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Design of an active workpiece holder

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Abstract

Milling is one of the most used machining processes thanks to its high flexibility and high achievable quality. The performance of milling machines is constantly increasing, improving the convenience and increasing the competitiveness of this operation. However, the trend of performance improvement has found a technological limit: self-excited vibrations due to the dynamics of the system machine-workpiece-tooling (i.e. chatter). Chatter is the most dangerous dynamic phenomena that could happen during milling; due to its regenerative nature, it could lead the machine and the tooling system to a heavy fault or to the disruption of the workpiece. This paper develops an active workpiece holder that avoids chatter vibrations by a smart actuation of the workpiece. The design of the workpiece holder is a difficult task due to strict product requirements and the need to create a decoupled structure. The decoupling of the structure is a fundamental requirement of the product because this affects the controllability of the system. Axiomatic Design Theory is used to support the definition of the product requirements and the product architecture. After the definition of the optimal structure of the workpiece, the design features are integrated in order to obtain a functional decoupled structure.

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1. Introduction

Milling processes are continuously increasing their market penetration due to the need for products with high surface finish and dimensional accuracy, such as dies/molds for household appliances, automotive components, and products in the medical and aerospace industries. The required performance in milling operations is becoming more demanding, stressing the process parameters to their maximum extent; however, this is often limited by chatter vibrations that causes damage on the surface and shorten tool life [1]. Figure 1 reports a comparison between two machined surfaces: the first with the chatter phenomena (i.e., evident chatter marks on the machined surface) while the second is chatter free. Due to the importance of chatter on machining performance, recently many authors have focused their attention on the task of predicting chatter. The most used model to find chatter free machining parameters is the one proposed by Altintas [2], that uses a set of analytical differential equations to represent the phenomenon; by solving these equation is possible to find the border of a stable and unstable zone, usually graphically reported with a Stability

Lobe Diagram (SLD). An example of SLD is reported in Fig. 2. This approach requires as input the tooltip dynamics (i.e., tooltip Frequency Response Function, FRF) and the experimentally defined material cutting coefficients. However, the approach shows some limitations when these parameters are not available with good accuracy before the cutting operations. The main issue about the accuracy of the input parameters is mainly related to their strong dependency to the used cutting conditions and the specific use case. The tooltip FRF could be easily acquired with a static tap test but it is proven that it changes with the rotation of the spindle due to a stiffness change in the ball bearings and the gyroscopic effect of the rotating axis [3]. The same uncertainty is associated to the cutting force coefficients of the materials, which vary with the choice of the tooling and with the cutting speed and tool engagement [4-7]. For this reason, many experimental approaches have been developed to acquire the SLDs of the actual configuration with simplified test procedures. Among these, the most interesting are the ones that allow a point by point definition of the SLDs with simple machining tests, as the one proposed by Quintana et al. [8], that uses slotting operations with increased depth of

cut and the one proposed by Grossi et al. [9] that uses a speed varying test. These experimental approaches allow a good forecast of the SLD and so an optimal selection of the process parameters. However, there are two main drawbacks: the optimal process parameters selection needs some preliminary tests, which is not a cost and time efficient solution for small batches production, and sometimes the allowable cutting speed range does not allow to select a set of cutting parameters that could assure an acceptable process productivity rate. The idea of this paper is to design an innovative workpiece holder that could suppress the arising of chatter vibrations using an active vibration counteracting strategy. This solution would allow optimizing the machining process productivity without requiring a preliminary testing phase, providing a better flexibility and ease of use in an industrial environment. The idea of active workpiece is fairly new in literature, even if some implementations have already been proposed [10,11]. All the presented techniques require preliminary investigation of system dynamics to tune fixture behavior, thus not yet representing a real alternative to chatter prediction approaches, such as SLD. The aim of our work is to create a system that could operate without the need of any a-priori knowledge of the process or system dynamics. This constitutes a challenge both for the structural and control design of the system; the general idea has been to use a powerful and general architecture selection tool for the design of the system: the AD.
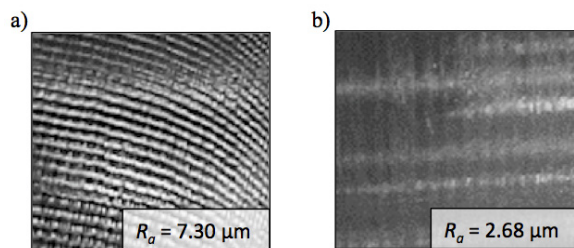

Fig. 1. Example of (a) chatter affected and (b) chatter free surfaces and roughness values.
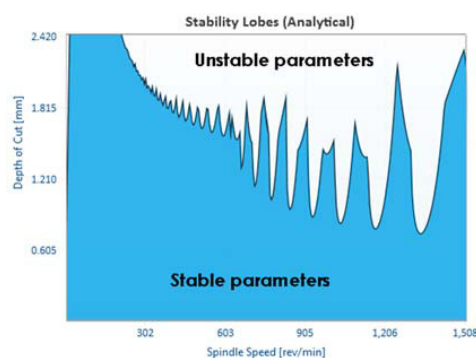

Fig. 2. Example of Stability Lobe Diagram

2. Axiomatic design methodology

Axiomatic Design (AD) is a methodology developed by Suh [12] in order to describe systems and products systematically by generalizing the design principles thanks to the use of two axioms. Axiomatic Design has been applied to a variety of different fields for the design of efficient and functional product [13]; one of the main advantage of this approach is the ease of integration with other design methodologies and analysis, for example: innovative design (TRIZ) [14], QFD [15], reliability [16], sustainability [17], fuzzy logic [18].

Essentially, AD identifies four design concepts: domains, hierarchy, zigzagging and axiomatic selection. The first concept defines four domains: the customer domain, the functional domain, the physical domain, and the process domain. The elements within each domain are Customer Needs (CNs), Functional Requirements (FRs), Design Parameters (DPs), and Process Variables (PVs). The domains are used to collect and analyze information at the design stage. The design process starts with the basic CNs information and populates the following domains as a result of the design process. For the specific use case the CNs are the requests of the milling operator in terms of ease of use and performance of the system, the FRs are the counteracting actions to suppress the vibrations while the DPs are the physical components and control logic architecture designed to perform the task.

3. Axiomatic Design for workpiece holder

3.1. Requirements of the workpiece

Active workpiece holders, often referred to as active fixtures, should be capable of supporting the workpiece during the machining process, achieving adequate stiffness to ensure the required level of precision of the manufactured product. As for general fixtures, these are the major design requirements that an active workpiece holder should address. Nevertheless, an intelligent fixture should be capable of actively control the stiffness in specific directions to counteract the arising of unstable vibrations. This is usually achieved by means of dedicated actuators that control workpiece displacements during machining operations, by exerting counteracting forces on the flexible parts of the fixture itself. Controlling workpiece displacement could interfere with the arising of unstable vibrations by disrupting the regenerative effect, that is the main responsible for the onset and self-maintenance of chatter vibrations [19].

The design of the active workpiece holder should hence be consistent with the specific application and requirements mentioned beforehand and should be capable of operating and exerting the desired counteracting forces and displacements while withstanding to the cutting forces generated during the process.

To define design constraints and target fixture design and requirements, specific experimental tests have been carried out on high material removal rate roughing operations that represent the limit application due to the high magnitude of cutting forces generated in the processes. Cutting forces and tooltip displacements have been measured in real machining tests by means of dedicated force transducers (Kistler 9257A table dynamometer) and accelerometers; these are the maximum forces that the fixture should withstand in operative conditions and the displacement magnitude that could be required to mitigate or suppress chatter vibrations. Measured

data showed a maximum cutting force value around 5 kN and a maximum tool-tip displacement around 0.15 mm.

Additional tests have also been carried out to investigate machine tool dynamics in order to identify the frequency of the dominant modes that are mainly responsible for chatter vibrations [2]. From the analysis, it emerges that dominant modes are present on X and Y axes (i.e., plane normal to tool axis direction), as expectable for the cantilevered tool, hence suggesting that a planar fixture configuration should be adequate to mitigate or suppress the phenomenon. This analysis also returned valuable information about chatter frequency value, that is generally close to the dominant mode frequency; in this case the chatter frequency value is expected around 3 kHz. Again this value seems adequately conservative, because the tool used for the experimental tests is sensibly stiffer than the ones used in general operations. More flexible tools would lower chatter frequency values, less critical from an operative point of view.

Measured cutting forces in tool axis direction are reference values to be used in dimensioning fixture frame. It is important that the fixture will maintain its position during machining operations, potentially returning loss of accuracy of the machined products. In this case cutting forces should be maximized taking into account also potential workpiece weight: assuming that the active workpiece holder will be used mainly in the die manufacturing companies, where the surface finish and tolerance are usually more critical than other field of applications, a maximum weight of 80 kg has been estimated. This is consistent with the machining of a large steel component, as for a general mold&die industry application. Again this represents a conservative value, lower workpiece weights would return a less critical situation.

3.2. Alternative design solutions

Taking into account the technical and scientific literature [20,21] it is possible to identify different architectures for active positioning tables (i.e., micro and nano positioning applications) that could be used as a base for the design of an active workpiece holder. The main solutions are three: parallel kinematics with three degrees of freedom (DOF), mainly used for precision positioning purpose, where controlling testpiece rotation is crucial, parallel kinematic with integrated stages and serial kinematic with nested stages. These are reported in Fig. 3.

At first, AD has been used for the identification of the optimal architecture using the first axiom. Machining tests, carried out using a dedicated dynamometer and accurately placed accelerometers, have highlighted how the main workpiece displacements to be counteracted are translations on the plane. Dynamic rotational contributions are negligible, especially if compared to the magnitude of the planar displacement. Starting from these considerations is possible to define the first level FRs - DPs of the system. Assuming X and Y are put on the horizontal plane of the workpiece holder, the FRs and DPs could be: FR1: "Maintain X position" while DP1: "actuated system to counteract X cutting and inertia forces"; FR2 and DP2 will have the same definition but for the Y direction.
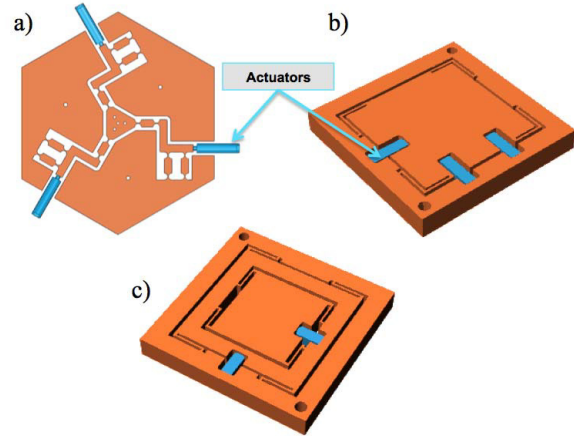

Fig. 3. Alternative workpiece holder structures: (a) parallel kinematics with three DOF, (b) parallel kinematic, (c) nested structure with serial kinematic.

For the parallel kinematic solution with three complete DOFs (i.e., Fig. 3a), the associated Design Matrix (DM) is rectangular due to the presence of three actuators that must be activated to have a force feedback on X or Y direction. This is a smart solution to obtain a fine positioning of the workpiece, as generally required in micro or nano positioning applications, but the higher complexity is beneficial only when also the rotations are taken into account. The DM shows a high degree of coupling for this solution.

$$
\begin{Bmatrix} FR1 - \text{Maintain X position} \\ FR2 - \text{Maintain Y position} \end{Bmatrix} = \begin{bmatrix} X & X & X \\ X & X & X \end{bmatrix} \cdot \begin{Bmatrix} DP1 - \text{Actuator 1} \\ DP2 - \text{Actuator2} \\ DP3 - \text{Actuator3} \end{Bmatrix}
$$

A less coupled result could be obtained for the integrated stages with parallel kinematic solution (Fig. 3b), where only one actuator is responsible for the force along the X axis. However, the system is still coupled.

$$
\begin{Bmatrix} FR1 - \text{Maintain X position} \\ FR2 - \text{Maintain Y position} \end{Bmatrix} = \begin{bmatrix} X & 0 & 0 \\ 0 & X & X \end{bmatrix} \cdot \begin{Bmatrix} DP1 - \text{Actuator X} \\ DP2 - \text{Actuator Y1} \\ DP3 - \text{Actuator Y2} \end{Bmatrix}
$$

The optimal solution for this application, where the compensation of the rotation is not required, is the last one (nested structure with serial kinematics, Fig. 3c), where the DM is uncoupled.

$$
\begin{Bmatrix} FR1 - \text{Maintain X position} \\ FR2 - \text{Maintain Y position} \end{Bmatrix} = \begin{bmatrix} X & 0 \\ 0 & X \end{bmatrix} \cdot \begin{Bmatrix} DP1 - \text{Actuator X} \\ DP2 - \text{Actuator Y} \end{Bmatrix}
$$

However, should be noticed that the coupling for this architecture could still arise if the perfect equal behavior of the flexure hinges is not assured. The natural tolerances of the production process could be responsible for different stiffness of each flexure hinge. In case of large difference on the dimensions of the four flexure hinges the system will provide

also a small rotation when a single actuator is activated. This rotation is due to the different stiffness of the hinges and could not be compensated by the other actuators. This phenomenon is usually referred to as cross-talk. So a very high attention must be used in defining the manufacturing tolerances of the hinges. It must be noticed that the rotation issue arise only when the four hinges have a different behavior: if the production machine has a systematic error (e.g. it create a slot with a dimension that is 2 micron larger than required) this is not a problem from the fixture point of view. The problem arise when the machine tool used for production provide low repeatability results. Since the operation is within the field of precision machining, the possible technical solution – highest level PVs – are micromilling and wire EDM. The first technology is usually the cheapest so the initial process planning will be based on this solution. The tolerance will be controlled using the same tool for the manufacturing of the four hinges – so the different runout of different toolkit will be not an issue – and controlling the tool wear thanks to a correct selection for process parameters (lower level PVs).

For large displacement the hinges the behavior could become nonlinear, this behavior would be eventually compensated by the control logic, tuned after an initial test of the prototype. The displacement needed to avoid the chatter phenomena has been estimated in 150 micron at maximum; considering the low value we expect that the nonlinearity of the system will be negligible since the designed hinges has a thickness of 1.59 mm and a length of 30 mm, so the displacement allows the material to remain safely in the linear zone. Based on the previous considerations the architecture of the active fixture selected has been the third: basically it provide the less coupled solution and any side issues, such as the non-linearity and possible rotation, could be solved during the detailed design phase.

3.3. Axiomatic decomposition of the system

Based on the choice of architecture for the active workpiece holder, a decomposition of the FRs and DPs for the system has been carried out. This is reported in Fig. 5.

It is interesting to notice how the system decompositions allows the optimal selection of the sensors type and their positioning. At the forth level of the DMs is possible to define also the logic of the control algorithm that must be implemented.

In the decomposition tree, the components have been reported just considering their type and functions. Next step is the definition of the components detailed characteristics. For this selection have been useful the preliminary experimental

tests carried out to characterize the problem. In particular, frequency bandwidth of actuators and sensors has been defined in agreement with the expected chatter frequency value (i.e., around 0-4 kHz range would be adequate). For the sensors we have an ample choice given that the frequency that must be measured is usually assured by most of the commercially available accelerometers. Machine tool environment, with the presence of lubricant, coolant and sharp metal chips, represents, in this specific case, one of the major constraints in component selection; armored versions must hence be chosen.

Maximum measured cutting forces have been used for the selection of suitable actuators; operatively, indeed, the actuator should be capable of withstanding and overcome cutting forces to generate adequate counteracting workpiece displacements. For the actuators, a couple of piezo having maximum force and dimension compliant with the specification has indeed been chosen: this is a market available solution by Physik Instrument. The control logic must have a high feedback rate and must be integrated with the active workpiece holder spending the least effort. To satisfy these requirements an embedded I/O and control logic equipment has been chosen: this is a compact programmable control logic and signal processing produced by National Instruments (this is a compact I/O device that has an internal control logic that could be programmed also in real-time environment thanks to a resident FPGA chipset). The final design of the solution is reported in Fig. 4.
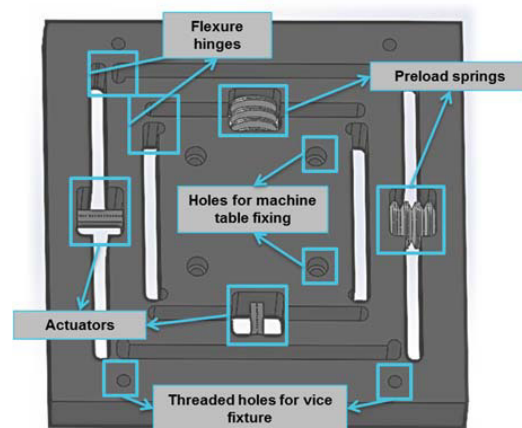

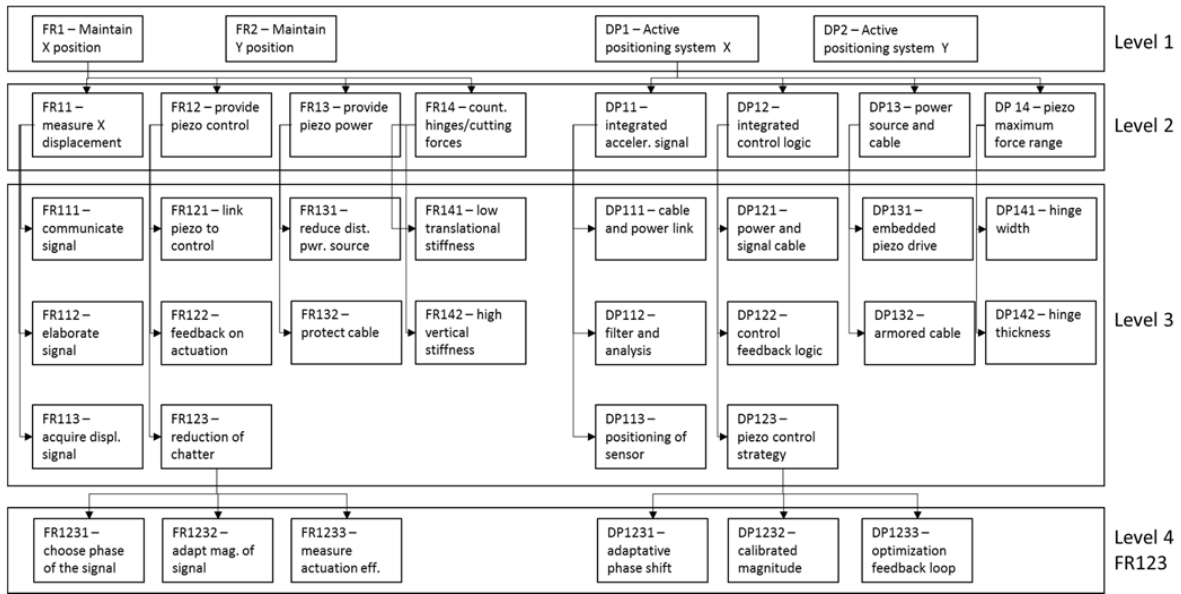Fig. 4. Design of the active workpiece holder

Fig. 5. Part of AD decomposition of active workpiece holder

3.4. Detailed design of the active fixture

After the definition of the general architecture of the product in needed to define the design parameters of the system. In particular a critical design issue for the active workpiece holder is the flexibility of the hinges and the dynamic response of the system that is strongly related to their dimensioning. To validate possible dimensioning of the hinges a FEM analysis has been carried out on the specific components and on the whole system in order to optimize the range of application of the active system. An example of the FEM of one configuration of the hinge is reported in Fig.6.

Thanks to FEM model has been possible to create a relation between the FRs and the DPs. Using a simulated test plan has been possible to measure the performance of the system for different configuration, and to interpolate the results to create a model. One example is the modeling of the hinge stiffness, that it is a very critical design variable. For example, the model of the hinge stiffness has been used to evaluate and define the dynamic behavior of the active workpiece holder, that has to be compliant with the expected performances. This model is reported in Fig. 7 together with the equation (1). This equation could be used for the correct dimensioning of the hinges design parameters, including it within the design matrix at level FR14-DP14.
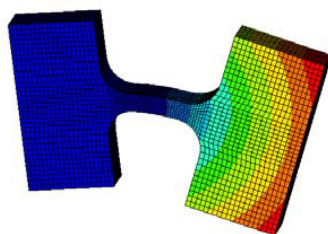

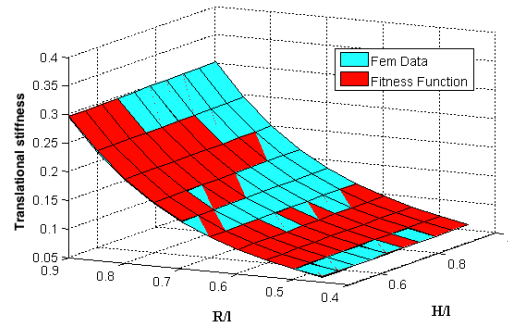Fig. 6. FEM analysis of the flexure hinges

Fig. 7. Model of the translational stiffness

$$K_Y = a_0\left(\frac{R}{l}\right)^3\left(\frac{t}{l}\right)^2 + a_1\left(\frac{R}{l}\right)^3\left(\frac{t}{l}\right) + a_2\left(\frac{R}{l}\right)^3 + a_3\left(\frac{R}{l}\right)^2\left(\frac{t}{l}\right)^2 + \ldots$$
$$a_4\left(\frac{R}{l}\right)^2\left(\frac{t}{l}\right) + a_5\left(\frac{R}{l}\right)^2 + a_6\left(\frac{R}{l}\right)\left(\frac{t}{l}\right)^2 + a_7\left(\frac{R}{l}\right)\left(\frac{t}{l}\right) + \ldots \quad (1)$$
$$a_8\left(\frac{R}{l}\right) + a_9\left(\frac{t}{l}\right)^2 + a_{10}\left(\frac{t}{l}\right) + a_{11}\left(\frac{H}{l}\right)$$

As a result of the design process, the dimension of the hinges are: thickness of the hinge 1.59 mm, length 20 mm and thickness of the base plate 20 mm. The most influent parameter on the stiffness is the hinge thickness, so it is necessary to assure that this value would be as precise as possible in order to grant that no rotation arise when actuating the workpiece holder. In order to assure this goal, the process planning of the workpiece has been studied. To create a similar feature is necessary to use a very thin end mill, at maximum with a diameter of 8 mm and a cutting length of 20 mm. Some experimental tests have been carried out with such tooling in order to assess the correct process parameters for obtaining a

good precision. For the tests have been used a tooling produced by Hoffmann on a Mori Seiki milling machine (MNV1500 DCG). At the end, with the selected parameters, it is possible to obtain a precision on the thickness of the hinges, measured thanks to a micrometer, of less than 20 microns. This is an error of 1.2% respect to the required dimension. This error is considered compliant with the required precision to assure the absence of rotation. Since it is not an easy task to create a relation model between the machining tolerances and the process parameters adopted, the definition of the machining parameters has been verified ex-post from the result of the experimental machining tests. The threshold that we consider acceptable for the product is an error of 5%. The selected process has been selected because it allows some safety margin between the specification and the expected result.

4. Discussion

The AD has provided a good framework to create a reliable model for the active workpiece holder. In this case two levels of application have been used: the definition of the architecture thanks to a DM filled with 0s and Xs and the optimal choice of the design and process parameters using a mathematical model of the product. The DM in the second case is complex due to the equation ruling the physical behavior of the system. In this case has not been possible to obtain the system equation analytically but a design of experiment test plan with FEM analysis has been used. This has been an interesting test about how to integrate the AD with a set of different modeling and design tools, such as FEM and analytical modeling.

5. Conclusions

This paper reports the application of the AD to the design of an innovative system to reduce the influence of chatter vibrations in milling operations. The AD has shown its capability in designing an efficient structure to control the phenomena and choose and dimension correctly the system's components. The final dimensioning has been carried out using a FEM approach that allowed a correct sizing of the holder framework, compliant with the static and dynamics specifications.

Acknowledgements

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