Journal of Machine Engineering, Vol. 14, No. 4, 2014

magnetic bearings, spindle, damping

Luis URIARTE[1*], Inigo ETXANIZ[1]
Ion ITURBE[1], Mikel ZATARAIN[2]
Jokin MUÑOA[2]

CHATTER SUPPRESSION IN A HIGH SPEED MAGNETIC SPINDLE BY ADDING DAMPING

Magnetic bearings are used in several applications, but they didn't succeed in the machine tool industry, due basically to their low damping, with its associated low chatter free cutting capacity. This work shows the development of a high speed milling spindle supported on magnetic bearings with high chatter resistance. That is achieved by using control strategies which add damping to the bearings. Excitation by means of the magnetic bearings is used to identify its dynamic behaviour and to optimize the damping parameters. Finally the results of the industrial validation are presented; showing an increase of six times in the cutting capacity.

1. INTRODUCTION

The applications of high speed milling are many today, in very diverse sectors such as the machining of large aluminium structures for the aerospace industry and machining of moulds and dies in special steels. Machine tools for these applications need to employ high power and high speed spindles in order to increase the productivity. Ceramic bearings are commonly used for these high speed applications. The bearings usually limit the lifetime of the spindle, and periodically, depending on the working conditions, they need to be replaced consequence of the wear. The operation and maintenance of high-speed spindles have high associated costs, and from the standpoint of total cost of the spindle during use (Total Cost of Ownership - TOC), the cost to purchase the spindle only accounts for 16% of total [1].

This limitation still unsolved justifies that spindles with non-contact bearings become in a very promising alternative, although they are inherently more complex and expensive. The active magnetic bearing technology, which allows support shafts without any mechanical contact and therefore no wear, allows the operation with much less maintenance requirements [2].

[1] IK4-TEKNIKER, Mechatronics Dept., Eibar, Basque Country, Spain
[2] IK4-IDEKO, Dynamics & Control Dept., Elgoibar, Basque Country, Spain
* E-mail: luis.uriarte@tekniker.es

Although the first experiences with magnetic spindles [3] were somewhat disappointing, especially by the limitations of electronics and sensors used, currently there are robust products for many applications requiring smooth rotation without friction, very high speed and a high level of reliability: semiconductor manufacturing, vacuum pumps, gas compressors, or flywheels for storage of kinetic energy [4]. In particular the higher price of the product outweighs when analyzed its cost over its use (TOC), mainly due to its higher reliability and reduced maintenance operations [5].

Magnetic bearings haven't succeeded in machine tool applications. Due to their limited force capacity, the benefits of magnetic bearings are found in high speed applications, where chatter instability tends to appear on tool and spindle modes. The damping of a high speed spindle determines its performance [6] against chatter even in a greater extent than its stiffness, and damping in spindles with magnetic bearings is extremely low in the current situation. For decades there have been researches on magnetic bearings, but without reaching the required damping for application in high-speed milling. Thus, in the literature we can find examples of active vibration control by controlling the magnetic bearings [7], showing effective bandwidths that are not sufficient for the case of high speed milling [3]. Also magnetic bearings have been applied into a spindle that already has ball bearings, in order to obtain the dynamic response of the head and introduce an actuator to control vibrations of the system [8].

Lack of damping makes milling spindles with magnetic bearings prone to instability problems during machining. These instabilities lead to very high levels of vibration, which can force the stop of the machine. The risk of instability prevents the use of greater depths of cut, and consequently limits the productivity of the machine.

In this paper we present the results of a research with the goal of adding damping and therefore to avoid chatter vibration in magnetic milling spindles.

2. DESCRIPTION OF THE MAGNETIC SPINDLE

The magnetic spindle used (Fig. 1) has 70kW of nominal power and 36000rpm of maximum speed. It is the third generation of magnetic spindles developed by the collaboration between the company Goialde High Speed and IK4-TEKNIKER [9]. Its weight is 175kg, being the rotor 35kg, and its external diameter is 240mm. It is able to withstand radial forces in the tool up to 3,200N and axial forces up to 2,500N. The whole system: spindle, bearings, control algorithms and electronics are own-developed.

The spindle is supported by two radial magnetic bearings, one in front side and the other in the back side. Each of these bearings is composed by four electromagnets and two non-contact position sensors (Fig. 2), which through attractive forces exerted on the rotor, allow controlling its position on a plane. By means of two radial bearings and one thrust bearing, the system is able to control the spindle levitation in five degrees of freedom, being the sixth d.o.f. the rotation of the shaft.

The control runs over self-developed electronics (Fig. 3), which consists of three main parts: an FPGA that performs the control of HW and also performs the filtering operations of highest sampling frequency; a DSP performs the real-time control of the position in five d.o.f.; and a processor dedicated to the monitoring and communication with the external elements.

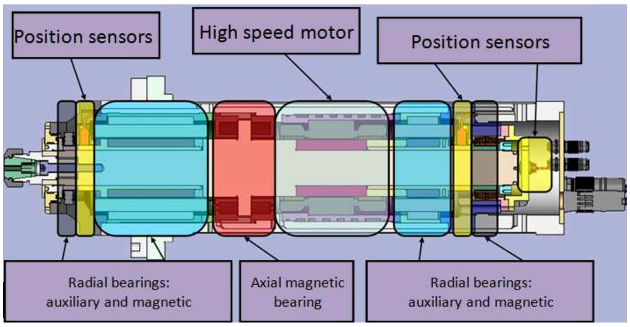


Fig. 1. Magnetic spindle design

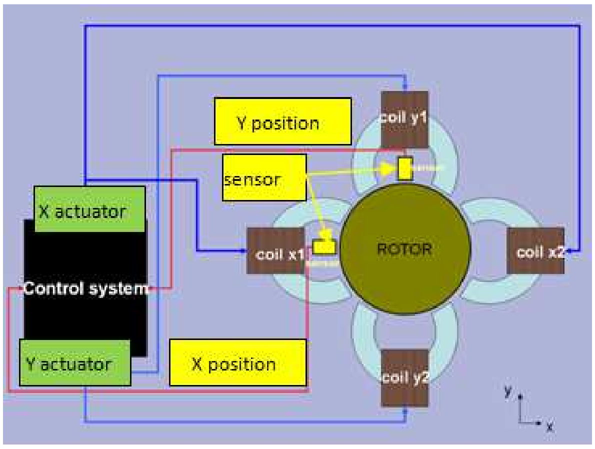


Fig. 2. Conceptual image of a radial magnetic bearing with four coils, two sensors and the control system

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.

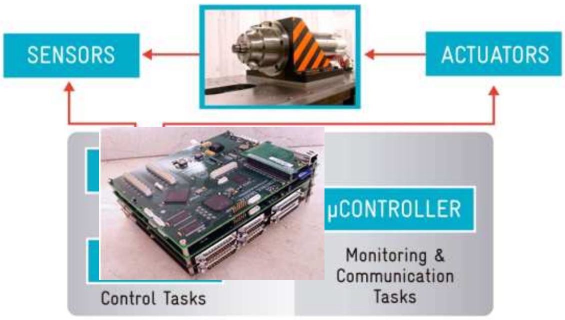
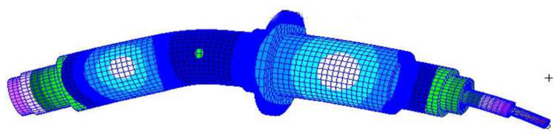


Fig. 3. Control electronics developed IK4-TEKNIKER to control the magnetic levitation system

3. DYNAMIC ANALYSIS

3.1. THEORETICAL AND EXPERIMENTAL DYNAMIC RESPONSE

The theoretical dynamic analysis of the spindle has been performed by FEA, showing a first mode of vibration around 660Hz (Fig. 4).



Description Tool-holder / Tool	1st natural frequency
Longest tool holder / Tool ∅ 24 mm	659 Hz
Shortest tool holder / Tool ∅ 20 mm	728 Hz
Medium tool holder / Tool ∅ 16 mm	674 Hz

Fig. 4. First mode of vibration of the spindle

A dynamic calibration function has been developed to obtain the experimental dynamic response of the spindle using the capabilities of the magnetic bearings. Therefore, a disturbance is introduced in the form of linear frequency modulation in the current of one of the bearings, and the displacements caused by the disturbance are measured. Then, the transfer function of the coil current and the measured position (Fig. 5a) is calculated. It is noted that the first vibration mode is at 602Hz, and it has a damping factor of 0.5% (shaft structural damping).

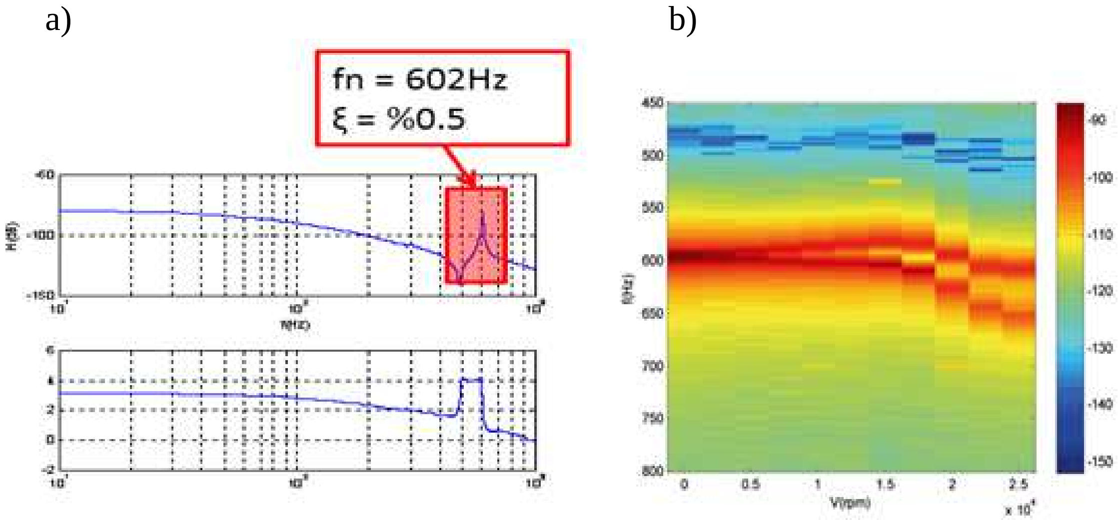


Fig. 5. Experimental open-loop Transfer Function at 0rpm (a), and the evolution of the first natural frequency with rotational speed (b)

In rotary systems, the natural frequencies evolve with rotational speed. This effect is usually more pronounced in high inertia systems, but it is also felt in high speed systems like the one we are dealing with. Obtaining the transfer function at different rotational speeds between 0 and 25,000rpm, we get the evolution of the first vibration mode (Fig. 5b).

3.2. STABILITY COMPARISON BETWEEN MAGNETIC BEARING SPINDLE AND BALL BEARING SPINDLE

The stability against chatter vibrations of the magnetic bearing spindles has been correlated with conventional spindles. The same tool/toolholder combination has be located in the magnetic spindle and in a FISHER MFW-1709/20/2 VCS HSK-A63 ball bearing spindle. This conventional spindle has similar dimensions. The stability of the cutting process has been analysed by means of stability diagrams and cutting tests using AL7075T6.

The stability diagrams show the maximum depth of cut which can be achieved free of chatter at different speeds. Lobe diagram has been obtained for both spindles following a multi frequency approach [10]. A three flutes end mill with a diameter of Ø20mm and helix angle of 30º has been selected. Aluminum AL7075T6 has been considered and the cutting coefficients used are Kt=796MPa, Kr=169MPa, Ka=222MPa. Radial immersion is 10mm and the down milling operation has been chosen.

The measured frequency response functions show that the dynamic stiffness of the conventional assembly has been seven times higher due to lack of damping of the magnetic spindle. The modes of the conventional spindle presented an average damping ratio of 2.8% in front of the 0.5% of the magnetic spindle.

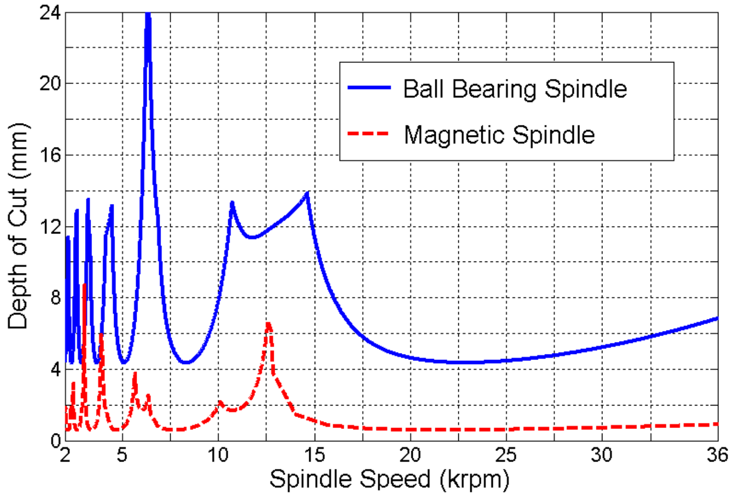


Fig. 6. Lobe diagram comparison (ball bearing vs magnetic spindle)

Figure 6 shows the comparison of limits of stability for a conventional ball bearing spindle (blue) vs a fully magnetic spindle (red). Main conclusion is that the chatter-free limit depth of cut is around 4.3mm for the ball bearing spindle and 0.6mm for the magnetic bearing spindle. Chatter frequency varies in the lower lobe between 595Hz to 650Hz.

3.3. CHATTER EXPERIMENTAL ANALYSIS

The modes of vibration of the machine with low damping are responsible of the instabilities (chatter) during machining. The dynamic analysis showed in the previous section, shows therefore a possible cause of chatter, because the spindle has vibration mode around 600Hz with a very low damping.

This cause has been confirmed experimentally by means of cutting tests. Several cutting tests has been performed to determine the actual stability limit considering the previous cutting conditions. Figure 7 shows, for instance, the frequency spectrum of the measured positions in the spindle sensors while chatter occurring (cutting conditions: 21,000rpm, feed=1200mm/s, depth=10mm). The vibration level is very high at a frequency close to the first vibration mode (indicated in red); there are also several peaks at frequencies which are rotation frequency apart from the chatter frequency (multi-frequency chatter), and also the forced vibration harmonics appear.

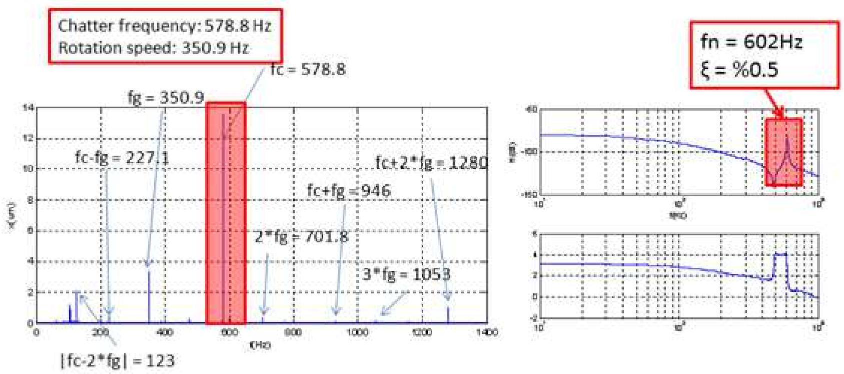


Fig. 7. Experimental study of chatter. It is observed that chatter vibration corresponds to the first vibration mode

We conclude, therefore, from this test that the instability observed during machining is related to the first natural frequency of the shaft.

4. INCREASE THE DAMPING OF THE FIRST MODE

4.1. CONTROL STRATEGY

So, the first goal is to increase the damping of the first vibration mode in order to machine with larger depths of cut, and therefore obtain higher productivity, without instabilities. This damping of the first natural frequency shall be achieved without affecting the stability of the levitation control. The levitation control is performed by a two-loop control: velocity loop and position loop. One way to damp the natural frequency with the same control would be to increase the bandwidth of the velocity loop up to the frequency that we want to be damped. However, the limited bandwidth of the magnetic actuators creates an important shift in relation to the frequency to be damped, so this limits the bandwidth of the velocity loop. So we are not able to damp frequency of interest.

The question is therefore the following: How a vibration mode at a frequency which is close to the limit of the actuators bandwidth can be dampened? And how we do it without affecting the stability of the levitation control? The solution found has been to introduce additional damping only in the frequencies close to the vibration mode. For that, the control structure shown in Fig. 8 was designed. A component at frequencies around the vibration mode is added to the commanded force signal by the two loops levitation control. The use of a bandpass filter enables the added component to affect the frequency range close to the dominant natural frequency only; therefore the stability of the control loop is not affected.

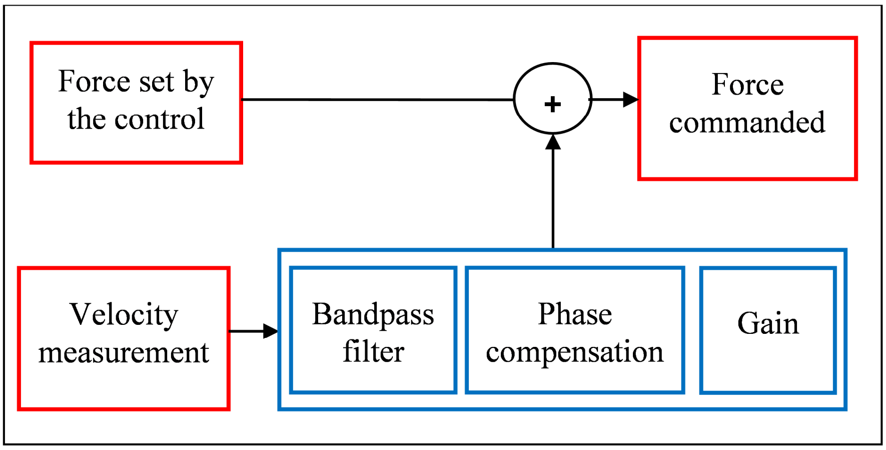


Fig. 8. Block diagram of the damping algorithm

The original idea was to use a Direct Velocity Feedback (DVF) for the controller acting on the vibration mode. However, due to the limited bandwidth of the actuators and to the large phase shift introduced at these frequencies, from 100 to 600Hz, the DVF would not dampen the system and even would have the opposite effect. Therefore the introduction into the control algorithm of a compensation for the phase shift of the actuators at the frequency band of the vibration mode was preferred, achieving this way the required damping at those frequencies.

4.2. EFFECT OF THE DAMPING

Figure 9 shows the system response to the impact test with the damping algorithm implemented (red line) and without it (blue line). It can be observed that the first vibration mode gets a noticeably damping. Without damping algorithm, the damping factor of the mode is %0.49 whilst with damping, it increases to %4.75. This new level of damping is higher than the average damping ratio showed by the conventional spindle, and according to the theoretical machining analysis shown in section 3.3, the stable limit depth of cut should increase importantly, the productivity of the spindle would also increase overcoming one of the main drawbacks of the magnetic levitated.

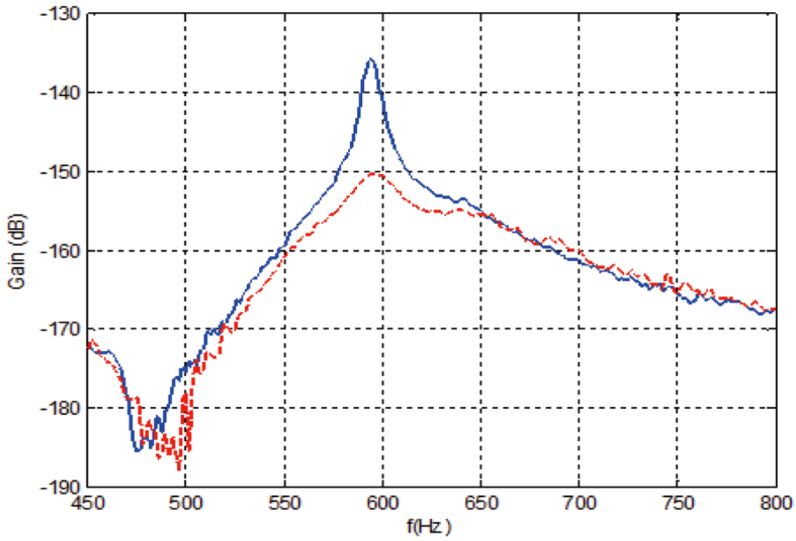


Fig. 9. Frequency response of the system to a perturbation without (blue) and with (red) damping algorithm

5. RESULTS OF THE MACHINING TESTS

5.1. STABILIZING THE MACHINING PROCESS

It was experimentally found that the activation of the damping algorithm is able to stabilize the machining process. The verification consisted in machining test at unstable conditions switching the damping algorithm on and off. Figure 10a shows the reduction of the vibration once the damping algorithm is active. The tool contact starts shortly before 1s, and remains up to 8.5s; the algorithm is activated after 4s.

The frequency content of this signal enables the analysis of changes in the frequency components of the vibration during the test. It is observed that up to the moment of activation of the damping algorithm there is a self-excited vibration close to the spindle natural frequency (see Fig. 10b), and the same pattern that was shown in Fig. 6.

Figure 10c shows the condition of the workpiece after the machining process. A transition at the time of activation of the damping algorithm is clearly observed, as after activation chatter marks disappear.

This test shows that the damping algorithm developed is able to stabilize the machining process.

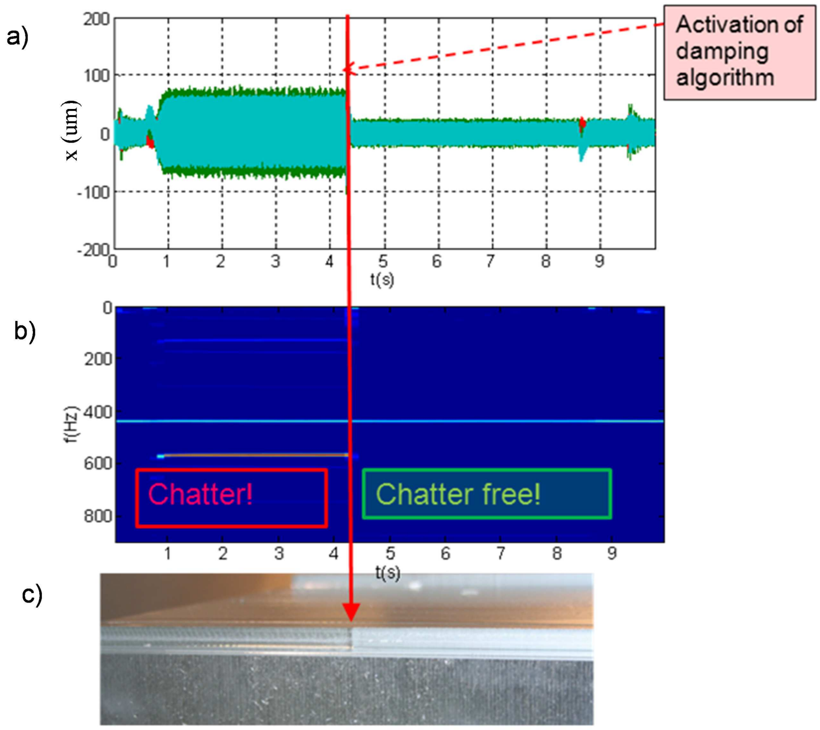


Fig. 10. Effect of the damping algorithm during machining: a) position sensor signals, b) time-frequency diagram, c) machined surface

5.2. INCREASE OF THE PRODUCTIVITY

At the beginning of the work the chatter vibration limited productivity at a maximum removal rate of 175cm^3/min. When trying to increase the removal rate, the chatter vibrations became so large that forced to stop the machine in order to avoid bearings damage.

After the implementation of the damping algorithm, the maximum removal rate was increased up to 1,125cm^3/min (cutting conditions: 25,000rpm, feed=7,500mm/s, depth=15mm, width=10mm; tool: 20mm, 3 flutes), i.e. 6 times higher productivity than without the damping algorithm. Moreover, the limitation in the new conditions is not due to chatter instability of the process but the impact at the workpiece entrance which causes large deviations of the tool.

6. CONCLUSIONS

Magnetic bearing spindles have a huge capability for high speed machining applications when damping factor is improved. The paper shows mainly the work done to increase the cutting capability of these systems; first making the dynamic characterization of the spindle; and later designing and implementing the damping algorithm for the first vibration mode of the spindle.

The new magnetic spindle was tested with real machining cases, and an increase of 6 times the original production capacity of the spindle was achieved, avoiding instabilities during machining.

ACKNOWLEDGMENTS

The works described in this article have been performed within the European Project Dynxperts (260073/FP7-2010-NMP-ICT-FoF). We appreciate the cooperation received by the Basque Country University (UPV- EHU), and in particular by the company Goialde High Speed.

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