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DESIGN AND MANUFACTURING OF A 70KW, 36000 RPM MILLING SPINDLE

Alberto Izpizua, Manex San Martin, David Cantero, Ion Perez
Mechatronics & Precision Engineering, Fundacion Tekniker-IK4, Eibar, Gipuzkoa, 20600, Spain
aizpizua@tekniker.es,msanmartin@tekniker.es, dcantero@tekniker.es, iperez@tekniker.es

Xanti Almandoz
Goialde High Speed, Zestoa, Guipuzkoa, 20740 Spain
xalmandoz@goialde.com

ABSTRACT
This work shows the development of magnetically levitated high speed spindles and the possibilities they give for the interaction with the milling process. A fully magnetically levitated spindle with 70 kW up to 36000 rpm is presented; its design and manufacturing issues and his performance in comparison with conventional high speed spindles. Furthermore the possibility to estimate cutting forces and parameters by an state-space approach, using the internal signals of the magnetic bearings as a whole adaptronic system, is shown; and even some preliminary attempts of damping process vibrations.

INTRODUCTION
Growing interest of industrial groups and research centers in the study and development of high speed machining processes has been generalized. Aluminium machining industry is requesting high speed and high power spindles for the machining of large-volume workpieces. Ceramic bearings are the weak point of high speed spindles. Their main drawback is that they are fragile, their life is limited, and the bearings replacement is a hard and expensive work in this kind of spindles. This is such a problem, that some aluminium manufacturers have spare high speed spindles to use as soon as one of the working spindles fails. In such way, that with a Total Cost of Ownership (TCO) strategy [1] the initial price of the spindle means only the 16% of the total cost for 24000 hours running.

This challenge has been mainly polarised towards the development of motors and rotating devices without mechanical contact between the rotating and static parts, controlling the rotors position by means of magnetic forces. Magnetic bearings introduce some advantages to this kind of spindles, where they provide superior value compared to other types of bearings [2]. Value is a function of the following: longer life, clean environment, extreme conditions, quieter operation, impact detection, and physical signal estimation (such as cutting forces and linear accelerations).

They have some drawbacks mainly that the spindle is bigger, since magnetic bearings need bigger volume for the same cutting force, and that the needed electric installation is also bigger. Products price could be another disadvantage for this kind of spindles, but as they are providing high reliability and long service intervals [3], and lower power consumption (due to frictionless bearings), the TCO should show better figures than ball bearing high speed spindles.

Tekniker-IK4 finished the manufacturing of its first magnetic bearing spindle for high speed machining at the beginning of 2004 [9]. This spindle has been machining aluminium from 17000 rpm to 32000 rpm, with a power up to 13.2 kW. Its weight is about 50 kg and its external diameter is 140 mm. It is able to withstand radial forces in the tool up to 1200 N and axial forces up to 400 N.

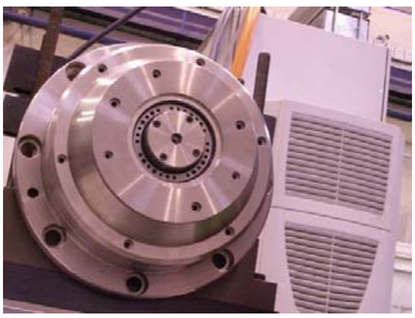

FIGURE 1: Magnetic bearing spindle and electrical cabinet.

This experience allows us to start a industrial magnetic bearing spindle design and building. Specifications for this magnetic spindle were taken by Tekniker-IK4, Goialde and end users TPA and MASA. These main specifications are the ability to machine with 70 kw at 36000 rpm. Its external diameter is 240 mm. It is able to withstand radial forces in the tool up to 3200 N force in radial direction and axial forces up to 2500 N. Because of its industrial feature some special issues are included in the design.

In this paper the design of the new spindle is presented and experiments done in the old one. This is presented in this fashion because, the big one is just finished when this article is writing. In this way, the "Cutting performance" section is evaluated in the old spindle, anyway all experiments and results are directly applicable to the new one. Actions to be taken due to results of those experiments are explained in the next section, "Passive damping by the control system". Those are now being developed for the new spindle, because is not possible to be implemented in the old one due to control electronics and sensors resolution.

MAGNETIC SPINDLE DEVELOPMENT

Aluminium aeronautics parts milling needs very high rotating speeds as well as high power. With those requirements Tekniker-IK4 took the challenge to start the design of a new spindle. This spindle should be equipped with a 36000 rpm (very high speed for machining aluminium) and 70 kw rated power motor

Spindle Design

Magnetic bearings have to manage all the force due to machining processes and they have to be able to maintain the rotor in its centre position and try to damp all undesired vibrations. According to aluminium machining characteristics, maximum speed advance per teeth and machine power, 3100 N force in tool is calculated. The magnetic force in bearings depends on this force and distance between magnetic bearings and distance to the tool. These forces should be added to the force generated by acceleration of the machine and force needed to damp undesired vibrations.

Position feedback is made by eddy current sensors which are placed appropriately out of any node of at least two first natural frequencies. Otherwise, it could not be possible to damp those bending frequencies and avoid chatter.

The selected motor is a commercial asynchronous high speed motor. This motor is placed in the shaft between rear magnetic bearing and axial bearing.

Two low cost auxiliary bearings have been placed. These bearings are not use in normal operation but they are included for security reasons, in order to prevent undesired touch between rotor and stator. Auxiliary bearings can be removed easily for maintenance purposes. Their replacement could be planned in a machine maintenance operation, it is not necessary to change them immediately when a problem is detected.

This spindle includes an automatic holder, with the ability of internal tool refrigeration. This is a need imposed by machine-tool users.

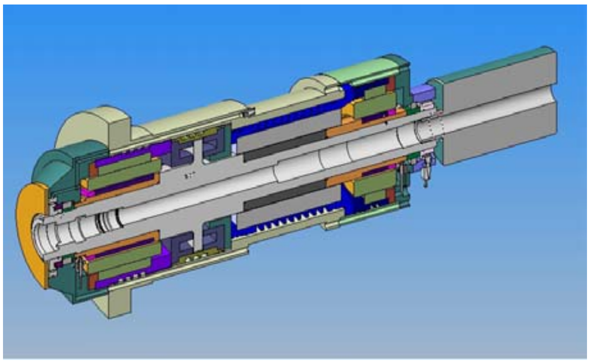

FIGURE 2: Magnetic bearing spindle scheme

Control Electronics

A hybrid control platform has been developed, which is composed of three main elements: FPGA, DSP, and a general purpose microcontroller. Combining these technologies, much better performance than conventional platforms is achieved in terms of calculation power, response time and number and quality of input/output channels.

Thus. the FPGA offers good interface skills and the capacity to process data at high rates in a parallel fashion. For that reason, the acquisition, pre and post processing of the signals is executed in this device. On the other hand, as the DSP presents a powerful core unit for working in floating point arithmetic at high speed, it executes the most complex control algorithms. Finally the microcontroller, due to its advanced communication capabilities, performs the user interface functions and monitors and supervises the status of the whole system.

Securities

As said before, there are low cost ball bearings included, but these are an insufficient security system in a machine tool, so that, at this moment three different types of securities are also implemented: thermal, impact and chatter.

Experience working in the first prototype has shown that auxiliary bearings do not work anytime. There are also some security algorithms in the control related to auxiliary bearings. For example, when there is an error in the program introduced in the milling machine, there is a chance to have an impact between the tool and the workpiece. When this situation is detected the machine is stopped immediately by the spindles control. The idea is to stop the spindle as soon as possible, in order to avoid any harmful damage in the levitation system or in the motor, but this is not the only one, its important to stop all the machine in order to avoid a damage without solution in the machined part. If the tool has a very small diameter, the impact could cause to break the tool. If the tool is bigger, the impact can cause the shaft to lose its central position. This kind of accident would probably cause a big failure in high speed spindles with ceramic bearings.

There are some temperature measurements that are made in some parts of the system. There are temperature sensors in bearing coils in order to avoid overtemperature damage in them. New spindle design includes non contact temperature measurements in different points of the shaft in order to evaluate thermal dissipation, produced mainly by the motor. Up to now, this security system has never stopped the spindle, although temperatures on some elements have caused some small modifications in the mechanical design of the system.

There is also implemented a real-time chatter monitoring. It is an algorithm that makes some calculations based on the signals from the sensors and principal chatter frequency. With this algorithm, it is possible for the machine to know the surface finishing that is being achieved. Spindles control sends a warning to the mills control when a magnitude of chatter vibration is reached. Currently, this magnitude could be changed manually, because there is no communication between these two controls. The best way to solve this inconvenience is to implement a communication between CNC and spindle control to fix a maximum chatter magnitude for finishing operations and an other one for roughing operations. The change between two states could be made easily analyzing force introduce by machining process in the bearings.

CUTTING PERFORMANCE

The damping capacity of a high speed spindle [4] determines its chatter-free performance even more drastically than its stiffness. Literature [5] shows cases of active magnetic bearings used for active damping or chatter suppression by means of appropriate control algorithms. However bandwidth of the magnetic bearings prevents active control of vibration in high speed applications [6]. Other successful examples integrate an additional magnetic bearing [7] over a ball bearing spindle for identification and vibration control purposes. However, the final goal of active bearings is to avoid the use of ball bearings, which are limiting the life of the whole spindle, and adding the extra overhang required, seems not to be a practical solution for industrial use.

Dynamic response

Dynamic response of the magnetic spindle has been obtained by means of experimental impact test. Both signals, excitation and response have been measured at the tool tip. A dummy tool of Ø 16 mm has been used. Same tests have been done at a ball bearing spindle of similar size and power. Next paragraphs and figures show the obtained response and the corresponding dynamic parameters for each case.

The magnetic spindle values (FIGURE 3) are a natural frequency of ω=554 Hz, damping factor of ξ=0.32 %, transfer function maximum magnitude H=164.8 $m/s^2/N$, and the corresponding reduced mass value is 0.948 kg. This response has been achieved with a position loop bandwidth of 200 Hz.

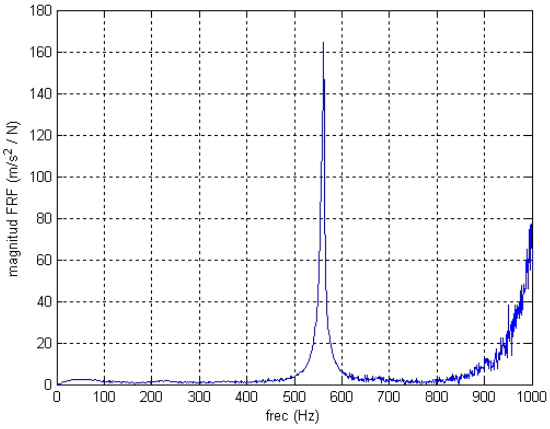

FIGURE 3. Magnetic spindle frequency response

The ball bearing spindle values (FIGURE 4) are a natural frequency of ω=540 Hz, damping factor of ξ=1.6 %, transfer function maximum magnitude H=49.2 $m/s^2/N$, and the corresponding reduced mass M=0.635 kg. Main difference is the damping factor which is 5 times smaller in the magnetic spindle case.

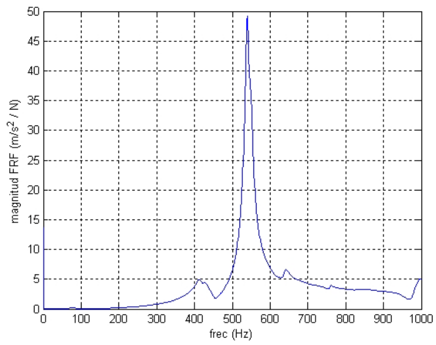

FIGURE 4. Ball bearing spindle frequency response

Lobe diagram

Lobe diagram in frequency domain has been obtained for both spindles. Simulation conditions have been a two flutes straight end mill, Ø 16 mm and helix angle 30º. Aluminum cutting coefficients used are Kt=804 MPa, Kr=331 MPa, Ka=-199 MPa. Radial immersion is 1 mm. Some experimental test results are also shown.

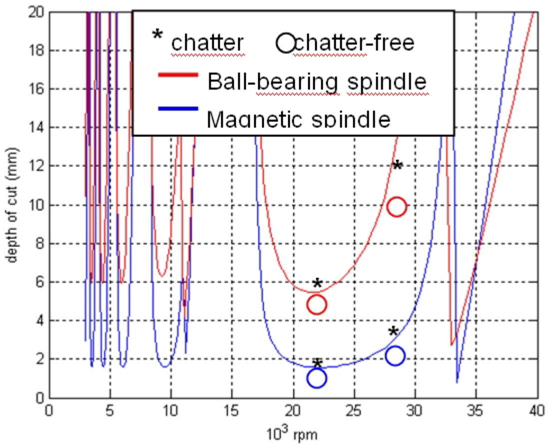

FIGURE 5. Lobe diagram comparison

The first and third order "flip lobes" can be noticed clearly. No helix effect has been considered due to the low axial depth of cut. Main conclusion is that the chatter-free limit depth of cut is around 6 mm (excluding "flip" effects) for the ball bearing spindle and 2 mm for the magnetic bearing spindle. Chatter frequency varies between 540 Hz to 650 Hz.

PASSIVE DAMPING BY THE CONTROL SYSTEM

Applying a force proportional and with opposite sign to the velocity, the control system is able to provide high damping values to mechanical systems. A velocity loop bandwidth of up to %80 of the lowest deformation natural frequency of the mechanical system must be reached to achieve this damping effect. On the other hand, for that effect to be effective, it is necessary to measure the velocity at the same position that the force will be applied (collocated system).

The first problem to achieve that bandwidth is the noise content at the velocity signal. This signal is obtained by differentiation of the position signal, and any noise at it is amplified by the differentiation process.

The second problem is that in the magnetic bearings, collocation is not fully obtained. Position sensors are located at the sides of the coils, and in most of the cases a nodal point appears between sensor and coil for the lowest deformation natural frequencies. Thus, for these frequencies collocation is not obtained, and as a consequence, the control system is unable to provide a noticeable amount of damping to the modes.

Possible solutions for these problems are:

- Using a Kalman filter, based on state space observers, for filtering out the noise at the velocity signal.
- Using a sensorless estimation of velocity for achieving a collocated system
- Using an state space observer for extrapolating the position measured at the sensors to the position and velocities at the centre of the coils.

Filtering of the noise at the velocity signals

As said above, velocity signals obtained by differentiation of the position signals will have a high noise level. This noise will impede raising the velocity loop gain to the figures required to get a good damping ratio at the main natural frequencies.

A Kalman filter based on a simple state space observer can be applied for observing 'noise free' velocity and position signals. The mechanical model of the system can be as simple as a single degree of freedom model (sdof) with a natural frequency equal to the lowest deformation rotor frequency. This model will include as inputs the force at the coils, obtained from the current supplied to it, and a cutting force function dependent on the angle of rotation of the spindle, with parameters that will have to be identified in real time.

Identification of the parameters that define the cutting force as a function of the rotation of the spindle will be the most difficult task in this filter. Without those parameters, the filtered velocity will have some delay with respect the actual velocity, which might be a limiting factor for the applicability of the technique.

Sensorless estimation

There are some techniques to obtain an approach to the air gap at the bearings. These techniques are based on introducing a high frequency signal into the coils and obtaining an estimation of the impedance by comparing voltage and current. The air gap is deducted from the estimated value for the impedance.

This method will provide a collocated estimation of the position and velocity at the bearings, but it will have a high level of noise, unless filtering techniques are applied.

State-space observer

The dynamics of the observer is defined by the state space approach for a discrete and linear system.

$$\begin{aligned} x(k+1) &= G(k)\cdot x(k) + H(k)\cdot u(k) \\ y(k) &= C(k)\cdot x(k) + D(k)\cdot u(k) \end{aligned} \qquad (1)$$

Where ***x(k)*** is the state vector of n-dimension; ***y(k)*** the output vector of m-dimension; ***u(k)*** the input vector of r-dimension; ***G(k)*** the state matrix of n x n-dimension; ***H(k)*** the input matrix of n x r-dimension; ***C(k)*** the output matrix of m x n-dimension; and ***D(k)*** the direct transmission matrix of m x r-dimension.

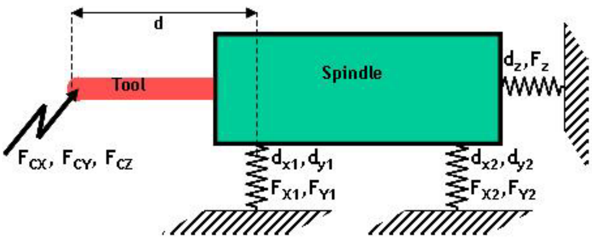

FIGURE 6. Model for the state-space Observer

Our state vector is composed by the displacement and speed at bearings (*dx1*, *dy1*, *dx2*, *dy2*, *dz*, *vx1*, *vy1*, *vx2*, *vy2*, *vz*); the natural frequency and modal displacements (*ωn*, *ψtool*, *ψ1*, *ψ2*); the axial distance from the tool tip to the first radial bearing (*d*); the cutting force coefficients multiplied by axial depth of cut (*b·kt*, *b·kr*, *b·ka*); the number of teeth (*z*) and the cutting arc (*φ1*, *φ2*). These two latest (*z*, *φ1*, *φ2*) can be introduced by the user.

The state-space observer allows identifying system variables. As example of the identification, next figure shows the displacements measured and estimated at the tool tip.

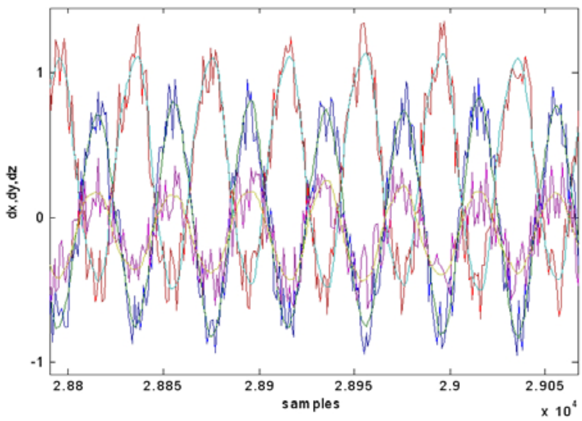

FIGURE 7. Measured and estimated displacements

CONCLUSIONS AND FURTHER WORK

Some of the huge capabilities of magnetic bearing spindles for high speed machining applications if damping factor is improved have been introduced. They are a promising alternative to high speed ball bearing spindles as they can rotate faster with no wear.

They have some inherent properties that will allow them to become in an adaptronic system, able to change its performance according to different machining needs. Although these spindles are already a mature product in other fields of application, the lack of damping prevents their industrial application for high speed machining.

The poor performance of magnetic spindles in relation with ball bearing spindles from the chatter behaviour point of view has been verified theoretically and experimentally.

Three different solutions (Kalman filter, sensorless estimation of velocity, and state space observer) are foreseen to overcome that limitation.

First estimations give achievable values of the damping factor ξ between 10 to 20 %, which means a fully chatter-free spindle from a practical point of view as is shown in FIGURE 8.

Next step is to implement those solutions in the existing new high speed magnetic spindle (70 kW at 36000 rpm).

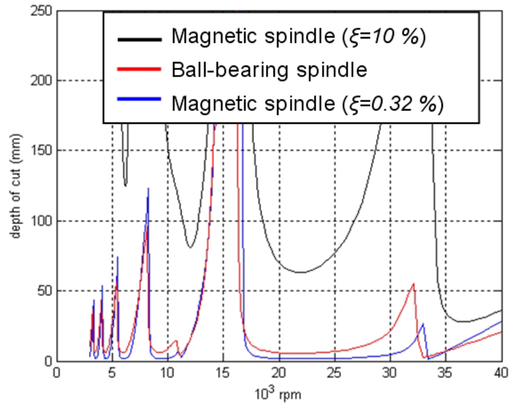

FIGURE 8. Magnetic spindle lobe diagram (ξ=10 %)

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