Mechatronic Spindle Head for Chatter Suppression in Heavy Duty Operations

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MECHATRONIC SPINDLE HEAD FOR CHATTER SUPPRESSION IN HEAVY DUTY OPERATIONS

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ABSTRACT

A new concept of spindle head for heavy roughing operations has been developed integrating an inertial actuator and an accelerometers in a heavy duty spindle head. With this head is possible to introduce active damping using the inertial drive, and improve the dynamic stability of the machine. A novel mechatronic milling model is employed for designing the actuator, where the effect of cutting process and inertial actuators can be simulated. The validation tests of the new actuator are shown and all mechatronic system is completely integrated on the machine. Finally, experimental cutting tests are performed, showing that material removal rate is doubled with this new spindle head.

Keywords: Stability, Milling, Chatter, Active Control.

1. INTRODUCTION

Actually, self-excited vibrations known as chatter are one of the most important restrictions in cutting processes. These vibrations are principally due to the regenerative effect [Tlusty and

Polacek, 1957; Tobias and Fishwick, 1958] and are the cause of unacceptable surface finishes, tool breakages and timelife reduction of different mechanic elements.

Probably one of the most employed solutions for increasing the stability is to add a tuned mass damper to the structure [Sims et al., 2005; Yang et al., 2010]. This is not a novel technique and it has been widely employed [Den Hartog, 1956]. However, the use of passive dampers is not feasible in many cutting processes where the dynamics of the system may vary during the process.

Active dampers can overcome these limitations by means of their adaptability to variable conditions. These actuators are based on performing a reaction force onto the main structure, controlled by a control algorithm.

The development of active dampers had its origin in the aeronautical industry. In 1968 Bies and Yang reported that the problems presented by passive absorbers could be solved by means of adding an active element. In 1970, Cowley and Boyle proposed to use an electromagnetic inertial actuator with an accelerometer in order to introduce active damping onto the structure of a machine tool.

The combination of active devices and sensors makes up an advanced mechatronic solution, whose behaviour is comparable to a smart material, such as piezoelectric actuators. In fact, inertial actuators have better performance for improving the dynamic behaviour of machine tool in heavy duty operations [Ehmann and Nordmann, 2002].

Piezoelectric actuators are usually located inside the machine in serial way due to their characteristics. It means that they are inside the force path of the machine, and therefore a big stiffness is required to transmit cutting forces. However, inertial actuators tend to locate in parallel way maintaining the original stiffness of the machine [Ehmann and Nordmann, 2002; Loix and Verschueren, 2004]. During the last years, the introduction of inertial actuators in heavy duty machines has been studied. For instance, Ehmann and Nordmann (2002) recommended the introduction of inertial actuators for big dimension machines.

In order to build an inertial actuator, some authors have employed hydraulic technology. Brecher et al. (2005) developed an electrohydraulic actuator for testing different control strategies in chatter problems. The hydraulic technology is interesting because it offers high forces and strokes. However it exhibit big disadvantages such as maintenance problems, non-linearities or delays.

From an energetic point of view, the piezo actuators merely need power to change their state. In contrast, the electromagnetic actuator need to continuously overcome resistor losses. Nevertheless, the cost for piezoelectric systems exceeds those of the electromagnetic actuator systems roughly by a factor of 3-4. Moreover, the electromagnetic actuators present another advantage of not having physical contact between the moving and fixed parts, so they are widely used in vibration applications.

Among electromagnetic devices, reluctance or attraction force actuators offer large forces, but this force is proportional to the current squared and inversely proportional to the air gap squared. Hence, they are difficult to control and are not usually employed in precision applications. Lorentz force actuators can produce less force, but their linearity makes them a good option for vibration control. In this case, the force is proportional to the strength of the magnetic field, and if a proper design with permanent magnets is selected, the force is strictly proportional to the applied current. Several authors have applied inertial electromagnetic actuators based on Lorentz forces for avoiding chatter on machine tools [Loix and Verschueren, 2004, Bilbao-Guillerna et al., 2012].

In recent years, actuators where both reluctance and Lorentz forces are combined have been developed and patented [Claeyssen et al., 2008]. These actuators, known as MICA, are reported as devices that offer higher forces but require a very accurate guiding system.

This work presents the development of a new spindle head, where an electromagnetic inertial actuator is integrated for active control of vibrations. The active devices have to be located close to the cutting tool, where machine tools offer reduced space for them. In this case, the design of a new spindle head provides the possibility of introducing a small inertial actuator inside it. The paper shows the design of the actuator by means of a mechatronic model and its validation test. Finally, the experimental cutting tests show the great improvement obtained by this new spindle head.

2. DYNAMIC CHARACTERISTICS OF A RAM TYPE TRAVELLING COLUMN MILLING MACHINE

Chatter vibrations appear in many different ways in the milling process. The prediction and suppression techniques vary depending on the process [Munoa et al., 2005]. For example, in high speed aluminium rough milling, the modes limiting the stability are associated to the tool and the toolholder, or to the spindle (the chatter frequency is roughly between 300 and 3000 Hz) [Mancisidor et al., 2011]. However in steel roughing the critical modes are related to the whole machine tool structure (the chatter frequency is roughly between 15 and 100 Hz).

The ram type machine tools have some special characteristics defined mostly by the ram. Basically, the ram is a special cantilever beam with a big mass in form of a spindle in its tip. The ram can change its overhang using one of the drives of the machine. This machine concept introduces flexibility in two axes (X and Y in our case) related to bending directions and maintains high stiffness in the axial direction (Z in our case). The overhang of the ram varies depending on the working position and this produces big variations in the dynamic stiffness of the machine.

Figure 1. Ram type Moving Column Milling Machine

The overhang of the ram has a capital influence, because the bending and the rotation of the ram are directly involved in the critical modes. These modes change their properties inside the workspace, produce displacements in the two bending planes and can have similar frequencies. Generally speaking, chatter is always generated by modes with important displacements in the tool tip.

In the present study a prototype of a SORALUCE milling machine (see Figure 1) has been selected for integrating the new active spindle head. First of all, a complete dynamic study was carried out including a complete modal analysis (see Table 1).

Considering the dynamical characteristics of ram type machine tools, the optimum damper should have an active behavior to adapt to dynamical variations inside the workspace and it should be placed close to the tool tip. In this work, it will be located on the top of the spindle head (point *q*), and the original stiffness of the machine will be maintained since it will be located in a parallel way as a suspended mass.

Table 1 Results of the modal analysis of a ram type milling machine (ram overhang=1000mm)

	Natural Frequency (ω_n)	Damping ratio (ξ)	Modal Stiffness (k)	Modal vector [Q_i]	
				Tool (p)	Spindle Head (q)
1	33.6Hz	5.5%	27.9 N/µm	(-0.70, 0.61, -0.38)	(-0.59, 0.54, -0.22)
2	51.2Hz	2%	145 N/µm	(-0.99, 0.05, 0.06)	(-0.80, 0.06, 0.11)
3	193Hz	1.5%	639 N/µm	(0.70,-0.56, 0.43)	(0.2, 0.04, 0.06)

3. MILLING MECHATRONIC MODEL

In the state of art, the need of design guidelines of inertial actuators is reflected, in order to assure certain stability to a certain machine. Some authors analyzed the integration of active devices in machine tools by means of simulations [Zaeh et al., 2009; Sajedipour et al., 2010]. However, no one verified their results experimentally and the different control parameters and limitations of actuators were not taken into consideration. Therefore, a mechatronic model has been developed in order to provide guidelines in the design of inertial actuators.

This model simulates the behaviour of a milling machine in time domain considering the cutting process and the behaviour of the actuator at the same time. This way, the stability of the system can be studied. The model provides the option of introducing different limitations of the actuator. The most important limitation is usually the maximum force of the inertial device.

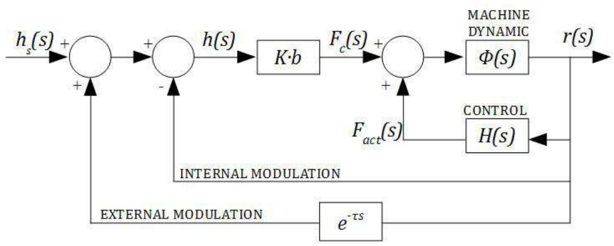

Figure 2. Closed loops taken into account in the mechatronic model.

3.1. Mathematical formulation

The simulation is characterized by two closed loop systems (Figure 2). Normally, it is not possible to measure and act directly in the tool. Therefore, different points and their relation have to be taken into account (see points *p* and *q* in Figure 1 and Table 1).

The first loop calculates the regenerative cutting force F_c on the tool tip (point p) following the formulation proposed by Altintas (2012). The tangential (F_t), radial (F_r) and axial (F_a) forces for each tooth j are calculated by the next equation:

$$\begin{Bmatrix} F_{t,j} \\ F_{r,j} \\ F_{a,j} \end{Bmatrix} = \left\{ K_{tc} \begin{bmatrix} 1 \\ K_{rc} \\ K_{ac} \end{bmatrix} h(\phi_j, \kappa) b + \begin{bmatrix} K_{te} \\ K_{re} \\ K_{ae} \end{bmatrix} S \right\} g(\phi_j), \tag{1}$$

where K_{tc}, K_{rc} and K_{ac} are cutting coefficients, and K_{te}, K_{re} and K_{ae} are the edge coefficients. b is the depth of cut of the tool and S is referred to the length of the flute. The immersion angle ϕ_j defines if the tool j is cutting and is defined as

$$g(\phi_j) = \begin{cases} 1 & \phi_s < \phi_j < \phi_e \\ 0 & \phi_j > \phi_e \quad and \quad \phi_j < \phi_s \end{cases} \tag{2}$$

where ϕ_s and ϕ_e are the cutter entry and exit angles, respectively.

The chip thickness (h), consists of a static part (h_s), which depends on the feed per teeth f_z, and a dynamic component (h_d), affected by the vibration (r) on the point p:

$$\begin{aligned} & h(\phi_j, \kappa) = h_s(\phi_j, \kappa) + h_d(\phi_j, \kappa) \\ & h_s(\phi_j, \kappa) = f_z \sin(\phi_j) \sin\kappa \\ & h_d(\phi_j, \kappa) = \{\sin(\phi_j)\sin\kappa \quad \cos(\phi_j)\sin\kappa \quad -\cos\kappa\}\{\Delta r(t)\}_p \end{aligned} \tag{3}$$

where $\Delta r(t)=r(t)-r(t-\tau)$, being τ the tooth passing period and κ is the lead angle.

Finally, adding the cutting forces contributed by all teeth, where Z is the number of teeth and projecting onto the xyz Cartesian axes,

$$\{F_c(t)\}_p = \begin{Bmatrix} F_x(t) \\ F_y(t) \\ F_z(t) \end{Bmatrix}_p = \sum_{j=0}^{Z-1} \begin{bmatrix} -\cos(\phi_j) & -\sin(\phi_j) & 0 \\ \sin(\phi_j) & -\cos(\phi_j) & 0 \\ 0 & 0 & 1 \end{bmatrix} \begin{Bmatrix} F_{t,j} \\ F_{r,j} \\ F_{a,j} \end{Bmatrix}_p \tag{4}$$

On the other hand, the second loop is due to the active control of the inertial actuator. In this way, the actuator force $\{F_{act}\}_q$ is calculated depending on the chosen control law and the actuator location q.

The behaviour of the system is described by a matrix equation defined considering the Cartesian coordinates of both points p and q, where [M], [C] and [K] are the mass, damping and stiffness matrices respectively.

$$[M]\{\ddot{r}(t)\}+[C]\{\dot{r}(t)\}+[K]\{r(t)\}=\{F(t)\} \tag{5}$$

where,

$$\{r(t)\}=\begin{Bmatrix}\{r(t)\}_p\\ \{r(t)\}_q\\ \{r(t)\}_m\end{Bmatrix} \text{ and } \{F(t)\}=\begin{Bmatrix}\{F_{\mathrm{c}}(t)\}\\ \{F_{\mathrm{act}}(t)\}\\ 0\end{Bmatrix}.$$

A modal transformation is performed using a modal vector matrix [Q]. Each column of this matrix coincides with the modal displacement η of each mode in points p and q, where n modes are considered. Therefore:

$$[Q]^{\mathrm{T}}[M][Q]\{\ddot{\eta}(t)\}+[Q]^{\mathrm{T}}[C][Q]\{\dot{\eta}(t)\}+[Q]^{\mathrm{T}}[K][Q]\{\eta(t)\}=[Q]^{\mathrm{T}}\{F(t)\} \tag{6}$$

Considering proportional damping the matrix equation is diagonalised and decoupled:

$$\ddot{\eta}_i(t)+2\xi_i\omega_{\mathrm{n},i}\dot{\eta}_i(t)+\omega_{\mathrm{n},i}^2\eta_i(t)=\frac{P_i(t)}{m_i} \tag{7}$$

where m_i, $\omega_{\mathrm{n},i}$ and ξ_i are the modal mass, the natural frequency and the damping of the ith mode of the structure respectively. P_i is the modal force which is calculated by the sum of modal milling force and modal actuator force:

$$P_i(t)=\left[Q_p^i\right]^{\mathrm{T}}\{F_{\mathrm{c}}(t)\}_p+\left[Q_q^i\right]^{\mathrm{T}}\{F_{\mathrm{act}}(t)\}_q \tag{8}$$

Modal displacements are calculated using the numerical integration proposed in [Bediaga et al., 2009], based on state space model:

$$\begin{Bmatrix}\dot{\eta}_i(t)\\ \ddot{\eta}_i(t)\end{Bmatrix}=\begin{bmatrix}0 & 1\\ -\omega_{\mathrm{n},i}^2 & -2\xi_i\omega_{\mathrm{n},i}\end{bmatrix}\begin{Bmatrix}\eta_i(t)\\ \dot{\eta}_i(t)\end{Bmatrix}+\begin{bmatrix}0\\ 1\end{bmatrix}\frac{P_i(t)}{m_i} \tag{9}$$

$$\begin{Bmatrix}\eta_i(t_{k+1})\\ \dot{\eta}_i(t_{k+1})\end{Bmatrix}=\begin{bmatrix}A_{11,i} & A_{12,i}\\ A_{21,i} & A_{22,i}\end{bmatrix}\begin{Bmatrix}\eta_i(t_k)\\ \dot{\eta}_i(t_k)\end{Bmatrix}+\begin{bmatrix}\dfrac{\Delta t^2}{2}\\ \Delta t-\xi_i\omega_{\mathrm{n},i}\Delta t^2\end{bmatrix}\frac{P_i(t_k)}{m_i} \tag{10}$$

Then, the Cartesian displacements of each point are calculated by modal vectors and they will be used in the calculation of the next forces.

$$\{r(t_{k+1})\} = \begin{Bmatrix} \{r(t_{k+1})\}_p \\ \{r(t_{k+1})\}_q \end{Bmatrix} = [Q]^{\mathrm{T}} \{\eta(t_{k+1})\} \tag{11}$$

3.2. Control law

Several control laws for suppression of chatter vibrations have been presented in the literature [Brecher and Schulz, 2004; Huyanan and Sims, 2007; Bilbao-Guillerna et al., 2010]. In this work the direct velocity feedback (DVF) strategy is employed, since it is the most common law for this application. It is based on the measurement of vibration velocity and its negative feedback is multiplied by a gain *G*. In this way, the control forces appear as viscous damping and the stability lobes increase their limit depth of cut [Munoa et al., 2013].

$$\{F_{\mathrm{act}}(t)\}_q = G \cdot \{\dot{r}(t)\}_q \tag{12}$$

In this way, DVF can actuate over all vibrations measured and thus it does not need any model of the system. So, such technique is widely used for its simplicity and leads to very good results. However, when a DVF is used, a high-pass filter should be added in the control loop in order to minimize the effects of low frequency dynamics of the actuator. In the study presented by this paper, a low-pass filter has been also added in order to neglect the high frequency noise. In addition, the closed-loop can easily become unstable for sufficiently large values of the feedback gain. For this reason, sometimes it is necessary to introduce more filters in order not to consuming significant energy trying to reduce the whole frequency range including regions with no contribution to chatter. This work proposes the utilization of a notch filter to avoid the spindle tooth passing frequency, when it is located far from the frequencies of interest. Finally, the force is limited by a saturation block, since actuators cannot perform more than a certain force.

3.3. Mechatronic model results

In this work, the mechatronic model has been used in order to design the electromagnetic inertial actuator for the proposed milling machine. In the design of the new spindle head, a space for the inertial actuator was provided on the top of the head. Taking into account the available space, different possible designs has been simulated over the machine dynamics.

As demonstrated by Munoa et al. (2013), a biaxial actuator is very efficient in ram type travelling column machines in order to act in the two flexible directions of the ram. However, the reduced space of the spindle head makes the design of different alternatives necessary.

After the first design of the electromagnetic part by means of a magnetic finite element model (Flux), two alternatives were considered for the same space. On the one hand, a biaxial actuator was calculated, which was capable of performing a maximum of 350N in each direction. On the other hand, a single axis actuator was designed for the same space. Since the space could be optimized better, a capability of 1000N was calculated for this actuator.

Using the modal parameters defined in Table 1, the stability results of each actuator were obtained (Figure 3). It can be observed that better results can be obtained by a 1000N single axis actuator around the minimum stability zone, although the biaxial actuator also improves the stability considerably. Therefore, in this case a higher force capability is preferred than a biaxial performance.

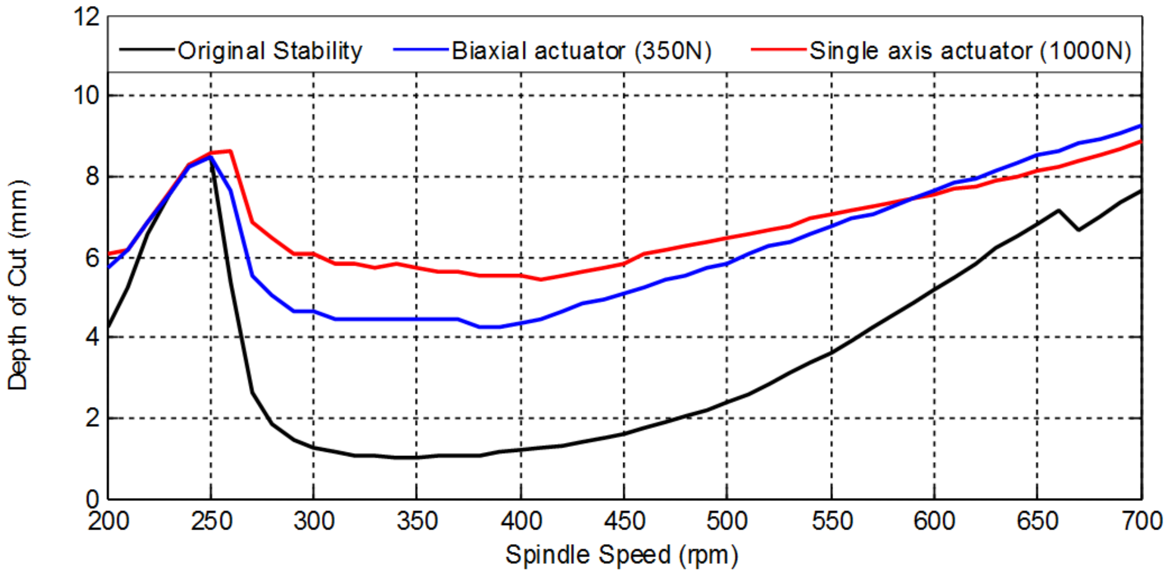

Figure 3. The comparison of the stability improvement by the two proposed inertial actuators.

4. DEVELOPMENT OF THE NEW SPINDLE HEAD

4.1. Introduction

The main objective is to develop an active spindle head with an integrated accelerometer and an electromagnetic actuator. The new device will be able to provide active damping to an ordinary milling machine.

SORALUCE has selected the H210 spindle head to be the base of the brand new spindle head with integrated sensors and actuator. It is a Huré type universal head and it is one of the most widely used spindle head.

The idea has been to design a spindle head able to provide a power of 32kW and a torque of 1200Nm. The maximum speed has been defined around 5000 rpm.

A new inertial actuator with 1000N has to be integrated inside the spindle head without any additional limitation or collision in the travel of the machine. This means that the actuator cannot be higher than 120mm.

4.2. Design of the inertial actuator

Once the single axis actuator is chosen, the challenge is to design in detail and develop the actuator capable of providing 1000N in the available space. The electromagnetic design is based on an alternative of linear motor with limited stroke. A sinusoidal current with the desired frequency is introduced in the static coils. In this way, due to Lorentz force, a linear displacement force is performed on the moving mass where neodymium permanent magnets are located. The magnetic flux density of the permanent magnets is 1.2 T, while each coil consists of 380 turns where a maximum of 3 A current is introduced.

The optimization of the space has been obtained by locating four parallel rows of permanent magnets and coils, which have been sized by magnetic finite element software (Flux), calculating the theoretical force obtained with the design. Figure 4 shows the finite element model and the magnetic flux obtained.

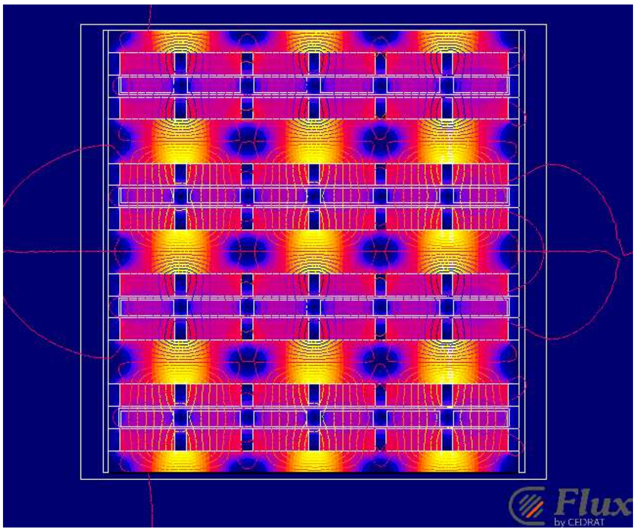

Figure 4. Magnetic finite element model built by Flux.

A simple flexure has been designed for the guiding system to endure the magnetic forces. In order to dimensioning this flexure, two simulations have been performed by finite element method (NX I-deas). On one hand, a fatigue simulation has been carried out to check the reliability of the flexure. The flexure has to be sufficiently stiff to assure that the displacement of the moving part is not too high. On the other hand, the dynamic behaviour of the flexure has been analysed. In this type of applications, the suspension mode of the actuator may occur at low frequency for not affecting the operating range of the active control. Therefore, this objective requires having a low stiffness without fatigue problems. Another option is to increase the moving mass, but it would require more space. Finally, the final design of the double flexure will be capable of enduring theoretically the forces and will have 17.5Hz suspension mode considering a 20kg moving mass.

Once the device is developed, the energy has to be provided to the actuator to generate sinusoidal current signals and obtain the required force. In this case the command voltage is converted onto current by ELMO TUBA servo amplifier. The digital controller is a National Instruments CompactRIO with a 50 kHz sample frequency.

4.3. Validation of the inertial actuator

The validation of the manufactured actuator (Figure 5) has been performed by 3 tests. The first test analyses the dynamic behaviour of the actuator and the linearity that it offers according to the force level. The second test studies the maximum force level that the actuator does. After these two tests, the thermal behaviour of the actuator is tested.

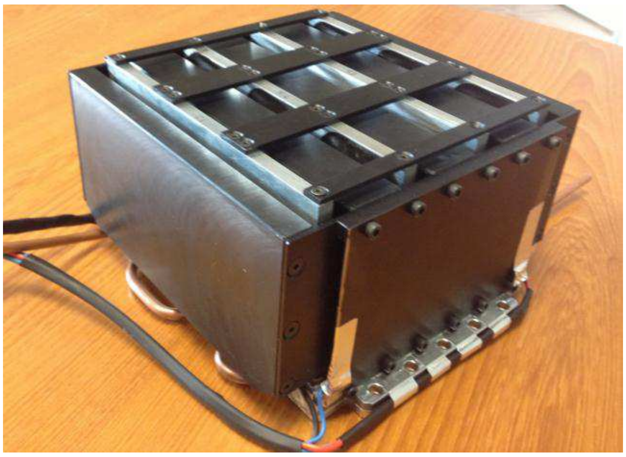

Figure 5. Electromagnetic actuator developed for the new spindle head.

In order to analyse the ratio between input voltage and force obtained, a chirp excitation signal with constant amplitude is commanded to the servo amplifier and force is measured by means of a dynamometric plate. In this way the frequency response function (FRF) between the generated force and applied voltage input is obtained (Figure 6). The suspension frequency is located at 17.1 Hz, as predicted by the finite element method, and it has a 3.51

% damping ratio. On the other hand, the actuator shows a linear behaviour, since the relation N/V is maintained in a wide frequency band.

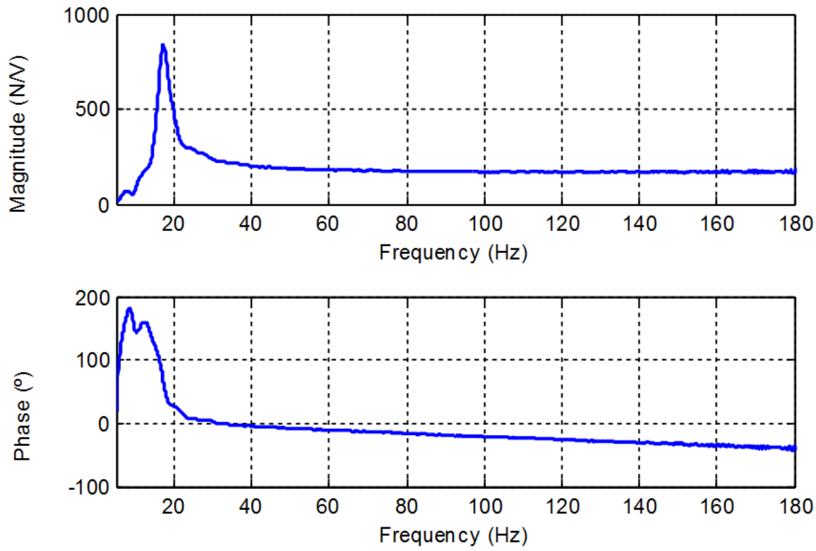

Figure 6. Force/Voltage ratio.

The force done by the actuator with different input amplitudes has been measured. A sinusoidal excitation has been introduced and as well as the previous test, the force has been measured by the dynamometric plate. In these tests, three frequencies close to the typical chatter frequencies have been selected.

Input voltage	Measured force 30 Hz	Measured force 60 Hz	Measured force 80 Hz
1 V	260 N	184 N	176 N
2 V	510 N	365 N	350 N
3 V	750 N	540 N	520 N
4 V	985 N	725 N	710 N
5 V	1200 N	900 N	875 N
6 V	1420 N	1070 N	1040 N

Table 2: Measured force for different input voltage and frequencies.

As shown in table 2, the actuator is capable of performing 1000N. Moreover, in some frequencies this capability is considerably increased due to the proximity of the suspension mode.

Finally, the evolution of the temperature inside the actuator has been analysed. The thermal test is a fundamental test for validating the actuator. When the current is introduced into the coils, they warm up due to their resistance. If this heating were very high, the permanent magnets could lose their properties and the force level would reduce considerably. In this actuator the maximum limit temperature has been established at 100ºC, introducing a safety margin before the apparition of commented possible problems.

In the tests, the actuator has been excited with sinusoidal signals of 2 different amplitudes and the temperature evolution has been measured by temperature sensors. The frequency of the signals is set to 60Hz.

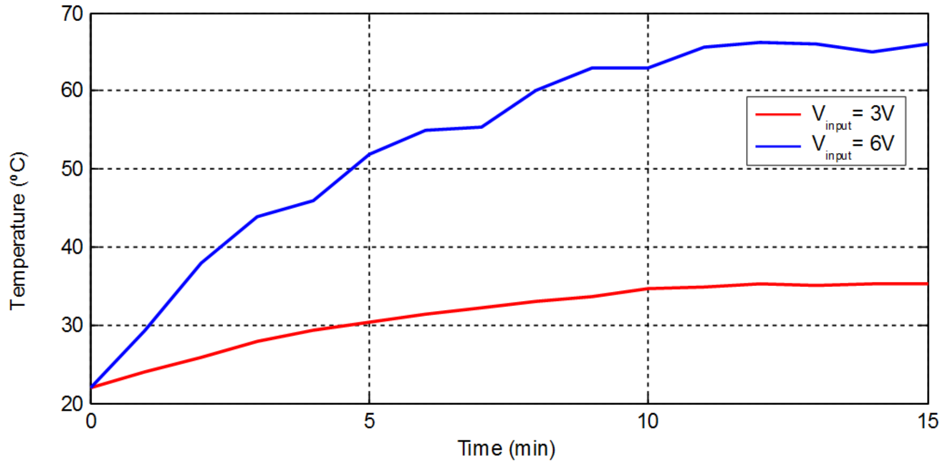

Figure 7. Thermal evolution of the actuator.

Figure 7 shows that the temperature is stabilized within the first 15 minutes and they are far from the limit. However, a water cooling system has been introduced to prevent the possible heating problems and thermic distortions. This system works as a thermic barrier between the actuator and the spindle head.

4.4. Final design

SORALUCE has experience introducing small electric motors in this spindle head. The idea was to redesign basically one of the parts to have more space to introduce the actuator and use the connections of the H210 spindle head to provide the required power to the inertial actuator and send the signal of the accelerometer to the control cabinet of the machine.

The controller (NI CompactRIO) and the power electronics (ELMO Tuba) have been integrated in the electronic cabinet of the machine and all cables have been connected.

A new part was designed and manufactured in cast iron to place the new actuator and the accelerometer. Finally, the new spindle head was assembled and mounted in a SORALUCE FL6000 milling machine (Figure 8).

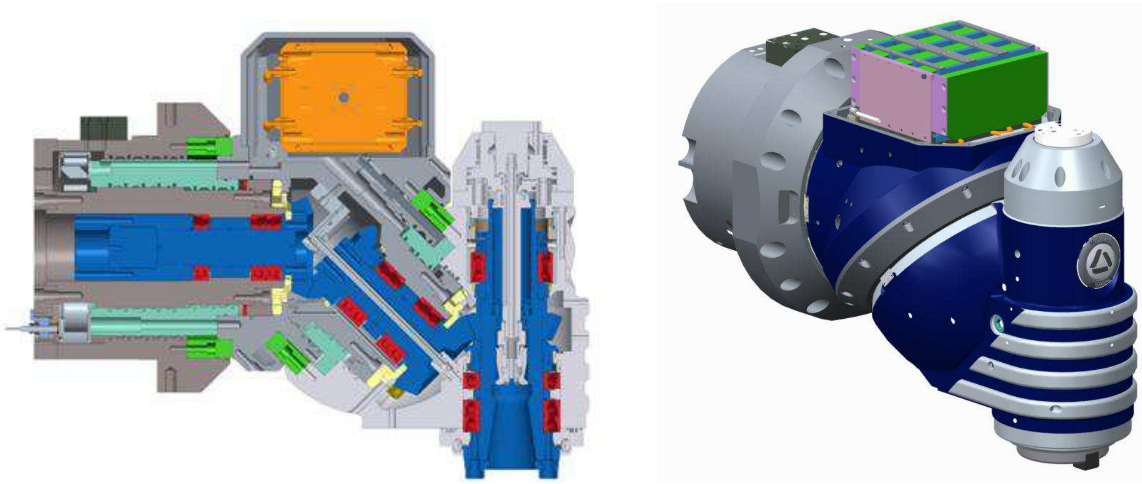

Figure 8. Integration of the inertial actuator over the new spindle speed.

5. EXPERIMENTAL RESULTS

Finally, the stability of the machine with the new spindle speed has been verified by experimental cutting tests. Table 3 describes the cutting tool, the cutting conditions and the cutting coefficients employed in these tests. In this way, different ram overhangs have been studied and the improvement due to the integration of the inertial actuator is analysed in Figure 9.

Cutting tool	
Reference	SANDVIK R245-125Q40-12M
Diameter	125 mm
Number of teeth Z	8
Inserts reference	SANDVIK R245-12 T3 M-PM 4030
Lead angle κ	45°
Cutting conditions	
Spindle Speed	440 rpm
Feed per tooth	0.15 mm/z
Radial engagement	100 mm (downmilling)
Workpiece material	F1140 Steel (C45)
Cutting coefficients	
Specific tangential Ktc	1883 N/mm
Specific radial Krc	0.38
Specific axial Kac	-0.25

Table 3. Description of cutting tool, cutting conditions and cutting coefficients.

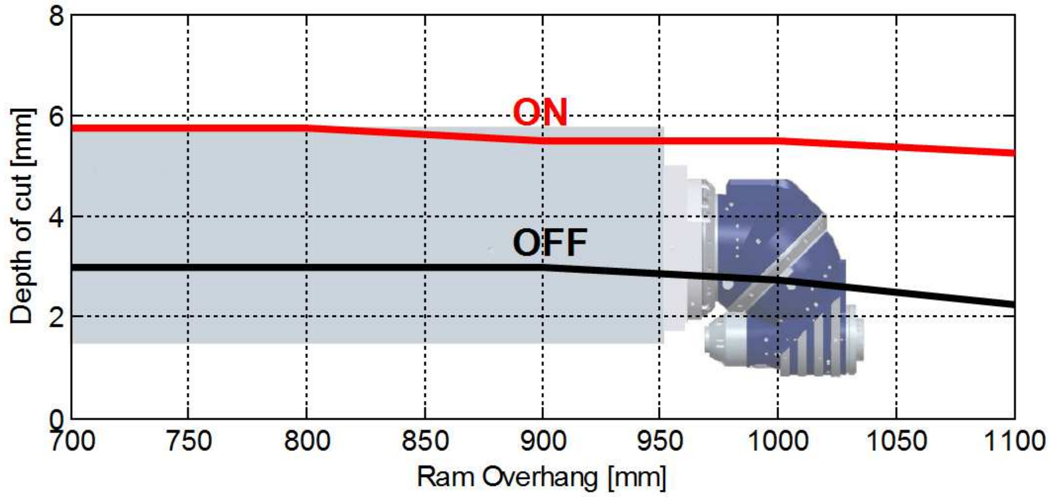

Figure 9: Stability improvement obtained by the new active spindle head.

The results show that the new spindle head with the active damper can double the cutting capability of the machine. The 1000 N actuator is capable of reducing the cutting forces from around 20000N (when the process is unstable) to around 7000N (when the process is stable). The effect of the actuator in machine vibrations is clear if the actuator is deactivated during the cutting process (Figure 10). The surface finishing shows the benefits of these results as well.

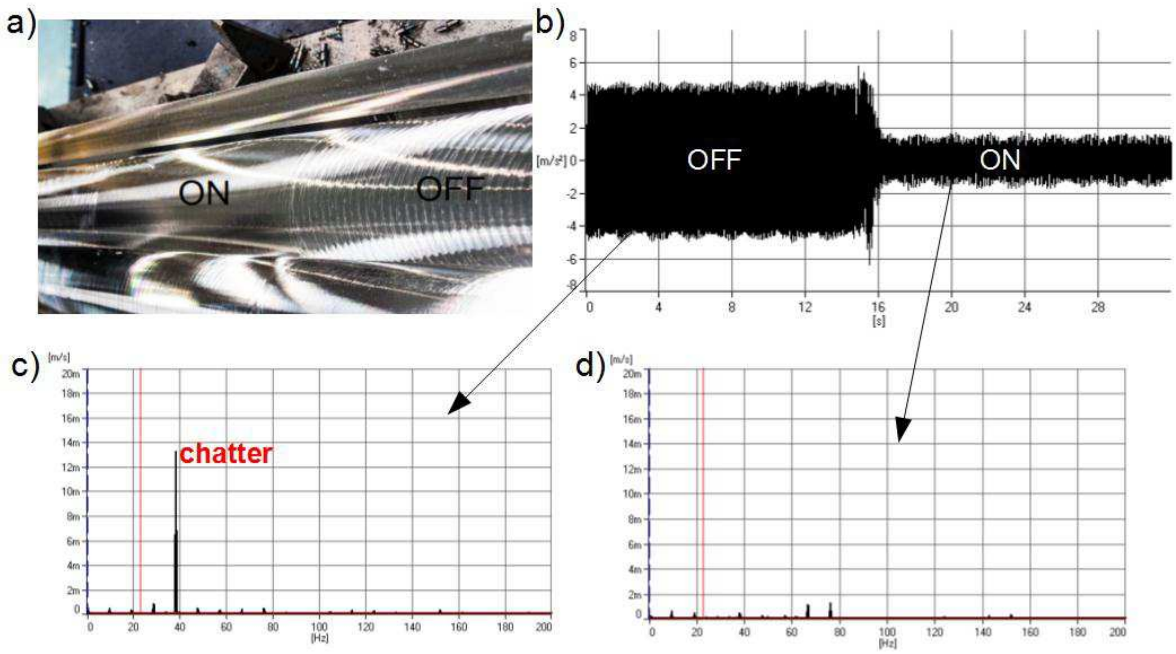

Figure 10: Cutting test (ram overhang 1000 mm, spindle speed of 440 rpm and 5.5 mm depth of cut), with and without active damping; a) Surface finishing; b) Time domain signal of machine acceleration; c) and d) Vibration spectrum with actuator deactivated and activated, respectively.

Nevertheless, it should be commented that when the maximum force is required during a long time, the actuator is heated. The performance of the actuator does not change while higher temperatures than 100ºC are not achieved, so the cooling system employment is strongly recommended for these cases. In reference to mechanical behavior, it remains constant, but a revision of the guiding system is recommended after a long operating time. Sometimes, a dynamic response measurement of the actuator can be sufficient for proving that it remains fine.

5. CONCLUSIONS

The work presents the development of a new concept of spindle head for heavy roughing operations. It is based on the integration of an electromagnetic inertial actuator inside the spindle head and overcomes the flexibility limitations of this type of milling machines.

A mechatronic model, where the effect of inertial actuator control loop is simulated including the regenerative effect of milling process, is employed for evaluating different actuator designs. The model has provided the design guidelines for the active device and control parameters.

In this way, a new single axis electromagnetic actuator with 1000 N force capability has been developed and validated by different experimental tests. Then it has been completely integrated on the spindle head and SORALUCE FL6000 milling machine.

Finally, experimental cutting tests are presented, where the machine with the new integrated actuator has been able to double its cutting capability, achieving the design objective.

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REFERENCES

1. **Altintas, Y. (2012). “Manufacturing Automation”.** *Cambridge University Press*, 2nd edition, New York, USA.

2. **Bediaga, I.; Zatarain, M.; Munoa, J.; Zelaieta, O. (2009).** "Analysis of Stability of Structural Modes in Milling Processes". *5th CIRP International Conference and Exhibition on Design and Production of Machines and Dies/Molds,* Kusadasi, Turkey.
3. **Brecher, C.; Schulz, A. (2004).** "Active Auxiliary Mass Damper". *Proceedings of ISMA 2004 – International Conference on Noise and Vibration Engineering*, Leuven, Belgium.
4. **Brecher, C.; Schulz, A.; Week, M. (2005).** "Electrohydraulic Active Damping System". *CIRP Annals – Manufacturing Technology*, vol. 54, no. 1, pp. 389-392.
5. **Bies, D.A.; Yang, T.M. (1968).** "Hybrid Vibration Isolation System for Helicopters". *Shock and Vibration Bulletin*.
6. **Bilbao-Guillerna, A.; Barrios, A.; Mancisidor, I.; Loix, N.; Munoa, J. (2010).** "Control laws for chatter suppression in milling using an inertial actuator". *Proceedings of ISMA 2010 – International Conference on Noise and Vibration Engineering*, Leuven, Belgium.
7. **Bilbao-Guillerna, A.; Azpeitia, I.; Luyckx, S.; Loix, N.; Munoa, J. (2012).** "Low Frequency Chatter Suppression using an Inertial Actuator". *9th CIRP International Conference on High Speed Machining*, San Sebastian, Spain.
8. **Claeyssen, F.; Magnac, G.; Sosnicki, O. (2008).** "Moving Iron Controllable Actuators". *Actuator 2008 – 11th International Conference on New Actuators*, Bremen, Germany.
9. **Cowley, A.; Boyle, A. (1970).** "Active Dampers for Machine Tools". *CIRP Annals – Manufacturing Technology*, vol. 18, no. 1, pp. 213-222.
10. **Den Hartog, J.P. (1985).** "Mechanical Vibrations". *McGraw-Hill*, 4th edition, New York.
11. **Ehmann, C.; Nordmann, R. (2002).** "Low Cost Actuator for Active Damping of Large Machines". *IFAC Conference on Mechatronics Systems*, Berkeley, USA.
12. **Huyanan, S.; Sims, N.D. (2007).** "Vibration Control Strategies for Proof-Mass Actuators". *Journal of Vibration and Control*, vol. 13, no. 12, pp. 1785-1806.
13. **Loix, N.; Verschueren, J.P. (2004).** "Stand Alone Active Damping Device". *Actuator 2004 – 9th International Conference on New Actuators*, Bremen, Germany.
14. **Mancisidor, I.; Zatarain, M.; Munoa, J.; Dombovari, Z. (2011).** "Fixed Boundaries Receptance Coupling Substructure Analysis for Tool Point Dynamics Prediction". *Proceedings of 13th CIRP Conference on Modelling of Machining Operations*, Sintra-Lisbon, Portugal.

15. **Munoa, J.; Zatarain, M.; Bediaga, I.; Lizarralde, R. (2005).** “Optimization of Hard Material Roughing by means of Stability Model”. *Proceedings of the 8th CIRP Conference on Modelling of Machining Operations*, Chemnitz, Germany.

16. **Munoa, J.; Mancisidor, I.; Loix, N.; Uriarte, L.G.; Barcena, R.; Zatarain, M. (2013).** “Chatter suppression in ram type travelling column milling machines using a biaxial inertial actuator”. *CIRP Annals – Manufacturing Technology*, vol. 62, pp. 407-410.

17. **Sajedipour, D.; Behbahani, S.; Tabatabaei, S.M.K. (2010).** “Mechatronic Modeling and Control of a Lathe Machine Equipped with a MR Damper for Chatter Suppression”. *Proceedings of 8th IEEE International Conference on Control and Automation*, Xiamen, China.

18. **Sims, N.D. (2007).** “Vibration Absorbers for Chatter Suppression: A New Analytical Tuning Methodology”. *Journal of Sound and Vibration*, vol. 301, pp. 592-607.

19. **Tlusty, J.; Polacek, M. (1957).** “Beispiele der behandlung der selbsterregten Schwingung der Werkzeugmaschinen FoKoMa”. *Hanser Verlag*, Munchen, Germany, pp. 47-56.

20. **Tobias, S.A.; Fishwick, W. (1958).** “Theory of regenerative machine tool chatter”. *The Engineer*, vol. 205.

21. **Yang, Y.; Munoa, J.; Altintas, Y. (2010).** “Optimization of Multiple Tuned Mass Dampers to Suppress Machine Tool Chatter”. *International Journal of Machine Tools & Manufacture*, vol. 50, no. 9, pp. 834-852.

22. **Zaeh, M.F.; Waibel, M.; Baur, M. (2009).** “A Computational Approach to the Integration of Adaptronic Structures in Machine Tools”. *Computational Structural Engineering*, pp. 1017-1028.