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IDENTIFICATION AND ANALYSIS OF LINKED SYSTEMS' DYNAMIC ACCURACY – PROPOSAL OF A MEASUREMENT APPROACH AND INSTRUMENTATION

In this paper a measurement approach and instrumentation for capturing the dynamic stiffness of a linked system is proposed. The approach is based on the loaded double ball bar (LDBB) in combination with an integral dynamic shaker. The innovation is the introduction of a dynamic load besides the static one given by the LDBB and consequently able to extract the frequency response functions, taking advantage of the possibility of orienting the LDBB in different directions of the machine tool. The dynamic behaviour is studied through experimental modal analysis. The test conditions are therefore more similar to a cutting situation in which both a static and a dynamic component of force characterise the system. However, it must be underlined that the introduced dynamic force does not replicate the one arising in a cutting operation, but it is chosen instead for its spectrum characteristics, i.e. the energy introduced must equally cover a given range of frequencies. The test method is able to reveal machine tool characteristics not obtainable with existing methods, for instance the variation of dynamic stiffness in the working space. The paper will include some theoretical aspects on the approach as well as some experimental investigation.

1. INTRODUCTION

The metal cutting technologies were developed as a consequence of the steam engine's advent, which posed the problem of machining metals. Nowadays, evaluation methods of machining systems are necessary to the combination of several factors which are: the need for increasingly higher accuracy of the machined parts together with the tougher raw materials used, and the greater importance of manufacturing non defective components at the first attempt, while at the same time containing the costs [1]. The need for evaluation methods of machining systems is also underlined by the introduction of ISO standards for testing [2].

One way of increasing the efficiency of a production system is to continuously improve, and develop new tests and evaluation methods of the machining systems. This is especially important when the goal is to produce a specific part correctly at the first attempt,

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in the quickest and most cost effective way. In this regard, new or improved test methods help to gather information about system status and can be stored in digital machining system capability models used for analysis and optimization.

1.1. MACHINING SYSTEM

Machining system can be defined as the closed loop interaction between the machine tool elastic structure and the cutting process, where the machine tool elastic structure includes workpiece, cutting tool and clamping device [3]. The machining system is defined as a close loop because there is an interaction between the cutting process and the physical entities involved. The cutting forces cause a deviation from the theoretical cutting line, which in turn causes a deviation in the forces, turning into a closed loop process as shown in Fig. 1.

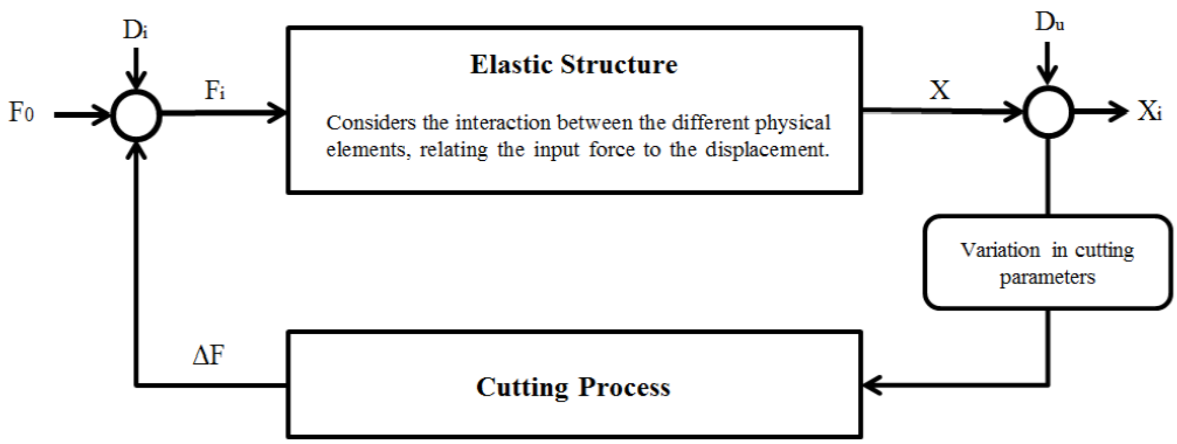

Fig. 1. The machining system seen as close loop interaction between the elastic structure of the machine tool and cutting process

The loop is an attempt to describe the close interaction between the cutting process and the physical entities involved the machine tool specifically. For instance, referring to Fig. 1 it is easy to observe that a deflection, X, in the elastic structure occurs due to the nominal cutting force F_0, in turn, the deflection causes a change of the cutting parameters and therefore of the actual cutting force. The actual force, F_i, is then given by the nominal value and the variation ΔF. In addition, D_i and D_u represent the disturbances in the system of the input and the output respectively.

The structural components in a machining system are always of stable dynamic character whereas the system, under influence of the cutting process can be unstable. A machine tool´s structural characteristics change due to various factors, such as cutting forces and subsystem configuration. For instance, if a force is acting between the spindle nose and the machine table, the stiffness can be very high. If the same force is acting between a tool holder system and the machine table, the resulting static stiffness can drop as much as 20 to 30 times, and if not compensated for will result in accuracy errors [4].

1.2. MACHINING SYSTEM CAPABILITY

In the case of machining system, the capability can be studied as machining system accuracy. There are several factors that can influence the accuracy and they can be grouped into four different categories of accuracy factor [5]: kinematic accuracy, thermal deformations, static deformations and dynamic flexibility. This paper focuses on static deformations and dynamic flexibility of machining systems.

Among the international standards, there are test methods which allow an evaluation of the accuracy of the machine tool in unloaded condition, therefore only the geometric accuracy is tested and neither the stiffness of the structure, nor the cutting process effects are considered. In contrast, the concept of elastically linked systems (ELS) presents a new idea on how to develop a test method [6].

1.3. ELASTICALLY LINKED SYSTEMS (ELS)

During, for instance, a face milling operation, due to the entrance and the exit of the teeth and the variation in chip thickness (both in down and up milling) the generated cutting force can be considered as the sum of two components: one that does not vary during the time (when, for instance, at least two teeth are always in cut) and it is addressed in this presentation as the static component of the cutting force; the second component, the dynamic one, which is due to the variation in chip thickness.

According to the ELS concept, the interaction between machine tool and cutting process is emulated by inserting an “elastic link” between tool and workpiece that closes the force loop [6]. In other words, by measuring the system deformation for various magnitudes and directions of the force vector acting on the structure, the positional and static accuracy of the machine tool under load conditions is evaluated. One example of a ELS is the loaded double ball bar (LDBB), which is an instrument consisting of four main parts: the mechatronic detecting and loading instrument, the two joints for table and spindle, the control system and the software for the analysis [6]. The LDBB is similar to a conventional double ball bar (DBB), both their appearance and the measurement procedure are similar, i.e. generating a circular path at a constant speed. In addition to a double ball bar, the LDBB allows the introduction of a static load between the table and the spindle joints, in order to evaluate the static deviations occurring in the machining system under loaded condition and to define the static stiffness as a function of the angle in the chosen testing plane.

Since loading the LDBB represents the effect of the static component of the load, it is then interesting to study the dynamic behaviour in this situation, i.e. a dynamic force can be introduced and then the response of the system is studied. The dynamic behaviour is studied through experimental modal analysis. The test conditions are therefore more similar to a cutting situation in which both a static and a dynamic component of force characterise the system (the static one is here dominant, like, for instance, in a face milling operation where the number of teeth is high therefore the static component of force is predominant over the dynamic one). However, it must be underlined that the introduced dynamic force does not replicate the one arising in a cutting operation, but it is chosen instead for its spectrum characteristics, i.e. the energy introduced must equally cover a given range of frequencies. The response is measured by accelerometers and in these points the frequency response functions are calculated.

1.4. EXPERIMANTAL MODAL ANALYSIS (EMA)

EMA is a technique developed in the 1970´s, which took advantage of the developments of digital computers. Its purpose is to determine the modal parameters of a physical model that describes the excitation and the vibration of the structure. In analogy to the Fourier series expansion, a vibrational field can be described as the superposition of functions that are typical for every structure; these functions are called mode shape functions [7].

EMA usually consists of mainly three steps:

1. A certain number of observation points must be chosen in order to be able to resolve the shape of the main different modes.
2. An exciting force is applied at a point and the acceleration/displacement is measured in the observation points and the FRFs are developed.
3. The modal parameters are calculated.

It is important to make a distinction between the concept of receptance and the one of dynamic stiffness. They are both frequency response functions, but the former is defined as the ratio between the motion and the exciting force, while the dynamic stiffness is the inverse ratio. Since the response of the system is usually measured with accelerometers, the acceleration response is captured and from there the other responses (typically the displacement FRF) can be obtained.

As previously stated, a central part of EMA involves generating an exciting force and two different sources are commonly used: the impact hammer and the shaker. Through the hammer, an impulsive force is introduced in the system, exciting all the frequencies with the same intensity. There are two main parameters that influence the generated force: the mass of the hammer and the material of the tip. The higher the mass the higher the intensity of the force, therefore, a higher mass is required when testing a heavier structure.

Exciting every frequency with the same amplitude, i.e. having an ideal impulse where the excitation time is zero, is impossible in practice. However, the harder the material of the tip of the hammer, the shorter is the impulse and therefore the condition will be closer to the ideal impulse.

The second most common way to generate the input force is by a shaker. Within this work only electrodynamic shakers are presented and used. An electrodynamic shaker consists of a moving coil placed in a permanent magnetic field. When an electric current passes through the coil, a force that follows the same time variation of the current is generated. Through a shaker it is easy to create a force with a chosen time history. It is usually recommended to use either a random noise (where all the frequencies are excited in the same way) or a sine sweep where all the frequencies within a chosen bandwidth are excited since the force presents a sinusoidal shaper with a varying frequency in the time from two chosen values [7].

2. COMPUTATIONAL MODEL

The objective of this paper was to both develop a test method for machine tools and at the same time provide proof that what has been assembled and tested has a solid theoretical foundation. To that end, and in order to develop an initial understanding of the machining system dynamic behaviour, a model of an ELS system has been developed with the Simulink (SimMechanics) environment of Matlab, see Fig. 2. The developed model is constituted by rigid bodies, masses, dampers and forces that represent the static and dynamic behaviour of the tool – tool holder – spindle joint and the front bearing in the spindle unit. The varying excitation force applied between table and spindle joint is simulated through the variation of spring stiffness. In this model the table joint is represented as a rigid body. Having rigid bodies means considering the deformations only in the joints between different bodies, this assumption does not constitute a problem since in a machine tool between 75% and 95% of the deformations occur in the joints.

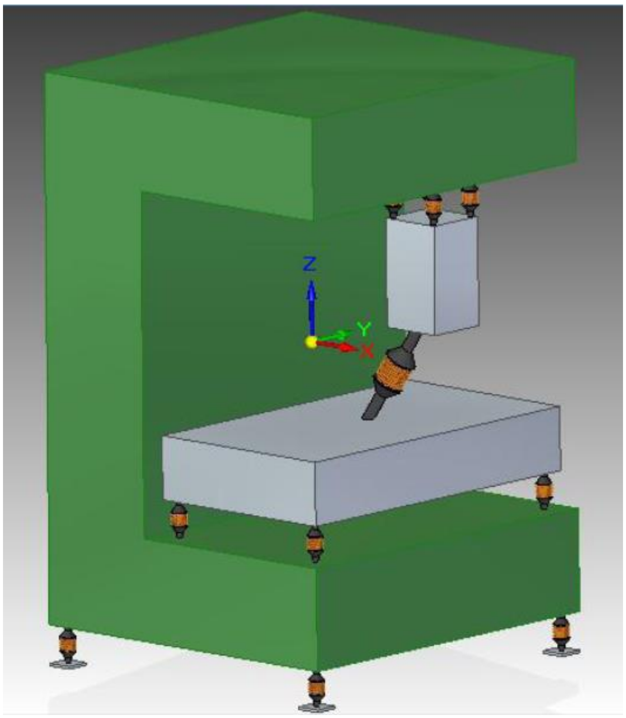

Fig. 2. Schematic representation of the developed Matlab SimMechanics model

The generated model attempts to emulate the ELS situation where the machine tool is considered as being composed of three masses, which are respectively: the frame of the machine, the table and the spindle-tool holder. The deformation in the connections between the different masses can be evaluated, since the joints in between are represented by springs and dampers. The input force introduced is a sine sweep vector of force and the actual deformation in the X direction has been detected and the maximum deformation at the different frequencies is showed in Fig. 3. The effect that increasing the stiffness of the elastic link has on the dynamic behaviour is also shown. Indeed, it is possible to appreciate how the frequency of the highest amplitude shifts on the right when a stiffer elastic link is introduced.

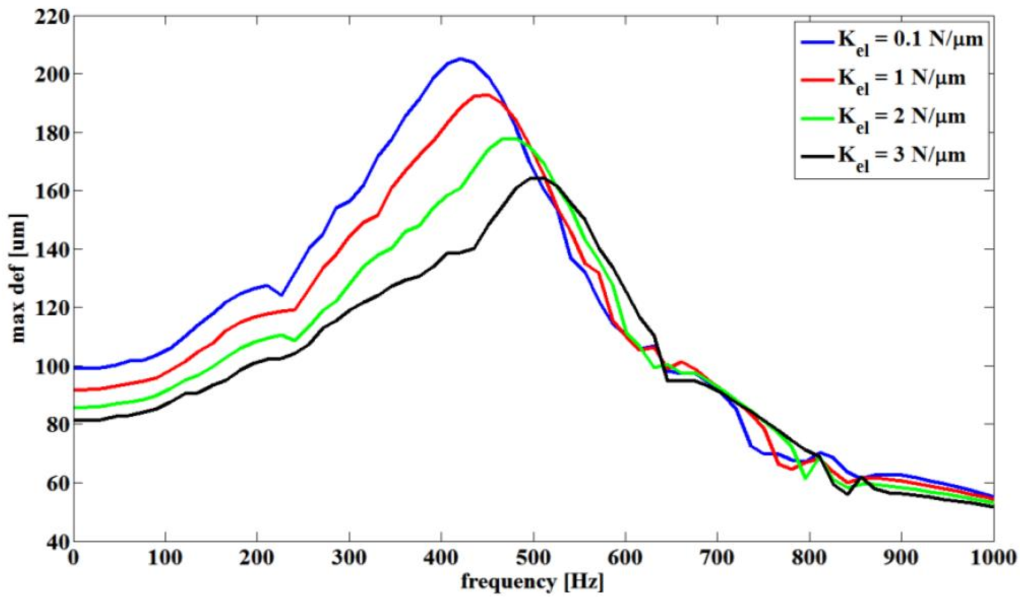

Fig. 3. Effect of the increment of stiffness over the dynamic behaviour

It is important to underline that the diagrams above are not frequencies response functions. Only the highest amplitude at each frequency is shown here. The FRF is calculated as the ratio between output and input of the system in frequency domain. Thus, even though the maximum deformation plots and the FRFs are very similar in shape, there is a conceptual difference between them that must be remembered.

The simulation results achieved from the developed model support the belief that a modal test of the machine tool would give different results when the static load is varied in the LDBB (which is the elastic link). An increment of the load in the elastic link should stiffen the system thus affecting the position of some modes. It is from this idea that the authors developed the experimental setup and the idea on how to test the dynamic stiffness of a machining system.

Starting from the principle of ELS and by means of the LDBB, which already proved to detect the static stiffness of machine tools [6], the innovative idea of the work consists of the introduction of a dynamic load besides the static one of the LDBB and the extraction of the frequency response functions of the new system. This basically means applying experimental modal analysis to the elastically linked system. Convinced that the best way to evaluate the dynamic behaviour of a machining system is by a closed loop of forces, to combine the static load given by the LDBB with a dynamic force, studying the system by EMA seemed to be a good way to leap forward. Thus, it has been decided to combine the setup of the LDBB test with a dynamic force (generated either by a hammer or by a shaker). Loading the LDBB simulates the effect of the static component of the cutting force, it is then interesting to study the dynamic behaviour in this situation, i.e. the dynamic behaviour of a closed loop system. The test conditions are closer to a cutting situation in which both a static and a dynamic component of force characterise the system.

It is worth noting that the dynamic force here does not have anything in common with the dynamic force of the cutting process, it is instead only meaningful based on its spectrum characteristics since it is needed to study the system through EMA.

In this research study what is not conventional is the use of EMA to test a loaded structure to evaluate how the dynamic behaviour of the machining system is related to the static load. In terms of dynamic force, different options were experimentally compared. The experimental setups and the obtained results are described in the following paragraphs.

3. EXPERIMENTS AND RESULTS

It is first of all important to clarify that the experiments performed aimed to support the following ideas:

1. Similar results were expected by the different setups adopted, a comparison between the FRFs attained by using different dynamic forces has also been done to be able to define the best way to test a machine tool.
2. As obtained from the model, it was sought to prove experimentally that a variation in the static component of force leads to a variation of the dynamic behaviour along a given direction of the machine tool workspace. This challenges the traditional way of studying the stability of the machining process through stability lobe diagrams, which are developed for a given natural frequency of the system coming from an open loop test, not considering the influence of the cutting process on the stability.
3. Machine tools are not symmetric.

To support the aforementioned ideas, several series of test have been performed. Initially, a hammer test to the open loop system has been done to be the reference of the study. Then, the loop has been closed by the LDBB, which has been aligned to the x-direction of a machine tool and both hammer and shaker test have been performed in different loading conditions. Similarly, the LDBB has been aligned to the Y direction in order to prove that the machine tool does not present an overall symmetric behaviour and the same tests as before have been performed. It is important to mention that the experiments have been run numerous times to estimate the repeatability of the achieved results. In the following paragraphs the different experimental setups are briefly described.

3.1. EXPERIMENTAL SET-UP

The data for the analysis are collected by measuring a gantry type machine tool using the LDBB test system and impact hammer or two different shakers. The LDBB's table joint was clamped to the machine table, with dimension 1240x550 mm, by means of a clamping screw at the middle of the table (see Fig. 4). The spindle joint was clamped in a standard ISO50 tool holder. The LDBB test system was employed to generate various static load magnitudes in the range 36 N (0.4 bar) the force required to hold the LDBB in place to 833 N (7 bar) a load between the machine tool table and spindle in XY planes of the machine tool workspace. One bar equals 119 N force. For further information about the LDBB instrument and test procedure see [6].

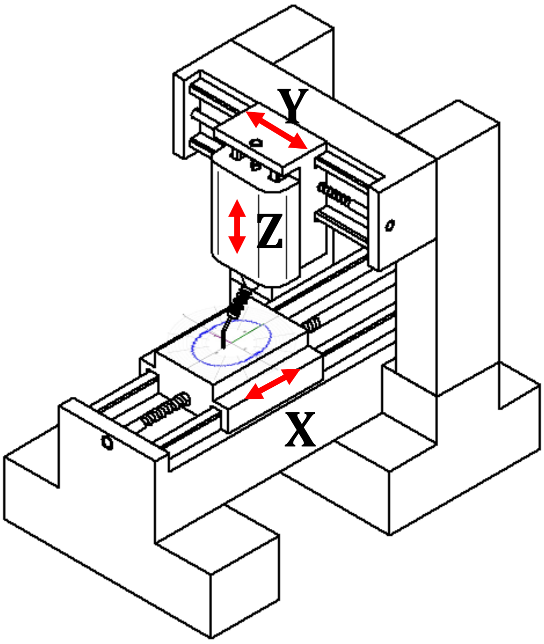

Fig. 4. Machine tool axis configuration of the investigated machine

3.2. IMPACT TEST

To perform the impact test the following equipment was used: the LDBB test equipment, an impact hammer, three accelerometers and the signal analyser. The experiment was performed both aligning the LDBB to the X direction and to the Y direction. In both cases, the dynamic excitation was applied at the spindle joint extremity of the LDBB following the same direction of the static load (axial to the ball bar). Three accelerometers were placed: two on the spindle joint, one as close as possible to the excitation point, the other one closer to the spindle and the last one on the table joint (see Fig. 5). The impact test was performed for four different loading conditions, i.e. for four different pressures in the LDBB: 1, 2, 4 and 7 bar, respectively.

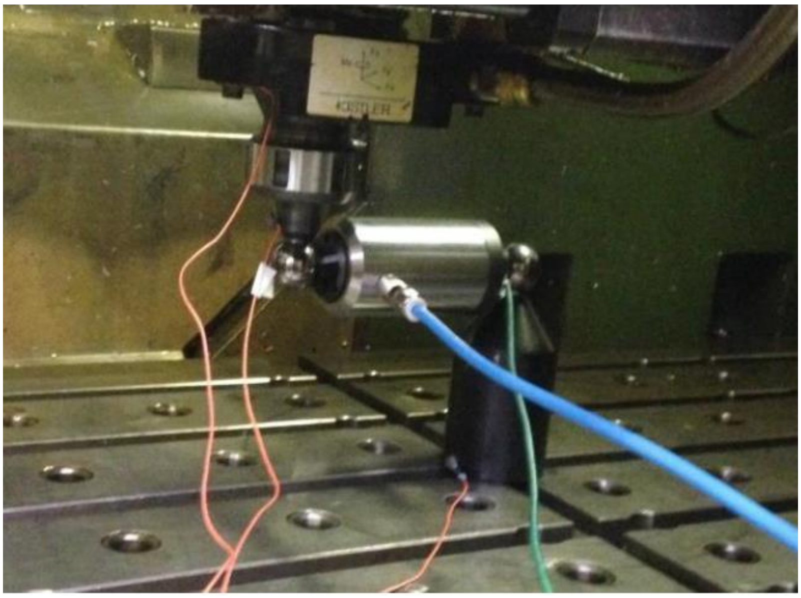

Fig. 5. Experimental setup of the impact test of LDBB static and dynamic characteristics

3.3. ELECTRODYNAMIC SHAKER TEST

When generating a random (or a sine sweep) excitation, the main difference in the setup involved the introduction of the shaker, which was aligned to the loading direction. Two different shakers were used, firstly an electrodynamic traditional one. The random force in the electrodynamic shaker varies between ±14 N and it covers a spectrum of 4000 Hz. The experimental setup is shown in Fig. 6.

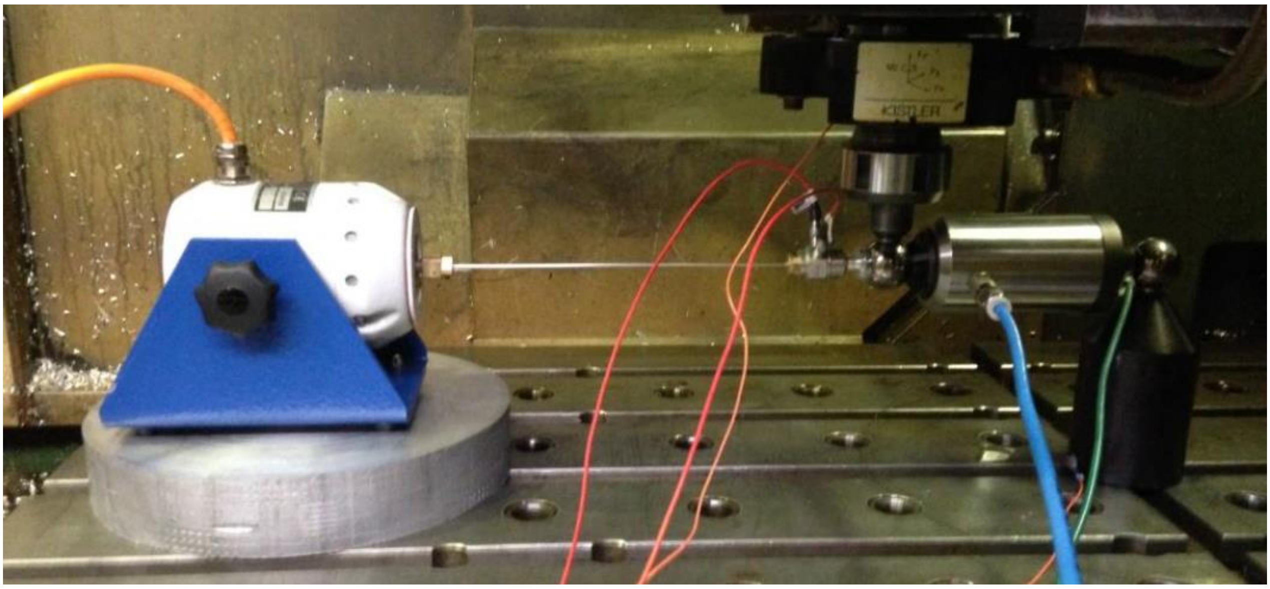

Fig. 6. Electrodynamic shaker and LDBB setup

3.4. THE INTEGRAL SHAKER TEST

Since the objective was to develop an ensemble of technologies that together could give origin to a new evaluation method, the traditional shaker did not seem to be a comfortable solution, indeed, it is not practical when it is necessary to test different directions in the plane/space. Therefore a more practical alternative than the traditional electrodynamic shaker was tested: the LSM Integral Shaker (Q-ISH). With it the resulting set-up was more compact allowing for an easier change of testing direction, see Fig. 7. An adaptor to connect the shaker to the spindle joint ball was developed to match the tip of the shaker to a non-flat surface (the spindle-joint ball).

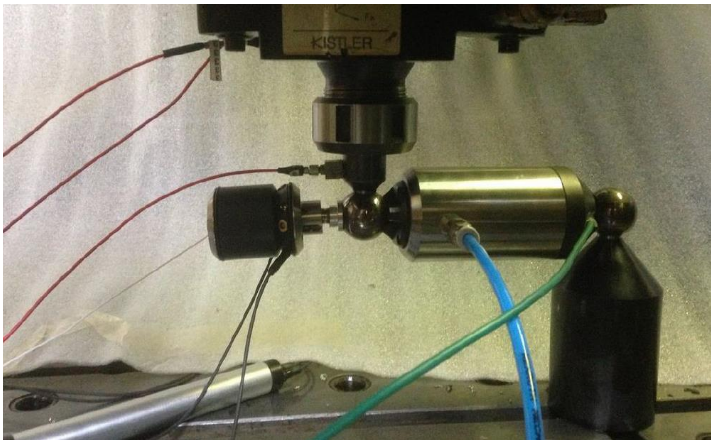

Fig. 7. Integral shaker and LDBB setup

The integral shaker has a stricter limit in frequency at 2000 Hz that is much lower than the other one, and lower than what can be achieved with an impact test. However, the results showed that the most influential modes are found between 400 Hz and 1500 Hz, thus the lower frequency limit can be considered as a secondary problem.

After performing a sufficient amount of tests with the described setups, it is with confidence that some patterns were caught, allowing to drive some conclusions, concerning:

- the comparison between different methods of gathering the frequency response;
- the effect of the preload used;
- the dynamic behaviour of the machine tool in different directions of the XY plane.

3.5. IMPACT VS SHAKER TESTS

As expected, the excitation mechanism has in impact on the obtained frequency response function, and this became more evident when the static load in the LDBB was higher. Indeed, running the test with low static load, the FRFs obtained through the hammer test are close to the one attained with the shaker test as can be seen in Fig. 8.

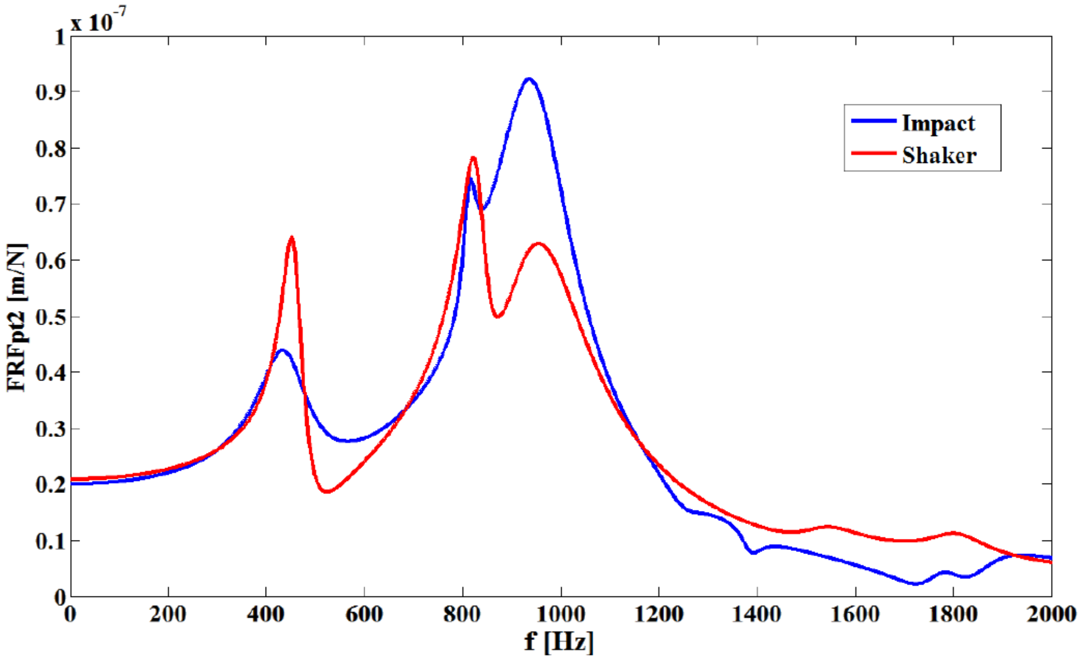

Fig. 8. FRFs through impact (blue) and shaker (red) tests, with a load of 1 bar (119 N) in the LDBB

On the contrary, increasing the static load causes a different effect in the two setups. In the impact test, the second mode tends to disappear with the increment of the static force, while the second and the third modes still remain distinct with a shaker test as shown in Fig. 9.

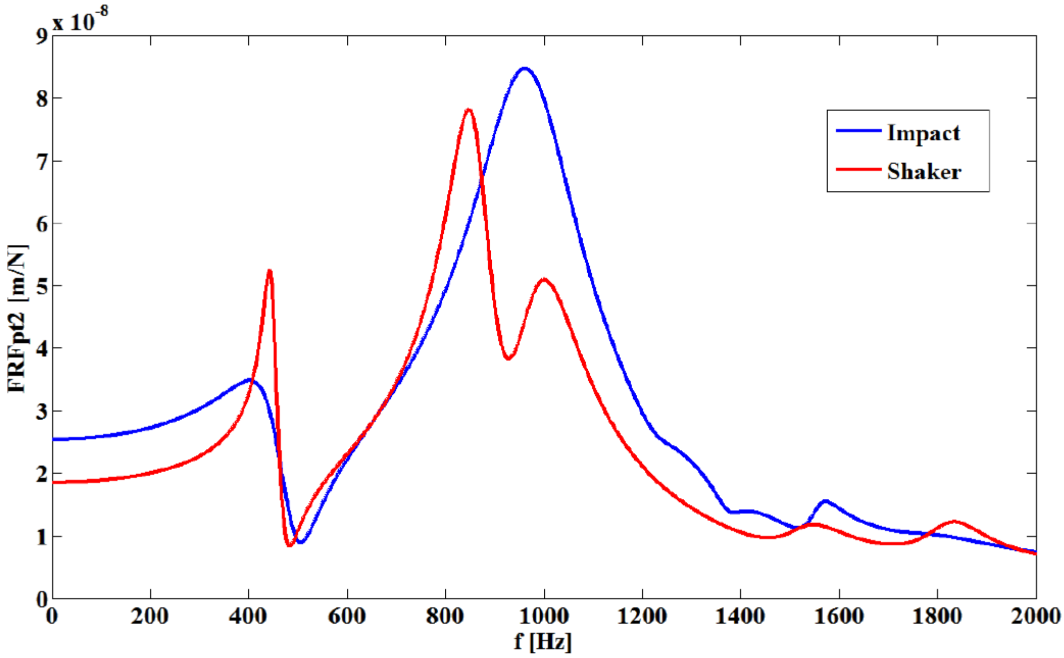

Fig. 9. FRFs obtained through impact (blue) and shaker (red) tests, load of 4 bar in the LDBB

Through a shaker test it is usually possible to have a better frequency resolution, this seems the case since the second and the third mode were really close to each other and the conclusion that can be driven here is that for our purposes the shaker is a better option.

3.6. EFFECT OF THE ELASTIC LINK

As stated, the innovative idea consisted in testing through EMA a closed loop machining system; therefore it was important to compare the result with the traditional open loop condition that was taken as a reference in the beginning.

Comparing the results between close and open loop conditions a significant difference can be appreciated, indeed, as it is shown in Fig. 10 the second and the third mode results shifted in frequency of around 300 Hz on the left.

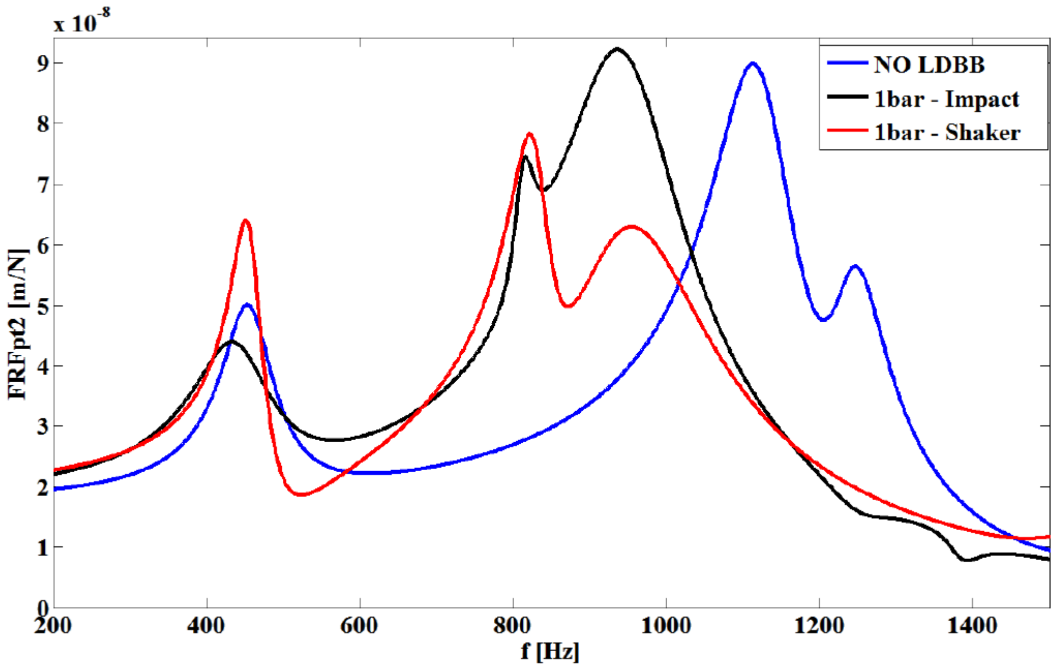

Fig. 10. Open (blue) and Closed (red/black) loop systems

This shift of modes can only be related to the introduction of the LDBB. Even though it is true that in closing the loop a mass was added to the system and this can only cause a shift on the left of the resonant frequencies, it cannot be considered as the reason for such a movement, indeed, even though a mass is added, it is only the vibrant mass associated to the mode that generates a variation thus not exactly the added mass to the system. Instead, the introduction of the ball bar changed the configuration of the structure, thus changing the studied system and the way it vibrates. Since a so big difference is perceived, it is important to understand how to relate the effect of the elastic link to the cutting process from a dynamic point of view.

3.7. EFFECT OF THE STATIC LOAD

One of the aspects of greater interest in the experiments was the influence of the static load (in the elastic link) over the dynamic behaviour. The pressure in the LDBB was adjusted to the following values: 1, 2, 4 and 7 bar. Performing the test in different loading conditions sought to describe the effect that the static component of the cutting process could have on the dynamic behaviour of the machining system.

It can be appreciated from the obtained frequencies response functions that an increment of the load leads to a movement to the right of the natural frequency of the second mode. The obtained FRFs from different loading conditions are presented in Fig. 11.

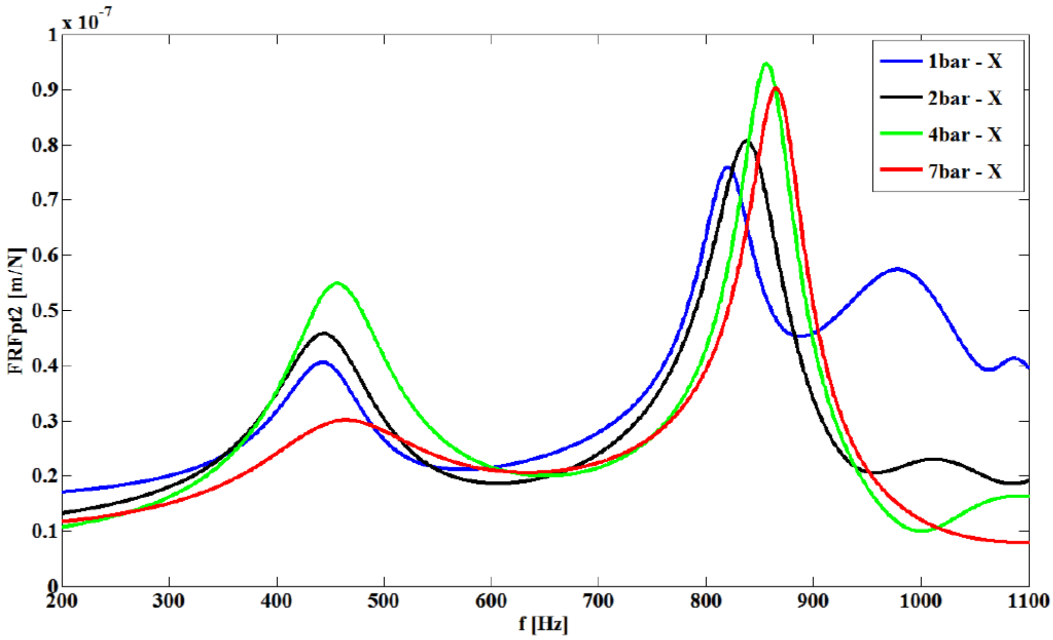

Fig. 11. Influence of the static load on the dynamic behaviour

As obtained from the model in the beginning of this work, an increment in the static load in the LDBB makes the system stiffer and therefore the main mode shifts to the right in the FRF. What can be concluded from this result is that a similar effect would occur while machining, i.e. a variation in the cutting parameters that leads to a variation of the static component of the force turns into a variation in the dynamic behaviour of the system.

3.8. COMPARISON BETWEEN X AND Y DIRECTIONS

The experiment has been performed with analogous setup both in the x-direction and in the y-direction, to give further proof that the machining system is not completely symmetric, especially from a dynamic point of view. The same loading conditions as for the

X direction were chosen, i.e. 1, 2, 4 and 7 bar of pressure in the LDBB. In Fig. 12 a comparison between the dynamic behaviour in the x-direction and the y-direction is shown.

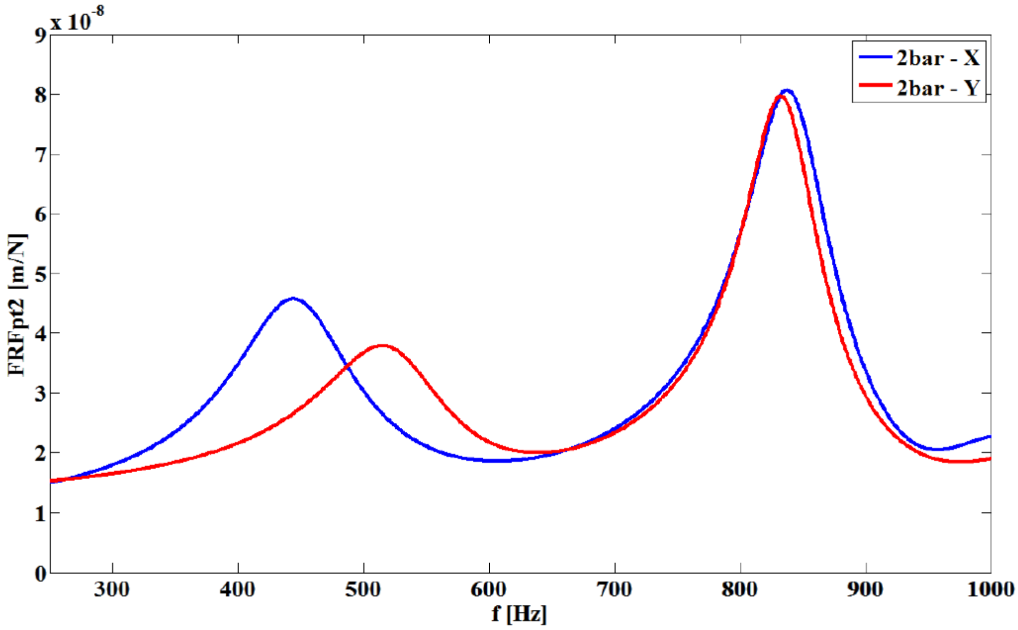

Fig. 12. Dynamic behaviour in the X (blue) and Y (red) directions

From the figure above, it can be recognised that the second mode has a symmetric behaviour, it is likely that the second mode corresponds mainly to the tool which is symmetric in shape and stiffness concerning the XY plane. Instead, the first mode varies showing that the dynamic behaviour of the system depends also on the direction in the workspace.

3.9. FACTORS AFFECTING UNCERTAINTY OF MEASUREMENTS

Dealing with measurements, correctness of the results remains even when the suspected components of error are carefully considered and proper action is taken to minimize their effect [8]. In the experimental setup that has been put in place, some factors might have influenced the quality of the results and affected the uncertainty related to them. These main factors can be summarised as:

1. A variation is shown between the results from the different experiments due to the fact that, each time a slightly different system was tested, due to the impossibility of re-creating exactly the same setup.
2. The integral shaker introduces a mass in the system, even though the producer guarantees the integral shaker introduces just 16 gr in the direction of measurement.
3. The positioning of the accelerometers and the number of accelerometers can influence the quality of the results.

4. The presence of some noise in the measurements can be seen from the coherence function that was not always close to the value one.
5. Some uncertainty is introduced by EMA and the subjective bias in the synthesis of FRF through the software LMS.

4. CONCLUSION AND DISCUSSION

Based on conventional modelling and measuring methods new methodologies are presented to fulfil requirements in machine tool characterisation supporting simulation for robust and predictive machining. Analysing results from the experiments, it seems possible to evaluate the dynamic behaviour of the machining system in a close-loop condition through the ELS concept. Following can be concluded:

1. A shaker test is preferable to an impact test to achieve results with better frequency resolution. The integral shaker represents the best option among the tested ones, being more flexible and providing the same quality of result.
2. Closing the loop of force causes a drastic change in the dynamic behaviour. This challenges the common way of studying the stability of the machining process and at the same time encourages that emulating the presence of the workpiece and the cutting process through an elastic link it is the proper way to go.
3. Varying the static component of the force influences the dynamic behaviour. A higher static load corresponds to a higher value of the natural frequency of the second mode.
4. It has been proven that the machine tool presents a different dynamic behaviour in different directions of the plane.

It was possible to distinguish modes coming from the machine tool and modes coming from the tool, and this implies that the applicability of the results is related to the tool used, which in this case was the spindle joint of the LDBB equipment. All the tools being different, this limits the applicability of this test method, however, it speaks hopefully of the concept which succeeded in capturing the behaviour of the entire machining system.

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