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UNBALANCE ANALYSIS OF HIGH SPEED MOTOR SPINDLE

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Abstract: *The main spindle or the motor spindle is the key element of a machine tool and in the condition of high speed movement the dynamic behavior is absolute necessary to be stable. This paper highlights the need for control over the spindle unbalance along with the action needed to reduce it. The research proposes an experimental protocol to clarify differences that occur at high rotational speed. Even in the situation of a minor residual unbalance obtained at low spindle balancing or during the rig test, the appearance of critical speed is very probable. The paper proposes an experimental procedure based on two type of balancing, at the low speeds on the balancing machine and at high speed directly on the machine tool. The study provides also an analysis of the balancing appearance and the classification of the vibration level according to the ISO standard conditions.*

Key words: *machine-tool, motor spindle, unbalance, balancing, vibration.*

1. INTRODUCTION

The growing demands of the current industry impose an increasing control on the cutting technology, which determines an increase of precision, respectively of the machine-tool quality. Using the high speed spindle needs more improvement regarding the dynamic behavior. The knowledge of the behavior is one of the skills to be capable of the future integration in digital configuration.

Many research works are related to the dynamic topic regarding machine tools and spindle, but even in these conditions, the dynamic problems related to spindle vibration remain still a topic with many questions and issues. Even in high precision conditions and high technology, the main spindle presents critical speed range, which leads to critical cutting zones. These critical zones can be sourced by the main shaft/spindle-machine-tool, being influenced by the dynamic parameters such as mass and stiffness. The use of methods for diagnosing the cause of vibrations, especially at high speeds, becomes mandatory. The main vibration form and fault of high speed machine tool spindle is the imbalance [1].

The analysis of the balancing condition at high speeds was subject of several works, both in terms of diagnosis and reduction [2, 3]. The high level of unbalance generated an excessive forced vibration can cause the adverse effects on the life span of the spindle and on surface quality of the workpiece. The paper aims to highlight the importance of the spindle condition.

2. DESCRIPTION OF THE PROBLEM

The dynamic behavior of the spindle is influenced by different factors which are dependent of the spindle and machine tools assembly [4, 5, 6]. One of the most important factors influencing the proper behavior is the unbalance. The causes of the spindle unbalance can be varied depending on manufacturing method and technological procedure, repair method, balancing quality or final assembly on the machine-tool. The unbalance is highlighted both in case of new spindles and in case of repaired spindles. The spindles used at high speeds are the most sensitive. The character of rigid shaft or rigid rotor confirms a good stability except for high or maximum speeds. In order to reduce the unbalance of the spindle, the following two methods are applied: the spindle balancing on the balancing machine and the field balancing or balancing directly on the machine-tool. In any of the cases, for the grade of balancing quality the ISO1940-1/2003 and ISO21940-2/2017 is applied. Depending of the spindle type and speed, quality grade of balancing can be G2.5, G1, G0.4 [7, 8]. For high speed spindle, the quality grade of balancing is usually G0.4.

Under this condition, the vibration velocity is:

$$v = e_{per} \cdot \omega, \tag{1}$$

where v is the vibration velocity, e_{per} − permissible vibration unbalance according to ISO1940 and ISO21940, and ω − angular velocity.

In the case of maximum speed of 20 000 rpm, e_{per} is 0.45 gr·mm/kg or the eccentricity is 0.00045 mm (Fig. 1). The unbalance can be described by,

$$U_{per} = \frac{e_{per}}{M}, \tag{2}$$

where U_{per} represents the residual unbalance in g·mm and M is the rotor weight. According to Eq. (2), for a rotor mass of 9 kg, calculated symmetrical for the two plans, the unbalance corresponds at 0.1 g·mm.

The vibration velocity is measured using two unidirectional accelerometers, fixed on the rear side support and front side support.

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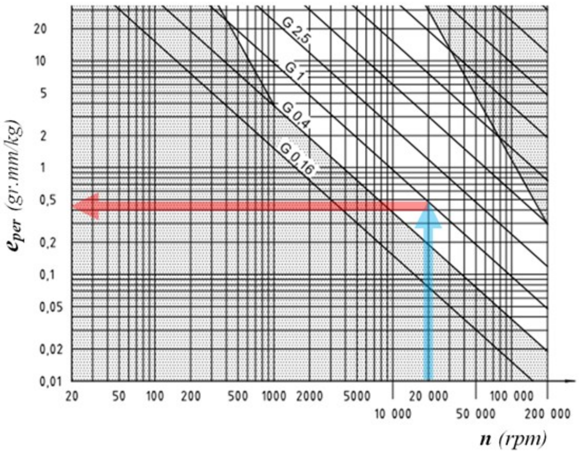


Fig. 1. The permissible residual unbalance based on balance quality grade and rotational speed, according to ISO1940-1 [4, 5].

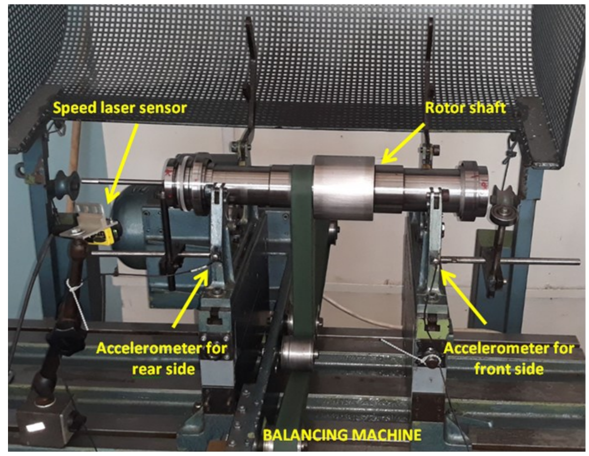


Fig. 2. The rotor shaft balancing on the balancing machine.

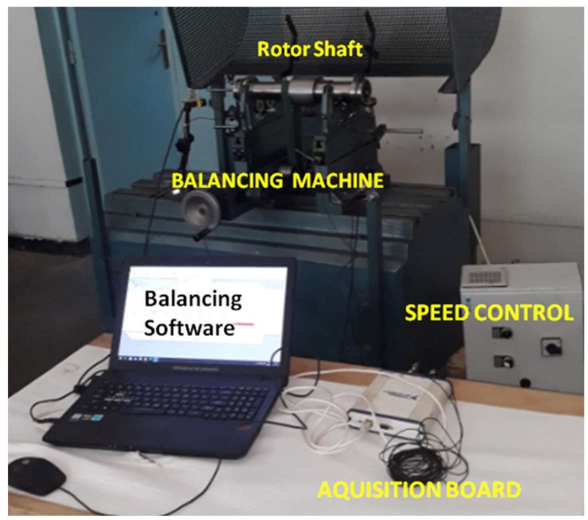


Fig. 3. The rotor shaft balancing device.

To know exactly the unbalance weight, the radius of the balancing plan is needed:

$$U_{per} = m \cdot R, \tag{3}$$

where m is the unbalance weight and R is balancing plan of the spindle. Thus, for a radius of 30 mm, the residual unbalance weight is 0.0033 g.

The accomplishment of the quality conditions imposed by the standard for rotor shaft balancing does not offer the guarantee of obtaining the same quality level on the machine tool. After assembling of the spindle on the machine tool, it can have a different dynamic behavior, as a result of the influence given by the dynamic parameters of stiffness and mass [4, 5]. The paper aims to highlight the condition of the spindle during the occurrence of vibrations due to critical speeds.

3. EXPERIMENTAL PROCEDURE

The balancing spindle is made in two stages, the stage one is for rotor shaft balancing on the balancing machine at the low speed and the stage two is the spindle balancing directly on the machine-tool.

3.1. Experimental balancing on the balancing machine at the low speed.

The rotor shaft is balanced on the balancing machine under the ISO standard condition, respectively ISO 1940-1 and ISO21940-2. The balancing procedure is made using a balancing machine with elastic and rigid option (Fig. 2).

The rotational speed of the rotor shaft is achieved using a laser sensor, type Banner QS30LLP. The accelerometers are of type MMF with 100 mV/g. The vibration measurement is made by a National Instrument USB 4432 board and a Fastview software (Fig. 3). The measurement parameters in the case of low speed balancing are given by a sampling rate of: 25 000 samples/sec, frequency range of 12.5 kHz, buffer size of 32 768 samples and a block size of 10 000 samples.

3.2. Experimental description of the spindle on the machine-tool

The unbalance characteristic of the rotor shaft is further verified on the machine tool. The spindle is a type of motor-spindle with 35kW, mounted on the Milling Center Doosan NX500. Vibration measurement is performed in three-dimensional configuration, using three accelerometers, fixed on radial (2 directions, 0 ° and 90 °) and axial direction (Fig. 4). The speed is measured with a laser sensor type Banner QS30LLP and synchronized with vibration signals (Fig. 4). Given the two configurations of test and measurement precision, the same type of accelerometers is used having a sensitivity of 100 mV/g. For a correct understanding of the vibration cause, an accelerometer is mounted on axial direction on the front side of the spindle.

The measurement during the run-up test is made using the same parameters: sampling rate of 25 000 samples/s, frequency range of 12.5 kHz, buffer size of 32 768 samples and a block size of 10 000 samples.

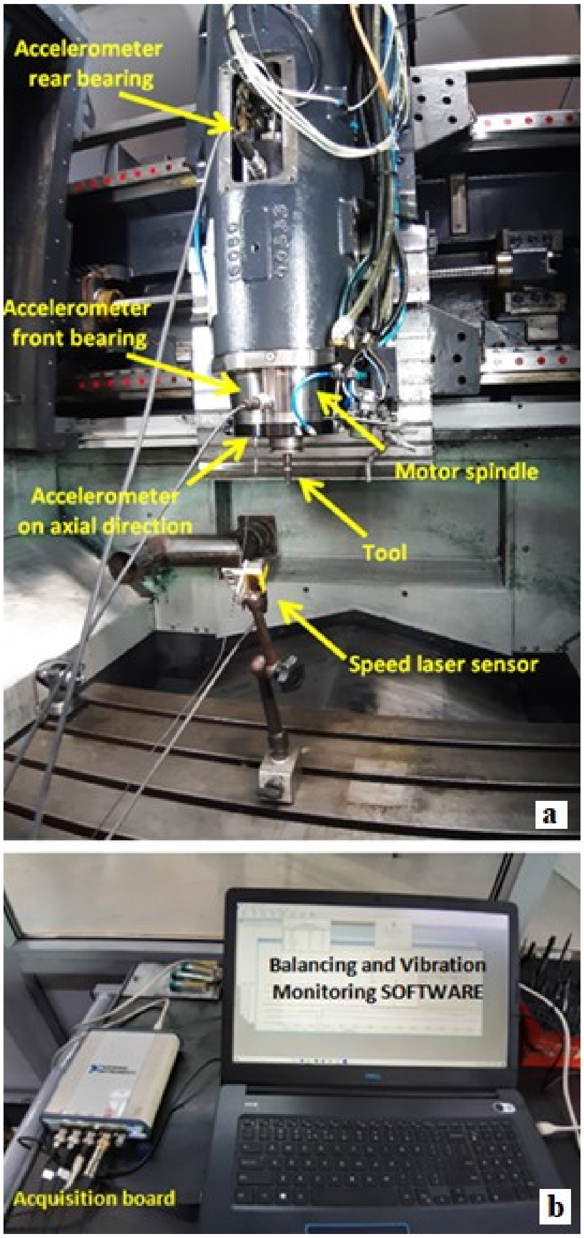


Fig. 4. The motor-spindle balancing done directly on the machine tool: *a* – machine tool and experimental set up; *b* – acquisition board and software interface.

4. RESULTS AND DISCUSSION

For a complete experimental analysis, the testing is performed considering the two stages: rotor shaft balancing on the balancing machine and spindle balancing directly on the machine tool.

4.1. Experimental balancing on the balancing machine at the low speed.

The balancing of the rotor on the balancing machine is performed at low speed in elastic configuration so that the response to high speed is as accurate as possible, with a low level of vibration. The balancing speed is 1936 rpm. The elastic setup of the balancing machine allows obtaining a more precise response to the centrifugal force.

During the testing for balancing in elastic configuration, the clear highlighting of the 1st order harmonic is observed, Fig. 5. In this situation, the balancing becomes much easier to be achieved.

In the rigid configuration, the 1st order harmonic can be penalized by the different position errors of the rotor shaft in relation to the balancing machine. There are many spindle repairs that perform balancing in a rigid system, which does not always provide proper results. This is also the reason for conducting an in-depth analysis of dynamic balancing techniques for a better choice regarding the quality of rotor shaft balancing. The initial unbalance generates an initial vibration of 0.47 mm/s rms for front bearing side and 0.37 mm/s rms for rear bearing side (Fig. 6).

After balancing and according to ISO1940-1/2003, the imbalance is reduced to 0.03 mm/s for front bearing plan and 0.01 mm/s rms for rear bearing plan (Fig. 7).

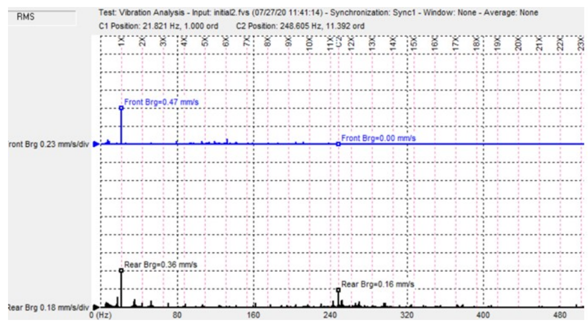


Fig. 5. The frequency spectrum of the rotor shaft on the balancing machine, before balancing.

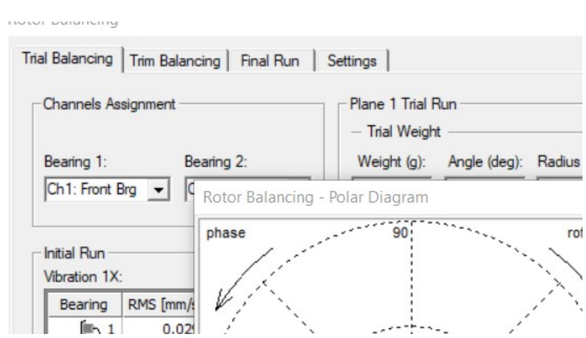


Fig. 6. Initial unbalance during the rotor shaft balancing.

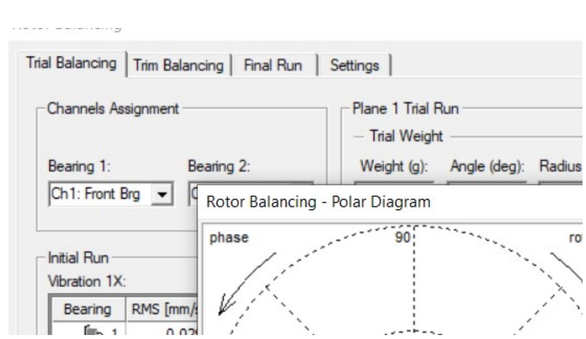


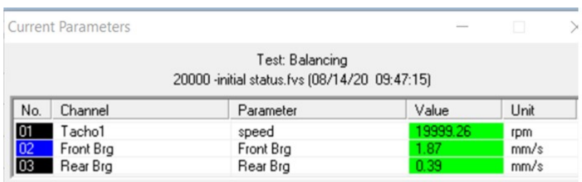
Fig. 7. Final unbalance during the rotor shaft balancing.

4.2. Experimental balancing on the machine tool

After vibration monitoring during running spin the spindle is inspected and unbalance identified. Because the stiffness and mass parameter change on the machine in relation to the spindle the appearance of the vibration at the critical speed it is very likely to occur. The overall vibration level is quite high (Fig. 8).

Taking into account the correct diagnosis of the spindle behavior, a detailed analysis of the main vibration components is required. The frequency spectrum shows an existence of the 1X – first order harmonic with the highest amplitude (Fig. 9). The polar diagram shows vector amplitude at 1.87 mm/s rms with a phase angle of 9°. The phase position of the vibration vector shows the existence of a dynamic balancing (Fig. 10).

The solution for decreasing the vibration level is the balancing directly on the machine tool, also called field balancing. Using the balancing method of influenced coefficient, the weight is added in to the plan holes of the spindle. The balancing procedure can be applied if there is a specific balancing plan. Also, another important condition is to use a master tool holder or a new tool holder, without wear and with a good geometrical precision.


Current Parameters

Test: Balancing
20000 -initial status.fvs (08/14/20 09:47:15)

No.	Channel	Parameter	Value	Unit
01	Tacho1	speed	19999.26	rpm
02	Front Brg	Front Brg	1.87	mm/s
03	Rear Brg	Rear Brg	0.39	mm/s

Fig. 8. Initial condition of the spindle at 20 000 rpm.

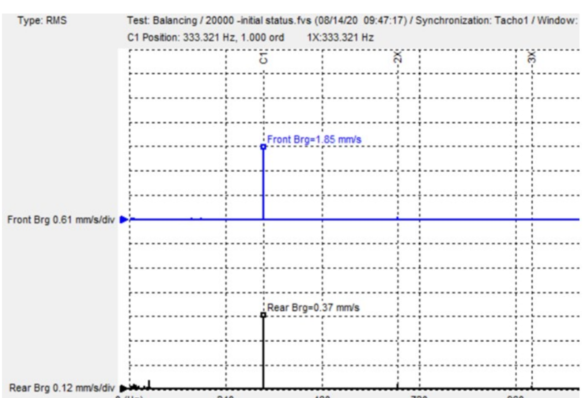


Fig. 9. The frequency spectrum before field balancing.

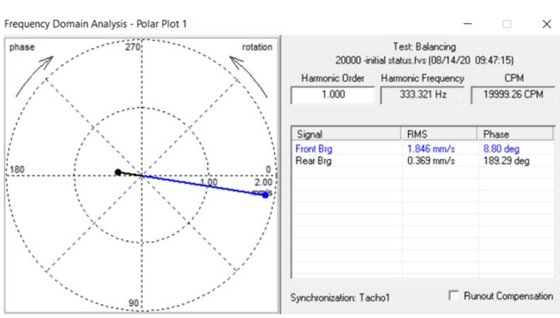
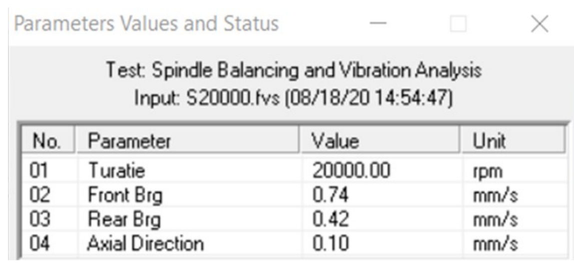


Fig. 10. Polar diagram at 20 000 rpm before balancing.

After field balancing, the vibration level at high speed is according to ISO standard 10816-1, group 2, 15−75 kW, and Fig. 11. The vibration level decreased due to the balancing and can be observed also in the frequency spectrum (Fig. 12). To analyze the vibration behavior after field balancing the axial vibration is measured in order to evaluate the frequency components also in this direction (Fig. 12). The vibration level on axial direction is low and shows a corresponding stiffness of the spindle in this direction. After balancing, the phase angle changes the position, which is also normal, leading to a slight increase of static imbalance and also to a vibration at lower speeds (Fig. 13). The residual imbalance is according to ISO 1940-1 and ISO 21940-2, being within the vibration limits.

The appearance of imbalance is much more probable in the case of the repaired spindles, because the degree of fatigue is higher and the risk of the critical speeds present in the cutting speed domain is inevitable.


Parameters Values and Status

Test: Spindle Balancing and Vibration Analysis
Input: S20000.fvs (08/18/20 14:54:47)

No.	Parameter	Value	Unit
01	Turatie	20000.00	rpm
02	Front Brg	0.74	mm/s
03	Rear Brg	0.42	mm/s
04	Axial Direction	0.10	mm/s

Fig. 11. Overall vibration at 20000 rpm, after field balancing.

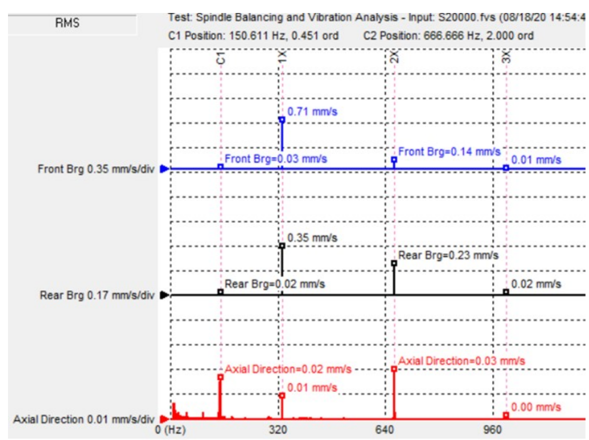


Fig. 12. The frequency spectrum at 20 000 rpm, after field balancing.

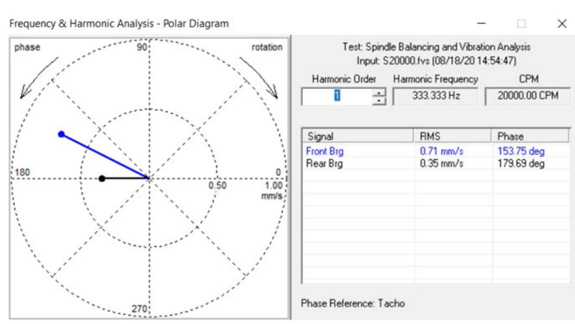


Fig. 13. Polar diagram at 20 000 rpm.

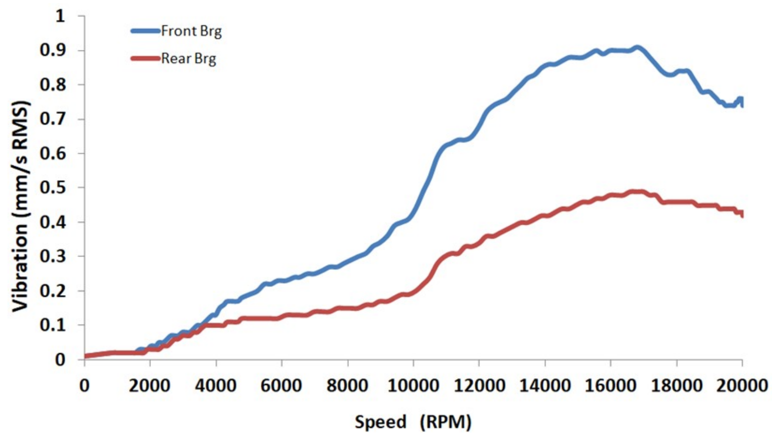


Fig. 14. The vibration evolution during the speed increase.

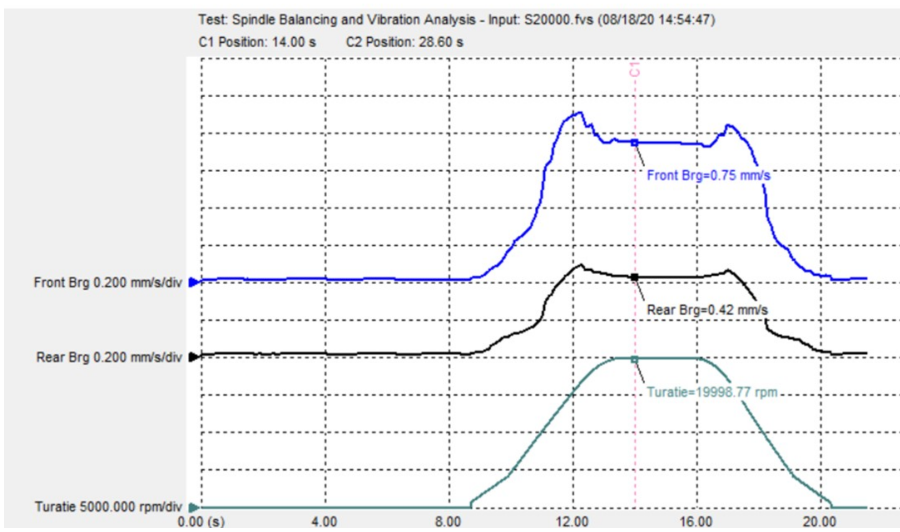


Fig. 15. The vibration trend during the run-up.

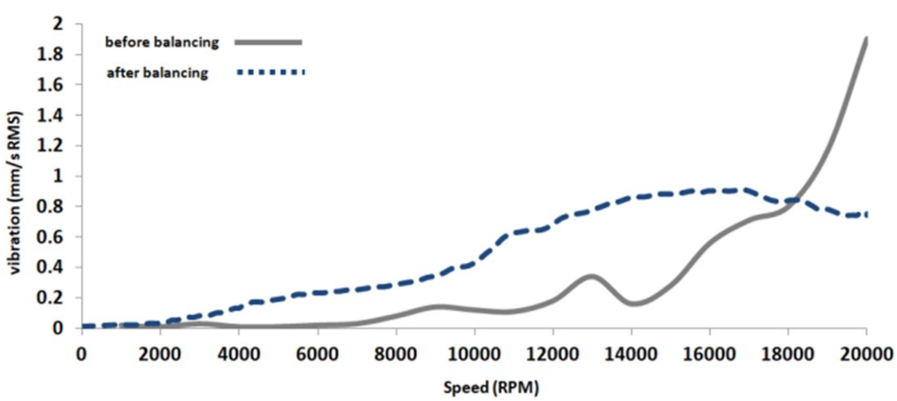


Fig. 16. The vibration evolution before and after balancing.

Figure 14 shows the evolution of the vibration after balancing. It can be observed the slight increase of vibration level at the low speed, but the vibration amplitude does not exceed 1 mm/s rms. The dynamic behavior presents a normal characteristic between run-up and cost-down situation, observed in the Fig. 15.

After balancing the vibration level offers the possibility to work on the whole speed range without limitation of the controller speed or avoid critical speeds (Fig. 16).

Considering the conditions presented, the balancing solution done directly on the machine-tool is a favorable one. At the same time, other possible causes can be analyzed that can influence the proper functioning of the machine during the cutting process. The field balancing can be used both in the case of the new spindles and in the case of repaired spindles, especially in the case of spindle in operation that can reach the situation of imbalance due to increased wear. The main condition to be fulfilled before applying balancing procedure is the analysis and identification of the imbalance, respectively of the vibration cause.

The balancing procedure can be difficult in certain conditions due to the type of spindle, defects or machine tool configuration. There are machine tools for which the use of wireless balancing solutions is mandatory, due to work safety conditions when the doors cannot be opened.

5. CONCLUSIONS

The paper presents a study of the spindle balancing at high speed. An experimental protocol was designed to highlight the balancing process from the rotor shaft phase to the spindle phase on the machine tool. The balancing procedure applied directly on the machine showed the improvement of the vibration behavior. The vibration level decreased, making it possible to use the spindle on the entire speed range.

The paper aims to highlight the importance of field balancing for research and especially for industry, where the need to solve the vibration problem due to the imbalance is quite common.

Also the stiffness parameter has very important role and the influence of the clamp/unclamp function, respectively the compression or decompression of the drawbar springs is very sensitive when the balancing is done only on the balancing machine. Even after test rig, the need for verification or field balancing is mandatory, because in the test rig conditions not all the parameters on the machine tools can be fully reproduced. The imbalance control of the spindle presents an influence on both condition: cutting process quality and a measure for predictive maintenance regarding the life span of spindle.

The research has different perspectives on the implementation of the wireless monitoring and field balancing solution. Another perspective is the rotor dynamic application in the case of the balancing improvement of the spindle at high speed. Knowing the influence coefficient as a result of the initial balancing they can be integrated in the active balancing system.

For industrial perspective, it is very important to redefine the design and construction of main spindle or rotor shaft so that the field balancing becomes a concrete solution.

REFERENCES

[1] S. Zhang, Y. Wang, Z. Zhang, *Online dynamic balance technology for high speed spindle based on grin parameter adaption and scheduling control,* Applied Science, Vol. 8, p. 917, 2018.

[2] X. Yun, X. Mei, G. Jiang, B. Wang, *A new dynamic balancing method of spindle based on the identification energy transfer coefficient*, Journal of Mechanical Science and Technology, Vol. 33, pp. 4595−4604, 2019.

[3] L. Zhang, J., C. Zou, X. Chen, Y. Che, *A new spindle method for field dynamic balancing of rigid motorized spindle based on real-time position data of CNC machine tools,* International Journal of Advanced Manufacturing and Technology, Vol. 102, pp. 1181-1191, 2019.

[4] S. Vafaei, H. Rahnejat, R.aini, *Vibration monitoring of high speed spindle using spectral analysis techniques*, International Journal of Machine Tools and Manufacture, Vol. 42, Issue 11, pp. 1223−12234, September 2002.

[5] Y. Li, X. Chen, P. Zhang, J. Zhou, *Dynamics modeling and modal experimental study of high speed motorized spindle*, Journal of Mechanical Science and Technology, Vol. 32, pp.1049−1056, 2017.

[6] Claudiu BISU, Miron ZAPCIU, Dorel ANANIA, Constantin ISPAS, Constantin MINCIU, *Dynamic Behaviour of Machine Tool Motor Spindle By Eddy Current Monitoring,* Proceedings in Manufacturing System, Vol.5, No.1, 2010.

[7] ISO1940-1:2003 *Mechanical vibration- Balance quality requirements for rotor in a constant (rigid) state- Part 1: Specification and verification of balance tolerances*

[8] ISO21940-11:2016 *Mechanical vibration- Rotor balancing – Part 11: Procedure and tolerances for rotor with rigid behavior.*