table.

Vibration velocity In/s peak	mm/s rms	Classification	Description
6 and up	10.7 and up	Very Rough	Severe vibration. Potentially unsafe. Make immediate detailed vibration analysis to identify trouble. Excessive vibration may cause oil-film breakdown. Consider shutdown to avoid in-service failure
3 to 6	5.4 to 10.7	Rough	Potentially damaging vibration. Make detailed vibration analysis to identify trouble. Rapid wear expected. Make more frequent periodic checks. Schedule for repair
2 to 3	3.6 to 5.4	Slightly Rough	Faults likely. Make detailed vibration analysis. Continue periodic checks. Schedule repairs as necessary.
1 to 2	1.8 to 3.6	Average	Minor faults. Continue routine periodic checks; watch for increase
0.5 to 1	0.9 to 1.8	Smooth	Typical well balanced. Well aligned equipment. Make routine periodic checks.
0.0 to 0.5	0.0 to 0.9	Very Smooth	Exceptionally well balanced, well aligned equipment. Make routine periodic checks.

This company also provides a specific chart for evaluation vibration severity in machine tool, which is display in the next page.

[7] Graph from https://rockwellautomation.custhelp.com/ci/fattach/get/11741/1193838370/redirect/1

Machine-Tool Vibration Tolerance Table

The chart below can be used as a guide for determining vibration tolerances for grinders and other machine tools where vibration can affect the quality of a finished product.

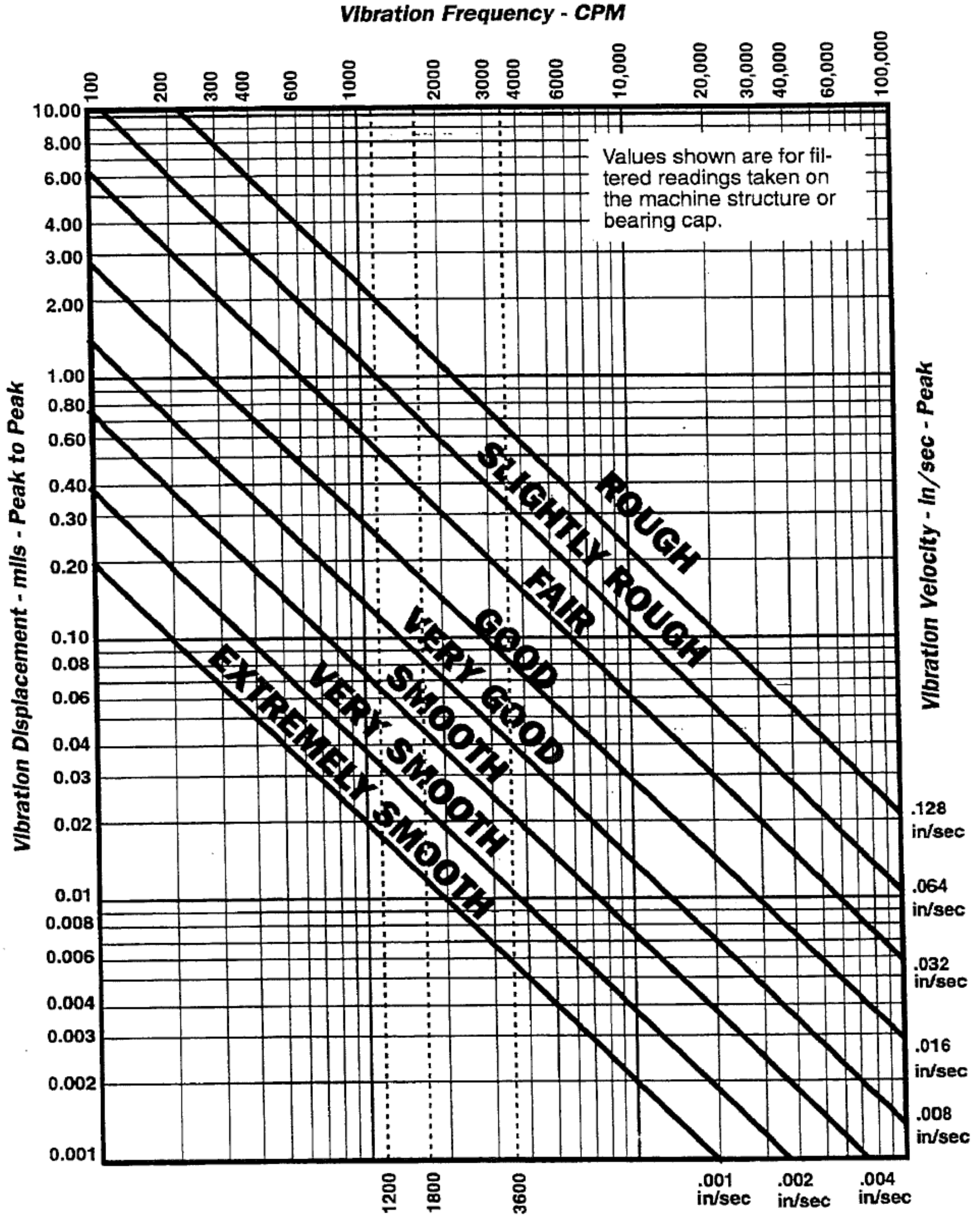


Figure 2-10. Typical vibration displacement tolerances for machine tools.

Technical Associates of Charlotte, PC

Technical Associate of Charlotte is a American company founded in 1961 dedicated to vibration monitoring services including training within vibration monitoring (TA, 2014). This organization recommends vibration alarms for different machines, covering machine tools within the speed range 500 to 600 000 rpm. Assuming the machine is not mounted in vibration isolators and the measurements are carried by accelerometers or velocity pickups as close as possible to bearing housing.

SUGGESTED OVERALL ALARMS BY MACHINE TYPE - METRIC
(Peak, Overall Velocity, mm/sec.)

MACHINE TYPE	GOOD	FAIR	ALARM
MACHINE TOOLS			
Motor	0 -2.5	2.5 -4.5	4.5
Gearbox Input	0 -4	4 -6	6
Gearbox Output	0 -2.5	2.5 -4.5	4.5
SPINDLES			
Roughing Operations	0 -2	2 -3	3
Machine Finishing	0 -1	1-2	2
Critical Finishing	0 -.5	.5 -1	1

Figure 107 Suggested Overall alarms by machine type (peak, overall velocity mm/sec)

Appendix B: balance quality grade according ISO 1940

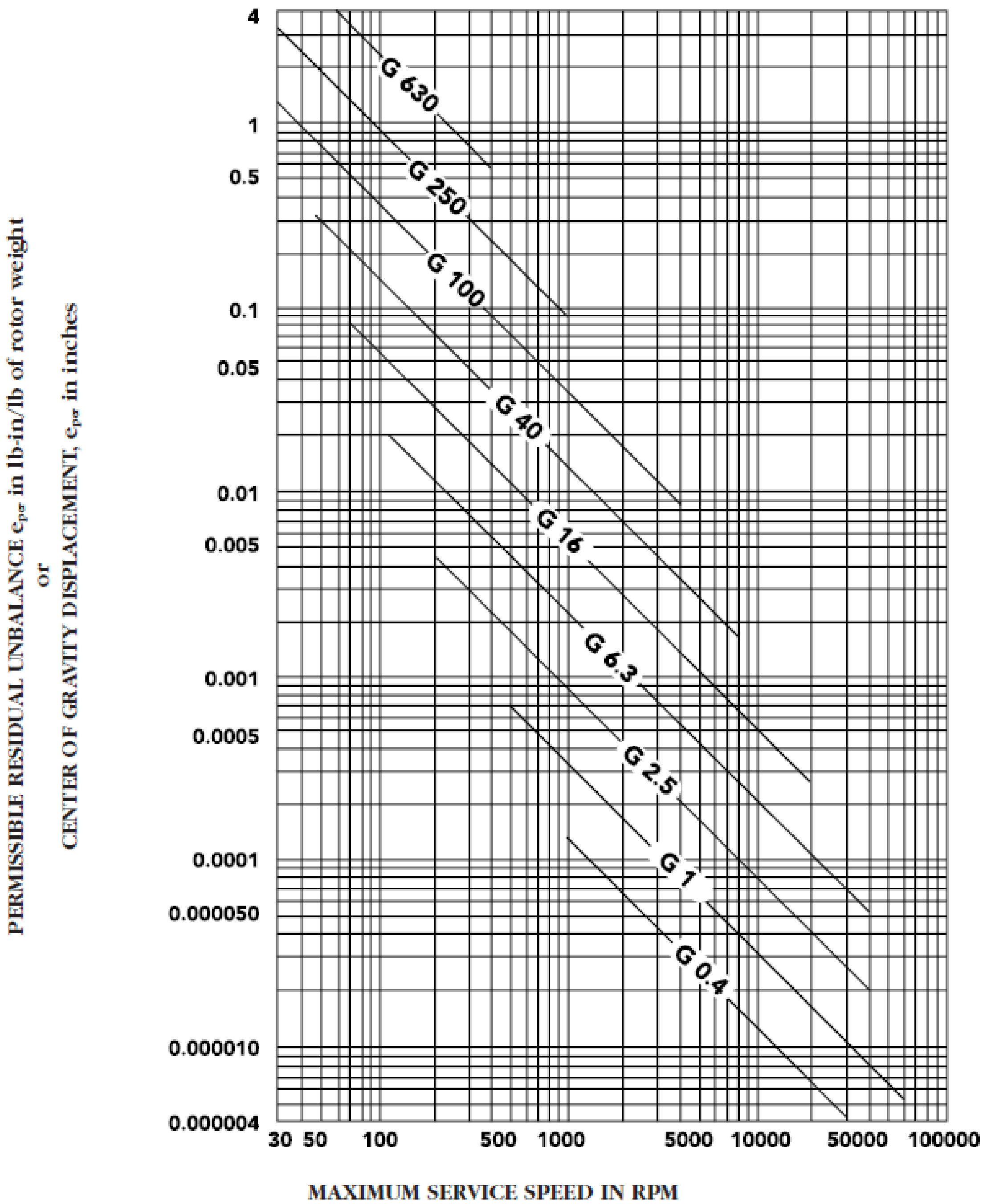
Table 1 Balance quality grades for various groups of representative rigid rotors (From ISO 1940/1)

Balance Quality Grade	Product of the Relationship ($e_{per} \times \omega$) [1] [2] mm/s	Rotor Types - General Examples
G 4 000	4 000	Crankshaft/drives[3] of rigidly mounted slow marine diesel engines with uneven number of cylinders[4]
G 1 600	1 600	Crankshaft/drives of rigidly mounted large two-cycle engines
G 630	630	Crankshaft/drives of rigidly mounted large four-cycle engines Crankshaft/drives of elastically mounted marine diesel engines
G 250	250	Crankshaft/drives of rigidly mounted fast four-cylinder diesel engines[4]
G 100	100	Crankshaft/drives of fast diesel engines with six or more cylinders[4] Complete engines (gasoline or diesel) for cars, trucks and locomotives[5]
G 40	40	Car wheels, wheel rims, wheel sets, drive shafts Crankshaft/drives of elastically mounted fast four-cycle engines with six or more cylinders[4] Crankshaft/drives of engines of cars, trucks and locomotives
G 16	16	Drive shafts (propeller shafts, cardan shafts) with special requirements Parts of crushing machines Parts of agricultural machinery Individual components of engines (gasoline or diesel) for cars, trucks and locomotives Crankshaft/drives of engines with six or more cylinders under special requirements
G 6.3	6.3	Parts of process plant machines Marine main turbine gears (merchant service) Centrifuge drums Paper machinery rolls; print rolls Fans Assembled aircraft gas turbine rotors Flywheels Pump impellers Machine-tool and general machinery parts Medium and large electric armatures (of electric motors having at least 80 mm shaft height) without special requirements Small electric armatures, often mass produced, in vibration insensitive applications and/or with vibration-isolating mountings Individual components of engines under special requirements
G 2.5	2.5	Gas and steam turbines, including marine main turbines (merchant service) Rigid turbo-generator rotors Computer memory drums and discs Turbo-compressors Machine-tool drives Medium and large electric armatures with special requirements Small electric armatures not qualifying for one or both of the conditions specified for small electric armatures of balance quality grade G 6.3 Turbine-driven pumps
G 1	1	Tape recorder and phonograph (gramophone) drives Grinding-machine drives Small electric armatures with special requirements
G 0.4	0.4	Spindles, discs and armatures of precision grinders Gyroscopes

1) $\omega = 2\pi n/60 \approx n/10$, if n is measured in revolutions per minute and ω in radians per second.
2) For allocating the permissible residual unbalance to correction planes, refer to "Allocation of U_{per} to correction planes."
3) A crankshaft/drive is an assembly which includes a crankshaft, flywheel, clutch, pulley, vibration damper, rotating portion of connecting rod, etc.
4) For the purposes of this part of ISO 1940/1, slow diesel engines are those with a piston velocity of less than 9 m/s; fast diesel engines are those with a piston velocity of greater than 9 m/s.
5) In complete engines, the rotor mass comprises the sum of all masses belonging to the crankshaft/drive described in note 3 above.

Appendix C: Maximum permissible residual imbalance according to ISO 1940

Figure 1-A Maximum permissible residual unbalance, e_{per}
(Imperial values adapted from ISO 1940/1)



Appendix D: Bearing failure mode and their root causes matrix

Failure matrix

Possible causes / Failure modes with characteristics			Operating conditions					Environ-mental factor				Lubrication					Mounting							Other					
			Overload	Overspeed	Excessive freq. of load/speed changes	Vibrations	Shaft/housing deflection	Temperature too high/low	Dust and dirt ingress	Water ingress	Electrical leakage	Wrong viscosity	(Consistency) additives selection	Lack of lubricant	Excess of lubricant	Impurities	Incorrect handling (shock loads)	Mounting procedures	Fit too tight	Fit too loose	Tilting/misalignment	Incorrect setting	Incorrect locating (clamping)	Storage	Transportation (vibration/shock)	Bearing selection	Equipment design	Manufacturing concerns	Material concerns
Fatigue		Flaking, spalling, peeling	•				•	•				•	•	•			•	•	•		•	•			•	•	•	•	•
		Burnishing, microcracks		•	•	•		•	•	•	•	•	•	•	•	•		•				•		•		•	•		•
Wear	Abrasive	Excessive wear		•	•	•	•	•	•	•	•	•	•	•		•			•	•	•	•	•			•			•
		Scratches, scores		•	•				•				•			•	•	•							•				
	Adhesive	Seizing marks, smearing	•		•		•	•				•	•	•	•			•	•	•	•					•	•	•	•
		Hot runners	•	•	•		•	•				•	•	•	•			•	•		•	•	•			•	•	•	•
Corrosion		Moisture corrosion								•			•			•		•						•	•		•		
		Fretting corrosion	•		•	•	•													•	•	•	•			•	•	•	
		False brinelling			•	•							•							•				•	•	•			
Electrical erosion		Craters, fluting									•																•		
Plastic deformation		Depressions	•				•	•							•		•	•	•		•	•	•	•	•	•	•		•
		Debris indentation							•							•	•	•									•	•	
		Nicks, gouges															•	•										•	
Fracture & cracking		Forced fracture	•	•				•									•	•	•		•	•	•			•	•	•	•
		Fatigue fracture	•	•	•	•	•													•	•	•	•			•	•	•	•
		Thermal cracking	•	•	•			•				•		•	•			•	•			•					•		•

Appendix E: Interview with Johan Hendén, application Engineer at SKF

Question form sent by email 2015/11/26. Answered 2016/01/11

According with your (SKF) experience in spindles…

1) Which components have most influence in spindle life length?

On Machine tool spindles the bearings are the most sensitive component in the application. The bearings are also the "heart" of the spindle so any main problems are often reflected and detected through the bearings. Often other issues are affecting the bearing performance and lifetime and thus bearing problems are the most common reason for service. But again, it is often other problems, one or several that cause unsatisfied bearing behavior.

2) Which are the most common bearing damages in spindle for machine tools SKF services?

When it comes to visual bearing defect that we can detect on the bearings raceways (Inner, Outer and roller element)it is very hard to point out one or another reason. A majority of all bearings have been burned up when SKF receive them. All initial damages has been erased due to extreme high temperature. Since spindles are running in such a extreme speeds and loads any problems impacting the bearings make this time to total jam very short.

3) Which are the common root causes to these bearing damages?

Since most of the evidence have been burned up we can't point out the most common reason reason behind bearing failures. But we always very carefully review the spindles different component to detect any issues that might or have an impact on the spindle performance and lifetime. Here we often need to adjust and restore bearing seats and journals due to geometry deviations, out of dimensional tolerance's, misalignment between front and rear bearing position, unbalance in shaft, exchange seals (water and contamination penetration) adjustments to obtain optimal bearing preload.

4) Among motor spindles (built-in-motors spindle) with rolling bearings (conventional or hybrids) is there any differences in the answer for questions 1 and 2 compared with other spindles with other transmission systems?

No, there is no major difference on the outcome of hybrids and standard bearing when the spindle jam is an fact. The hybrids are more forgiving when it comes to lubrication problems. Hybrids have many advantages compare to standard steel/steel bearings, e.g. they run run cooler due to lower density, higher speed capacity, act like electrical isolation.

But if we would be able to stop the spindle before break down it might be possible to see some differences between the two variants.

5) Is there any connection between vibration levels in spindles when a (spindle) fingerprint is taken and spindle life length? In other words, could high vibration levels imply a shorted life length? Justify your answer.

All rotating parts with bearings have vibrations. But you need to decide which level is acceptable for the application type and what operation it shall perform in (machine operation type). All vibrations will of course impact negative on the bearing lifetime, but might not always be the main reason for failure. Vibrations cause a higher stress on the raceway surfaces and thus the calculated bearing lifetime L10 will be reduced. So if high vibration levels the stress level in the bearing steel will increase even more. Further you need to verify incase the vibration are bearing related or mechanical related. Bearing related are measured often by the Envelope technique (Env) and the mechanical by speed (mm/s). Bearing related vibrations are given a status of the bearing (too high or too low preload, clearance, bearing damages, current patterns, lubrication issues e.g.). Mechanical vibrations are more related to unbalance, misalignment, looseness, high run-out. So depending what vibrations we are talking about and the level of it they all have different level of impact on the bearing lifetime.

Appendix F: Machine tool A's Spindle data sheet

Technisches Datenblatt AC-Motorspindel
technical datasheet AC-motorspindle

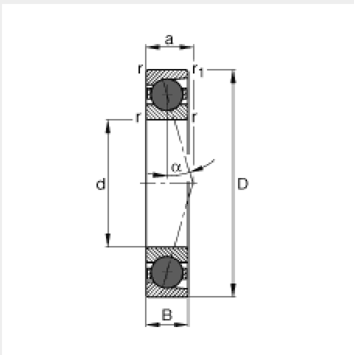
Artikelnr./article 000.650.354	**Kurzbez.**	**Datum**/date 20.04.10	
Impulsgeber *Encoder*	**Type** *Type* GEL 2443 KN RG 3 K150-09E	**Art**	Zahnradgeber *gear transmitter*
	Fabr. *Fabr.* Lenord & Baur	**Imp/Signal** *Pulse*	sin/cos 1 Vss, 256 Zähne
Regler *Amplifier:*	Simens Simodrive 611 D - 120/150/193 A - Hochlaufzeit: 4 sec		
Erforderl. Glättungsdrossel *Required smoothing coil*	mH	Hz	

		Wert:	**Bemerkung:**
Kühlung: Motor *Cooling motor*	**Medium** *Medium:*	Wasser *water*	mit Additiven *with additives*
	Volumen *Volume*	16 l/min	
	Eintrittstemp. max. *max. temperature at entry*	30 °C	
	Druck p_{max} *Pressure*	4 bar	
	Abzuf. Verlustlstg P_v *Max. loss power*	4,3 kW	
Kühlung:Spindel *Cooling spindle*			
Kühlung:Spindelkopf *Cooling spindle head*			Ja *Yes*
Kühlschmiermittel *Coolant:*	**Ausführung** *Design*	zentral	1K-GDV Fa. Ott
Inneres KSM *Inner coolant supply*	**Druck p_{max}** *Pressure*	80 bar	
	Filterfeinheit *Filtration grade*	< 50 μm	
Äusseres KSM *Outer coolant supply*	**Druck p_{max}** *Pressure*	bar	Nein *No*
	Anzahl der Düsen *Quantity of nozzles*		
Lagerung: Spindel *Bearing spindle*	**Lagerart** *Bearing construction*	HCB 71920	T-O-T hydr. variable Lagervorspannung
	Steifigkeit ax./rad. *Stiffnes ax/rad*	N/μm	
Lagerschmierung *Bearing lubrication*	**Art** *Oil/Grease*	☐ **Fett** *Grease* ☒ **Öl** *Oil*	Ölabsaugung
	Sorte *sort*	ISO VG68	
Abdichtung *Sealing*	**Art** *Type of construction*	Labyrinth *labyrinth*	
	Sperrluft (wasser- und ölfrei) *Air purge without water- and oilfree*	ca. 5 bar	Ja *Yes*
WZG.-Spannsystem *Tool clamping system*	**Bauart** *Type of construction*		Federspanner *spring pile*
	Wzg.-Aufnahme *Tool taper*	HSK-A-100	DIN 69893

Appendix G: Machine tool A's Spindle Bearings data sheet

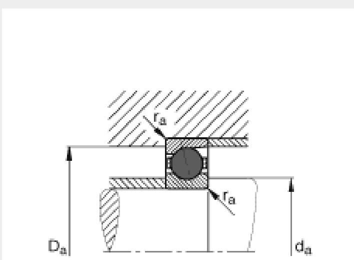
Spindle bearings HCB71920-C-T-P4S

adjusted, in pairs or sets, contact angle α = 15°, with ceramic balls, restricted tolerances



d	100 mm	
D	140 mm	
B	20 mm	

a	26 mm	
D_a	133 mm	Tolerance: H12
d_a	107 mm	Tolerance: h12
$r_{1\ min}$	1,1 mm	
$r_{a\ max}$	0,6 mm	
$r_{a1\ max}$	0,6 mm	
r_{min}	1,1 mm	
α	15 °	Contact angle



m	0,658 kg	Mass
C_r	60000 N	Basic dynamic load rating, radial
C_{0r}	31500 N	Basic static load rating, radial
$n_{G\ Fett}$	13000 1/min	Limiting speed for grease lubrication
$n_{G\ Öl}$	20000 1/min	Limiting speed for minimal quantity oil lubrication
C_{ur}	2950 N	Fatigue limit load, radial

medias

SCHAEFFLER

LuK INA FAG

Bearing analysis

Input

Bearing:

Designation HCB71920-C-T-P4S

Basic frequency factors related to 60/min:

Overrolling frequency factor on outer ring	BPFFO	11.2327
Overrolling frequency factor on inner ring	BPFFI	13.7673
Overrolling frequency factor on rolling element	BSFF	4.7148
Ring pass frequency factor on rolling element	RPFFB	9.4296
Speed factor of rolling element set for rotating inner ring	FTFF_i	0.4493
Speed factor of rolling element set for rotating outer ring	FTFF_o	0.5507

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References

Abele, E., Altintas, Y. & Brecher, C., 2010. Machine tool spindles. *CIRP Annals-Manufacturing Technology,* 59(781-802).

Abele, E. & Korff, D., 2011. Avoidance of collision-caused spindle damages—Challenges, methods and. *CIRP Annals - Manufacturing Technology,* Volume 60, pp. 225-228.

Archenti, A., 2014. *Process machine Interaction.Lecture 9. Material from the course Industrial Process Engineering. KTH (unpublished),* s.l.: s.n.,

Archenti, A., 2015. *Dr. Engineer and Professor of Production Enginnering Department KTH. (Personal Comunication).* s.l.:s.n.

Azima DL, 2009. *History of vibration analysis used for machinery maintenance.* [Online]
Available at:
http://azimadli.com/vibman/historyofvibrationanalysisusedformachinerymaintenance.htm

Bergbom, P., 2015. Bearing doctor's analysis find root causes of problems and breakdown in machinery equipment. (In Swedish) Original title: Lagerdoktorns analys hittar grundorsaken till problem och haverier i maskinutrustning. *VERKSTÄDERNA,* Issue 6, pp. 12-17.

Bóden, H., Ahlin, K. & Carlsson, U., 2014. Chapter 14. Machine Monitoring. In: *Applied Signal Analysis. Department of Aeronautical and Vehicle Engineering. Royal Institute of Technology KTH.* s.l.:s.n., pp. 246-274.

Brecher, C., Esser, M. & Witt, S., 2009. Interaction of manufacturing process and machine tool. *CIRP Annals-Manufacturing Technoloy,* Volume 58, pp. 588-607.

CADEM, 2014. *Maximizing spindle power usage.* [Online]
Available at: http://cadem.com/cnc-gyan/understanding-the-spindle-motor.pdf

Castelbajac, C., 2016. *Personal Comunication (telephone call)* [Interview] (06 01 2016).

Cease, B., 2011. *Machine vibration standards: Good, better and best.* [Online].

CECIMO, 2011. *Study on Compitetivness of the European Machine Tool Industry.* s.l.:s.n.

CECIMO, 2015. *HE EUROPEAN MACHINE TOOL PRODUCTION EXPECTED TO GROW 3% IN 2015.* [Online]
Available at:

http://www.cecimo.eu/site/publications/?no_cache=1&tx_ttnews%5Btt_news%5D=255&cHash=b3a99f3fed97660d8b10f0cbadad2021

De a Calle, N. L. & Lamikiz, A., 2009. Chapter1 Machine Tools for Removal Processes:. In: *Machine Tools for High Performance Machining.* s.l.:Springer, pp. 1-45.

de Castelbajac, C., 2012. Advance monitoring and improvement of HSM process. (In French). Original title: Surveillance avancée et amélioration du procédé d'UGV. *Doctoral thesis of the University of Nantes.*

de Castelbajac, C., Ritou, M., S., L. & Furet, B., 2014. Monitoring of distributed defects on HSM spindle bearings. *Applied Acostics,* Volume 77, pp. 159-168.

El-Thalji, I. & Jantunen, E., 2015. A summary of faur modelling and predictive health monitoring of rolling element bearings. *Mechanical Systems and Signal Processing,* Volume 60-61, pp. 252-272.

EMAG, 2015. *Vertical turning and grinding center: Hard turning, scroll-free turning, grinding.* [Online]
Available at: http://www.emag.com/machines/grinding-machines/vertical-turning-and-grinding-centers/vsc-ds-dds.html
[Accessed 10 december 2015].

ENKET, 2002. *VISTeCTM user's guide. Appendix A:Vibration Severity charts.* [Online]
Available at: https://rockwellautomation.custhelp.com/ci/fattach/get/100515/

FAG, n.d. *Super precision bearing machine tool products*. [Online].

Fan, H., Jing, M. & Liu, H., 2013. Experimental method to evaluate unbalance vibration for machine tool spindle integrated with automatic balancer.

Fleischer, J., Broos, A., Schopp, M. & Wieser, J., 2008. Sustainable Design of Machine Tools through Load-Dependent Interventions and Adapted Services. In: *Manufacturing Systems and Technologies for the New Frontier.* s.l.:Springer London, pp. 173-176.

Fleischer, J. et al., 2009. Lifecycle-oriented component selection for machine tools based on multibody simulation and component life prediction. *CIRP Journal of Manufacturing Science and Technology 1,* Issue 1, pp. 179-184.

GARDNER Business Media Inc., 2015. *2015 The World Machine-Tool Output & Consumption Survey.* [Online]
Available at: http://www.gardnerweb.com/articles/2015-world-machine-tool-output-and-consumption-survey
[Accessed October 2015].

Gardner, 2015. *International Machine Tool Production and Consumption (Metalcutting and Metal Forming).* [Online]
Available at: http://www.gardnerweb.com/forecast/international.htm

Gibson, D., 2000. *For high speed and high productivity, keep cool and stay calm.* [Online]
Available at: http://www.mmsonline.com/articles/for-high-speed-and-high-productivity-keep-cool-and-stay-calm
[Accessed 10 September 2015].

Girdhar, P. & Scheffe, C., 2004. *Practical Machinery Vibration Analysis and Predictive Maintenance. - Predictive maintenance techniques: Part 1 predictive maintenance basics..* s.l.:Oxford: Newnes.

GMN, 2015. *High speed spindle for manual tool change.* [Online].

Groschopp, 2012. *How To: Differentiate Synchronous and Asynchronous Motors.* [Online]
Available at: http://www.groschopp.com/synchronous-vs-asynchronous/

Hagberg, L. & Henriksson, T., 2010. *World cass Maintenance.(Swedish) Underhåll i världsklass..* s.l.:OEE Consultants.

Häggblom, A., 2013. CONDITION BASED MAINTENANCE ON MACHINE TOOLS Evaluation of the condition based maintenance on machine tools at Volvo Cars in Skövde and how it can evolve (in Swedish). *Bachelor thesis within Automation. University of Skövde.*

Haimowitz, J. & Warren, J., 2007. *Economic Value of Standardization..* [Online]
Available at: https://www.scc.ca/sites/default/files/migrated_files/DLFE-342.pdf

Hendén, J., 2015. *Servicing spindles at SKF. (interview carried of by email). Interviewer Eduardo Torres* [Interview] (December 2015).

He, X., 2015. Recent development in reliability analysis of NC machine tools. *Int J Adv Manuf Technol.*

Hoshi, T., 2006. Damage Monitoring of Ball Bearing.

HSD Mechatronics, 2015. *Spindle Catalog. HP Milling line.* [Online]
Available at: http://www.hsdusa.com/viewdoc.asp?co_id=1031
[Accessed 25 November 2015].

IRD Balancing, 2009. *Balance Quality Requirements.* [Online]
Available at: http://www.irdbalancing.com/downloads/techpaper1balqualityreqmts.pdf
[Accessed 11 december 2015].

ISO TC39/SC 2, 2011. *Supporting document for NWIP for Machine tool vibration (N1837). Reference number N 1869,* s.l.: ISO.

ISO, 2003. *ISO 1940-1:2003 Mechanical vibration — Balance quality requirements for rotors in a constant (rigid) state — Part 1: Specification and verification of balance tolerances.* [Online]
Available at: https://www.iso.org/obp/ui/#iso:std:iso:1940:-1:ed-2:v1:en

ISO, 2009. *ISO 10816-3:2009. Mechanical vibration — Evaluation of machine vibration by measurements on non-rotating parts — Part 3: Industrial machines with nominal power above 15 kW and nominal speeds between 120 r/min and 15 000 r/min when measured in si.* [Online]
Available at: https://www.iso.org/obp/ui/#iso:std:iso:10816:-3:ed-2:v1:en

ISO, 2013. *ISO/TR 19201:2013. Mechanical vibration — Methodology for selecting appropriate machinery vibration standards.* [Online]
Available at: https://www.iso.org/obp/ui/#!iso:std:61991:en

ISO, 2015a. *ISO members.* [Online]
Available at:
http://www.iso.org/iso/home/about/iso_members.htm?membertype=membertype_MB

ISO, 2015b. *Who develops ISO standards?.* [Online]
Available at: http://www.iso.org/iso/home/standards_development/who-develops-iso-standards.htm

ISO, I. O. f. S., 2015. *ISO derivelables: ISO/TR Technical Report.* [Online]
Available at: http://www.iso.org/iso/home/standards_development/deliverables-all.htm?type=tr
[Accessed 22 11 2015].

Jedrzejewski, J., Kwasny, W. & Modrzycki, W., 2008. Limiting Precision DistortionsIn Spindle Unit of HSC Machining Centre. *Journal of Machine Engineering8(1):44–53.,* 8(1), pp. 44-53.

Jorgensen, B. & Shin, Y., 1996. Dynamics of Machine Tool Spindle/Bearing Systems Under Thermal Growth.. *Journal of tribology. 119(4), 875-882 (Oct 01, 1997)* , 119(4), p. 8.

Jun, 2002. Hybrid rapid prototyping system using machining and deposition.. *Computer aided design,* Volume 34, pp. 741-754.

KENNAMETAL, 2013. *Kennametal tooling system cataloge.* [Online]
Available at:

https://www.kennametal.com/content/dam/kennametal/kennametal/common/Resources/Catalogs-Literature/Metalworking/Tooling%20Systems%20Master/A-12-02809_KMT_ToolingSystems2013_sectionM_Technical.pdf

Kennedy, B., 2004. Calming Chatter. Strateges for minimizing tool chatter. *CUTTING TOOL ENGINEERING Magazine*, 56(7).

KIT, K. I. o. T., 2007. *VIPRO Projekt.*. [Online]
Available at:
http://www.produktionsforschung.de/PFT/verbundprojekte/vp/index.htm?VP_ID=1884
[Accessed 10 December 2015].

Kumaraswamy, S. et al., 2002. *Standarization of absolute vibration level and damage factors for machinery Health Monitoring.* s.l., s.n.

Leach, B., 2011. *2011 Second Act winner: Tony Shipley.*. [Online]
Available at: http://www.bizjournals.com/cincinnati/print-edition/2011/09/16/2011-second-act-winner-tony-shipley.html

Mannan, M. & Stone, B., 1998. The Use of Vibration Measurements for Quality Control of Machine Tool Spindles.

Mannan, M. & Stone, B., 1998. The Use of Vibration Measurements for Quality Control of Machine Tool Spindles. *Advance Manufacturing Technology,* Volume 14, pp. 889-893.

Marsh, E. R., 2008. *Precision Spindle Metrology.* Lancaster, Pennsylvania 17602-4967, U.S.A.: DEStech Publications, Inc..

Masao, H., 2011. *The Uncertain Future of Japan's Machine Tools Industry.* [Online]
Available at: http://www.nippon.com/en/currents/d00007/#note01

McDermott, T., 2011. *Vibration institute Brg. Failure Presentation.* [Online]
Available at: http://www.vibration.org/Presentation/May%202011/SKF%20-%20Vibration%20Institute%20Brg.%20Failure%20Presentation%207-8-2010.pdf

Mikler, J., 2014. *Condition Based Monitoring (part 1).Course MG1007 Modern Maintenance Methods and Techniques .KTH Royal Insitute of Technology.(Unpublished).* s.l.:s.n.

Mikler, J., 2014. *L7: RCM/FMEA. Course MG1007 Modern Maintenance Methods and Techniques .KTH Royal Insitute of Technology.(Unpublished).* s.l.:s.n.

Mitchell, J. S., 2007. From Vibration Measurements to Condition Based Maintenance. Available at http://www.ludeca.com/casestudy/mitchell_doc.pdf. *Sound and Vibration Magazine,* Volume 40th Anniversary Issue, pp. 62-75.

MOBIUS INSTITUTE, 2008. *Calculators, Simulators & Guides:SEVERITY CHART: ISO 10816 Velocity*. [Online]
Available at: http://www.mobiusinstitute.com/site2/detail.asp?LinkID=54
[Accessed 03 march 2015].

Mobley, R. K., 2000. *Vibration Fundamentas.* s.l.:Butterworth-Heinemann.

Morales-Espejel, G. & Gabelli, A., 2015. A major step forward in life modeling. *evolution magazine by SKF,* Issue 4, pp. 21-28.

NAVSEA, 2007. *Reliability-Centered Maintenance RCM Handbook.* s.l.:s.n.

Neuebauer, R., Fisher, J. & Pracedicow, 2011. Condition-based preventive maintenance of main spindles.

Noel, D., Ritou, M., Le Loch, S. & t Furet, B., 2012. Bearings Influence on the Dynamic Behavior of HSM Spindle. *ASME 2012 11th Biennial Conference on Engineering Systems Design and Analysis,* Volume 4, pp. 107-114.

NP, n.d. *Slide shaft/Spindle shaft cataloge.* [Online]
Available at: https://www.nbcorporation.com/product/lineup/shaft/shaft.html
[Accessed 20 11 2015].

NSK, 2009. Machine tool spindle bearing: selection and mounting guide.

NSK, 2013. *Rolling Bearings.* [Online].

NTN, 2012. *NTN Precision Rolling Bearings.* [Online].

OPTILAST, 2013. *OPTILAST Das Projekt.* [Online]
Available at:
http://www.wbk.kit.edu/wbkintern/Forschung/Projekte/Optilast/Projektbeschreibung/index_d.php
[Accessed 10 December 2015].

OTT-JAKOB Spanntechnik, 2013. *Modular Clamping Technology Cataloge.* [Online].

Precision Spindle & Accesories Inc., 2015. *Checking Spindle Drawbar Force with Clamprite Spindle Drawbar Force Gage.* [Online]
Available at: http://www.precisionspindleinc.com/spindle-drawbar-force-gage-why-check.html
[Accessed 21 11 2015].

Quintana, G., de Ciurana, J. & Campa, F., 2009. Chapter 3: Machine tool Spindles. In: *Machine tools for High Performance Machining.López de Lacalle, L. N.; Lamikiz, A..* s.l.:Springer.

Ramachandran, K., Fathi, K. & Rao, B., 2010. Recent Trends in Systems Performance Monitoring & Failure Diagnosis. *ndustrial Engineering and Engineering Management (IEEM), 2010 IEEE International Conference on,* pp. 2193 - 2200.

Rao, S., 2004. *Mechanical Vibration.* Fourth ed. s.l.:Addison-Wesley.

Renche W., T. Z. C. Y. H. Z. P. Y., 2013. Study on effect of spindle on dynamic performance and surface quality of whole machine tool. *Applied Mechanics and Materials,* Volume 312, pp. 11-15.

Rivin, E., 2000. Tooling Structure: Interface between cutting edge and machine tool. *Annals of the CIRP - Manufacturing Technology,* 49(2), pp. 591-634.

Rivin, E. I., 2002. Chapter 40: Machine tool Vibration.. In: *Harris' shock and vibration handbook.* s.l.:McGRAW-HILL.

rockwell automation, n.d. *Interpreting severity charts.* [Online]
Available at:
https://rockwellautomation.custhelp.com/ci/fattach/get/11741/1193838370/redirect/1
[Accessed 05 march 2015].

SAE, 2010. *Northern Ireland shop employs DST six-axis machines.* [Online]
Available at: http://articles.sae.org/7377/
[Accessed 07 December 2015].

Sandvik Coromant, 2015a. *Balancing and RPM.* [Online]
Available at: http://www.sandvik.coromant.com/en-gb/knowledge/tooling-systems/Machine-and-tooling-systems-considerations/balancing-and-rpm

Sandvik Coromant, 2015. *Spindle selection/requirements.* [Online]
Available at: http://www.sandvik.coromant.com/en-gb/knowledge/tooling-systems/machine-and-tooling-systems-considerations/spindle-selection/spindle-selection-requirements/pages/default.aspx
[Accessed 20 January 2015].

Scheffer, C. & Girdhar, P., 2004. *Practical Machinery Vibration Analysis and Predictive Maintenance.* First ed. s.l.:Elsevier.

Schulz, H. & T., W., 1998. Balancing Requirements for Fast Rotating Tools. *CIRP Annals - Manufacturing Technology,* Volume 47, pp. 321-342.

Shen, C., Wang, G. & Liu, G., 2011. The imbalance Source of spindle-Tool System and influence to Machine Vibration Characteristics. *Second International Coference on Digital Manufacturing & Automation.*

Shen, C., Wang, G. & Liu, G., 2011. The Imbalance Source of Spindle-Tool System and Influence to Machine Vibration Characteristics. *Second International Conference on Digital Manufacturing & Automation,* pp. 1288-1291.

Siemens, 2009. *Spindle Solutions: Spindle motors and motor spindle for every requirement.* [Online]
Available at: http://www.weissgmbh.com/uploads/media/Spindelloesungen_en_02.pdf

SIS, 2014. *SS 728000-1:2014.Evaluation of machine tool spindle vibration by measurements on spindle housing-part1: spindle with rolling elements bearing and integral drive operating at speeds between 600 min-1 and 30 0000 min-1* , s.l.: Swedish Standards Institute.

SKF, 2012. *Bearing investigation.* [Online]
Available at: http://www.skf.com/binary/86-62751/RTB-1-06-Bearing-investigation.pdf

SKF, 2013. *SKF, Rolling bearings cataloge.* [Online].

SKF, 2014. *Supe- precision bearings Catalogue.* [Online]
Available at: http://www.skf.com/binary/57-129877/Super-precision-bearings---13383_1-EN.pdf
[Accessed 05 10 2015].

Smith, A., 1993. *Reliability-centered Maintenance.* New York NY: McGraw-Hill.

Smith, S., 2011a. Maximum speed may be the worst speed. *Cutting Tool Engineering,* 63(5), pp. 26-28.

Smith, S., 2011b. Managing hybrid spindle bearings. *Cutting Tool Engineering Magazine,* 63(Issue 9).

Smith, S., 2011. *Maximun speed may be the worst speed Cutting tool Engineering.* [Online]
Available at: http://www.ctemag.com/aa_pages/2011/110509-MachineTech.html

Stadelbauer, D. G., 2002. Chapter 39. Balancing of rotating Machinery. In: *HARRIS's SHOCK AND VIBRATION HANDBOOK.* s.l.:McGraw-Hill.

Statistiska Central Byrå SCB, 2015. *VARUIMPORT-EXPORT, 731 SPÅNAVSKILJANDE VERTYGSMASKINER,* s.l.: s.n.

Stenmark, D., 2014. *Personal Comunication when attending SIS meeting 10/12/2014.* s.l.:s.n.

Swanson, E., Powell, C. D. & Weissman, S., n.d. A Practical Review of Rotating Machinery Critical Speeds.

TA, 2014. *ABOUT US: Information about Technical Associates of Charlotte.* [Online] Available at: http://www.technicalassociates.net/

TAC Rockford, 2015. *Drawbar campling forces.* [Online]
Available at: http://tacrockford.com/pdf/100.311/100311-typical-power-drawbar-clamping-forces.pdf
[Accessed 21 11 2015].

Taneja, S., 2012. The effect of Unbalance on Bearing life. *IOSR Juornal of Mechanical and Civil Engineering. Available at http://iosrjournals.org/iosr-jmce/papers/vol1-issue2/I0124754.pdf?id=2670*, July-Agust, I(2), pp. 47-54.

Tsang, A., 1995. Condition-based maintenance: tools and decision making. *Journal of Quality in Maintenance. Engineering,* 1(3), pp. 3-17.

Vafaei, S. & H., R., 20002. Vibration monitoring of high speed spindles using analysis spectral techniques. *International Journal of Machine Tools & Manufacture,* Volume 42, pp. 1223-234.

Verlag, B., 2000. *Economic benefits of Standarization,* s.l.: DIN German Institute for Standardization.

Vogl, G. W. & Donmez, M., 2015. A defect-driven diagnostic method for machine tool spindles. *CIRP Annals-Manufacturing Technology,* Volume 64, pp. 377-380.

Weck, M. & Koch, A., 1993. Spindle-Bearing systems for high speed apcations in machine tools.

Wester Wind, 2013. *PCB Spindles.* [Online]
Available at: http://www.westwind-airbearings.com/pcb/index.html
[Accessed 16 11 2015].

Wohler's Associates, 2015. Wohlers report 2015..

Zitney, E., 2011. *MACHINIST, AMERICAN.* [Online]
Available at: http://americanmachinist.com/qc-and-inspection/turning-health-machine-tool-spindles
[Accessed 08 October 2015].