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A MECHATRONIC STUDY ON A MODEL-BASED COMPENSATION OF INERTIAL VIBRATION IN A HIGH-SPEED MACHINE TOOL

A significant limitation to machine tool productivity in high-speed operations is due to inertial vibrations. During strong accelerations, inertial forces generate oscillations that are translated into surface geometrical errors on the machined parts. Machine tools users minimize these problems by reducing machine axes quickness, thus affecting productivity. In this paper the effects of inertial deformations on machine tool accuracy have been studied to evaluate the possibility of adopting a software compensation strategy. The proposed model-based solution, based on a reduced model of the machine tool dynamics, has been tested in a mechatronic simulation environment. In order to meet industrial needs, the compensation scheme has been designed to allow its implementation on standard numerical controllers. A detailed feasibility analysis has been carried out studying all aspects that can potentially impede the application of such approach. Simulations and some preliminary experimental tests prove the effectiveness of the developed technique on a five-axis machining center.

1. INTRODUCTION

In the context of today's production environments, machining centers are increasingly required to be accurate, fast and flexible at the same time. In terms of accuracy, machines must be capable of reaching precise and repetitive axes positioning and withstand to high cutting forces during machining. On the other hand, powerful servo-drives must assure high [2]axes velocities and accelerations during both rapid and feed machining phases in order to reach short cycle times. In this scenario inertial forces, that act on the machine tool structure during motion, represent a serious limitation for the machine precision and productivity and therefore must be taken into account during machine design and controller tuning. Structural deformations due to inertia in fact may lead to unwanted displacements of the Tool Center Point (TCP), causing undesired footprints that degrade surface quality during finishing [13]. Realization of corners, sharp edges, S-curves during high-speed

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milling operations (e.g. in alluminium components for the aeronautical field) and the execution of a tool disengagement in boring operations are some of the technological features where inertial deformations exacerbates leading also to scraps and non-compliant parts, increasing costs and reducing production rates.

From the structural design point of view, large and heavy machines usually undergo larger structural deformation, for a given acceleration. More in detail, bigger machines tend to oscillate at lower frequencies during motion, requiring therefore an accurate smoothing of the motion, as discussed in the following paragraphs, while for smaller and more rigid machines this aspect is less important since machine resonance frequencies are higher and not easily excitable during motion. The machine architecture is also very important for the achievement of good dynamic performance: for example C-Shape and Box-in-Box machines are less prone to inertial errors than columnar and portal machines. However this article does not address these issues but indeed focuses its attention on the control strategies.

From the controller point of view Computer Numerical Controls (CNCs) made significant progress on vibration reduction, thanks both to fast servo-drive and powerful electronics. For example, Siemens Advanced Position Control (APC®)[17] damps mechanical vibrations closing an additional velocity feedback loop on the linear position scale, reducing therefore high-frequency oscillations involving the axis mechanical transmission. Other interesting examples are Siemens Dynamic Stiffness Control (DSC®) [17], Fidia Active Tuning [18], Fidia Active Damping [18]. It is important to notice that these functions can only deal with vibrations that are observable by the drive measuring systems, while they are insufficient in cases, such as inertial vibrations, where a distributed deformation involves the whole machine structure. Indeed for these cases, smoothing techniques play an important role in the industrial practice, thanks to theirs robustness and effectiveness in allowing good positioning performances. The concept of these open loop techniques is to limit the spectrum of the motion forces in order to limit the excitation of the machine natural modes. In literature they have been extensively studied [14]. One of their main drawbacks is that they imply a limitation to the achievable production rate in terms of cycle times.

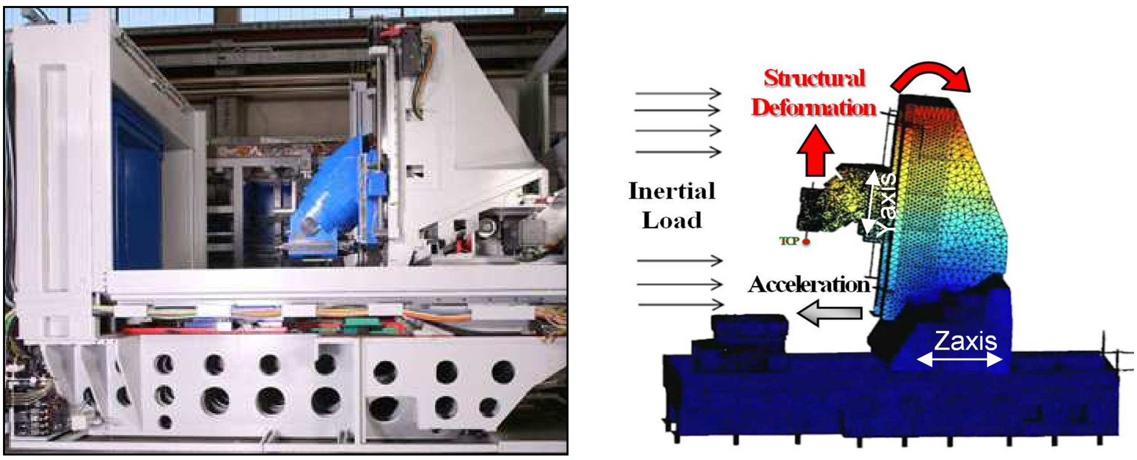

Fig. 1. The machining center Mandelli SPARK1600® and the related Finite Element Model

A typical example are the Jerk Control functions which generates smoothed acceleration commands limiting their first derivative. These functions allow an easy matching of the control response to the machine dynamics and therefore are usually implemented in the industrial NC Units. Command Shaping techniques [15] are similar but more complex since the reference commands are shaped accordingly to the machine dynamics (that must be known precisely).

In order increase the robustness of the vibration control systems, it is also possible to exploit feedback actions based on additional sensors and actuators implementing additional closed loop control [8,9,10,11]. Despite their better performances, demonstrated in an extended scientific literature, these kind of advanced solutions are rarely adopted in the machine tool industry.

In this paper the applicability to industrial machine tools of a model-based strategy to mitigate the effect of inertial vibrations has been studied. The proposed compensation filter is able to estimate the oscillation at the Tool-Tip starting from the motion reference commands on axes and to compensate the estimated error acting on the reference command of the compensating axis.

This software compensation scheme has been conceived to be compatible with classic controllers of industrial machines thus avoiding the requirement of additional hardware and sensors. Particular attention is hence given to the feasibility analysis that takes into account all phenomena that can potentially impede the application of such approach.

2. CONTROL AND STRUCTURE INTERACTION

A large machining center produced by Mandelli Sistemi (Fig.1) has been studied. It has five controlled axes (two of them are moved by two motors in parallel, in gantry configuration) that assure accelerations up to 4 m/s^2. In order to verify the feasibility of the proposed approach, a detailed mechatronic model, developed in the Matlab/Simulink® environment [1],[2],[3],[4], has been exploited (Fig. 2).

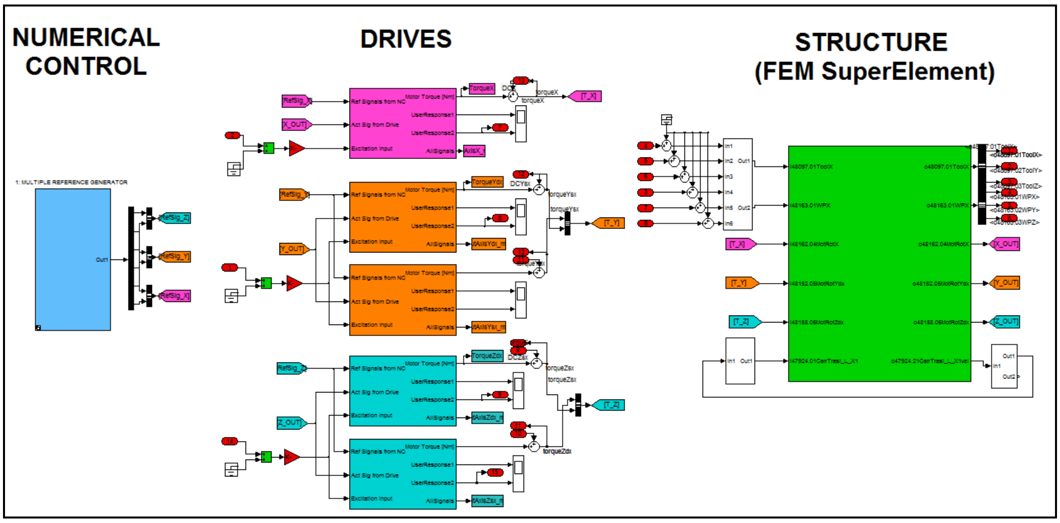

Fig. 2 The Integrated Mechatronic Model in Matlab/Simulink®

The integrated model is composed by a FEM Block of the mechanical structure (created exporting from a commercial finite element package a reduced order model obtained by the Craig-Bampton approach [16]), of Drives Blocks (containing standard cascade control structure inspired by the Siemens Simodrive 611D) and a Numerical Control Block (able to generate jerk limited and more complex position commands).

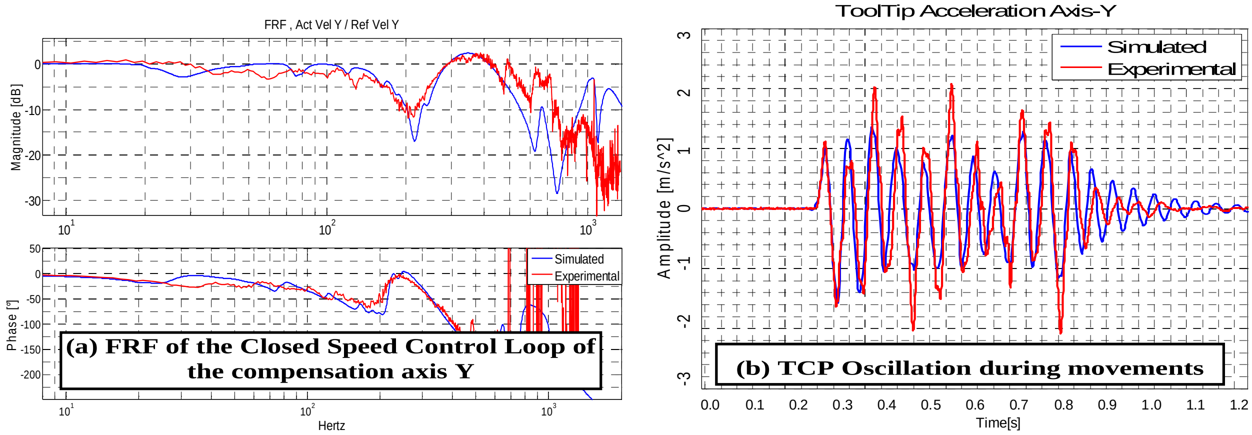

Fig. 3. Comparison in frequency and time domain between simulated and experimental data

Good and reliable simulations of the tilting effect have been attainted with a continuous-time version of the integrated model (Fig.5c). The good agreement with experimental data shown in Fig.3 has been achieved improving the model with additional drive features, such as Velocity Feed Forward and various Reference Filtering blocks.
When the structure, around 5 tons, undergoes high accelerations along the Z-axis, large and distributed inertial forces are generated. The structure deformation can be decomposed in a static component, proportional to the instantaneous acceleration, superposed to a dynamic component coming from the excitation of the vibration modes (Fig. 4a).

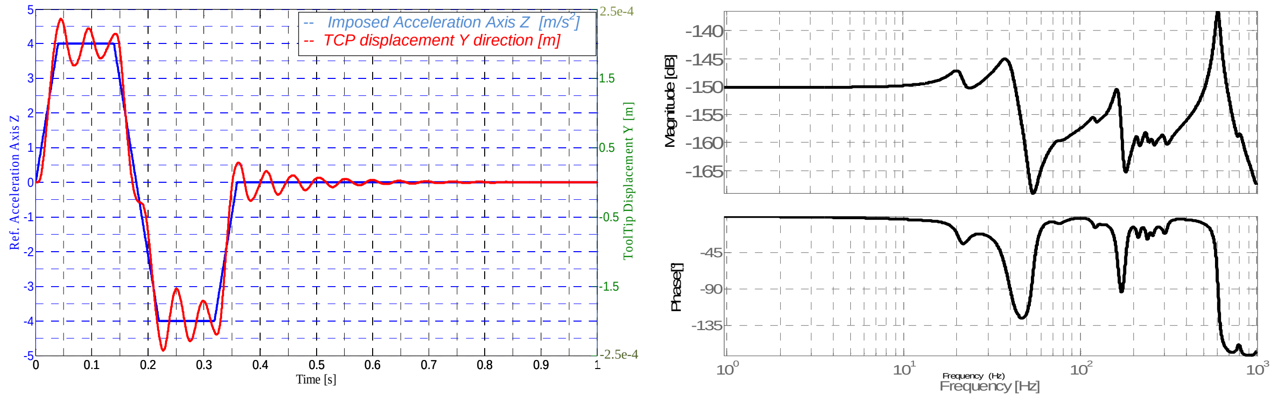

Fig. 4. Simulation of Inertial Deformation and FRF (Displacement/Force, Y direction) at TCP

Combining a typical acceleration spectrum (Fig. 5b) and the machine frequency response (Fig.4b) it can be pointed out that for the machine under study the dynamic response is dominated by the first structural resonance. The Tool Center Point (TCP) position after the forced action shows a damped decay typical of a single degree of freedom mechanical system. Considering the simple case of a jerk limited step along Z (while the Y position reference is kept constant), the absolute oscillation trajectory of the TCP (Fig.9) basically corresponds to a line in the ZY plane. From the technological point of view the vertical component of the oscillation (along Y) is typically the most harmful effect while the horizontal errors, parallel to the part surface are less important. The vertical oscillation increases almost linearly with the Z-axis jerk value even though the Y-axis control is able to nullify the position error measured at the linear scale, i.e. maintaining the spindle head at a fixed position along the Y-axis guideways (Fig. 5c).

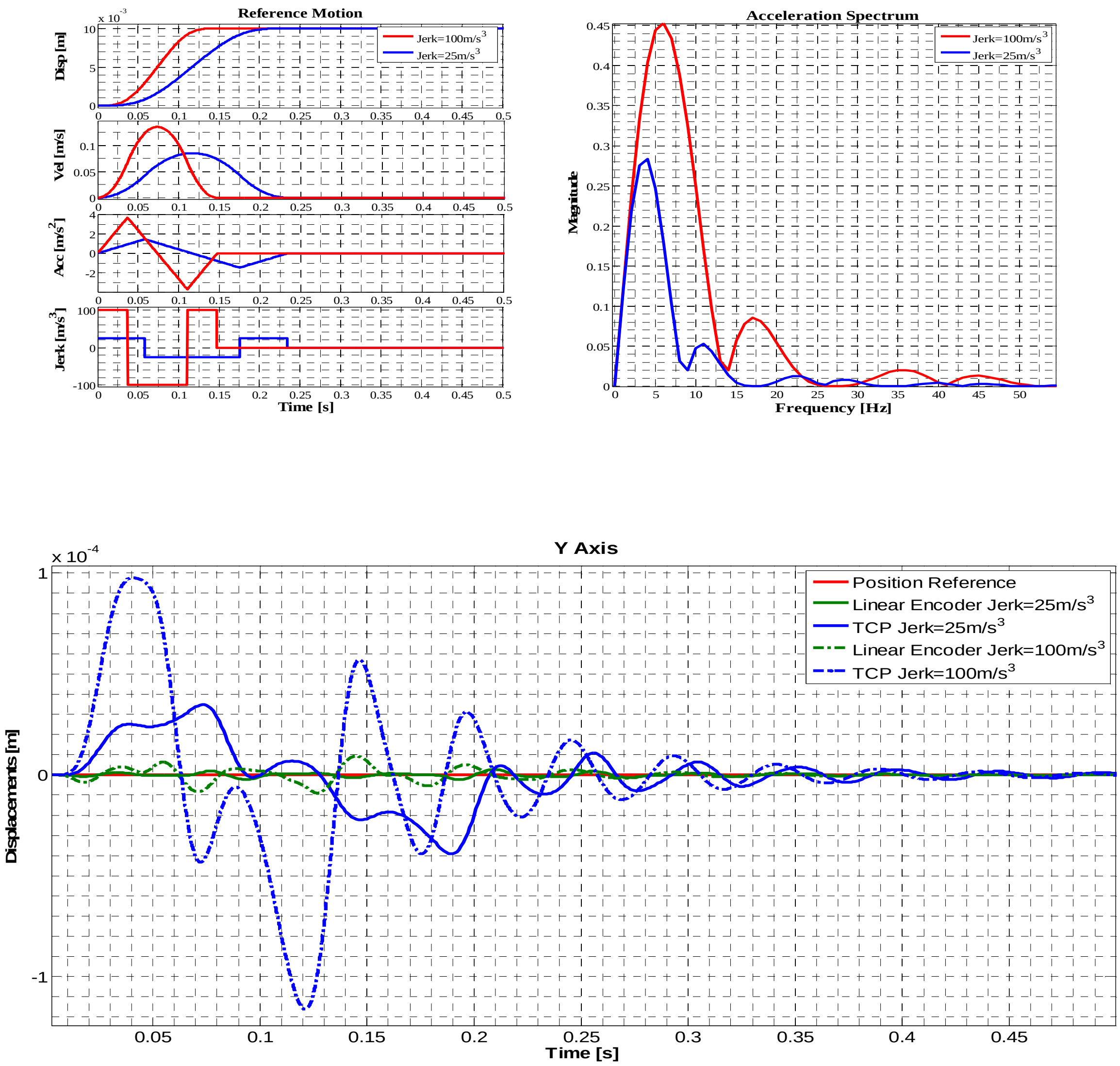

Fig. 5. Jerk limited positioning commands (Amplitude=10mm, feed=10m/min, acceleration=4m/s^2) on Z-axis and related TCP displacements on Y-axis (Simulation)

3. COMPENSATION STRATEGY

A “Tilting Filter” has been designed to forecast the system dynamic response at the tool tip and then use the standard machine controller and actuators to perform a feed-forward compensation. With this approach it is possible to reduce the tracking error in the most sensible direction, without slowing down the machine as done by other motion smoothing techniques, thus allowing shorter cycle times and better machine accuracy. As no additional hardware is required, the application of the strategy in a real system is particularly easy (see par.4.2). The feasibility of the presented approach is mainly related to the high dynamical capabilities of the compensation axis which has to perform the fast motions required (see par.4.1) and to the correct estimation of the TCP vibration by the adopted model [5]. The choice of the model complexity have been addressed taking into account different aspects. For the studied case a single DOF mechanical system (Fig. 6), forced by the inertia associated with the movement of Z-axis, is adequate to reproduce the oscillation (Fig.10). The model response, according to the real system, consist of a dynamical change of the Y coordinate lead by the three fundamental filter parameters as Frequency, Damping and Gain. The simulations prove that a Single-Input Single-Output (SISO) realization of the filter with the reference Z velocity as input and the estimated TCP Y position as output is suitable. Indeed the interaction effects of the compensation action on the oscillation are negligible for the adopted level of jerk: for higher levels the compensation motion generates an additional error in Y that has to be taken into account and compensated for, requiring a Multi-Input Multi-Output (MIMO) approach.

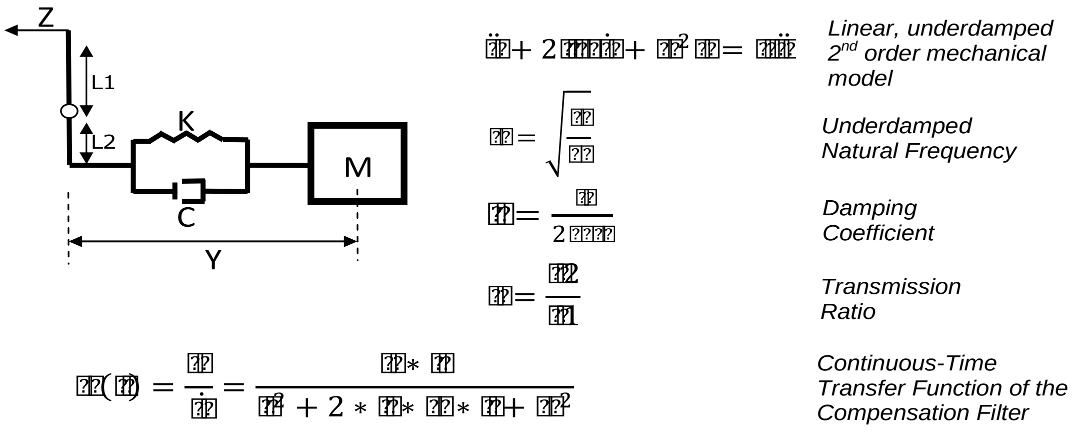

Fig. 6. Compensation Filter as a Single DOF Damped Mechanical System

The SISO model with 1-DOF in general supplies good estimates of the TCP Y position until the actual velocity of the Z-axis (which feeds the machining process) accurately follows the reference command generated by the interpolator (which feeds the Tilting Filter). An uncorrect tuning of the Z-axis controller may infact lead to velocity tracking errors, as delays, overshoots and oscillations which impact on the system behaviour thus limiting the prediction capacity of the filter [6]. In those cases it would becomes necessary to use the actual Z velocity as the filter input, implying a feedback approach or, more remarkably, to include in the filter new complex poles that describe the Z velocity control loop dynamics. In simulation this latter approach has not brought the desired benefits and therefore, considering the additional complexity of the filter, has been neglected. The approach based on the actual Z-axis velocity generates timing problems in the implementation on the industrial controller, due the to computation delays, as explained in section 4.

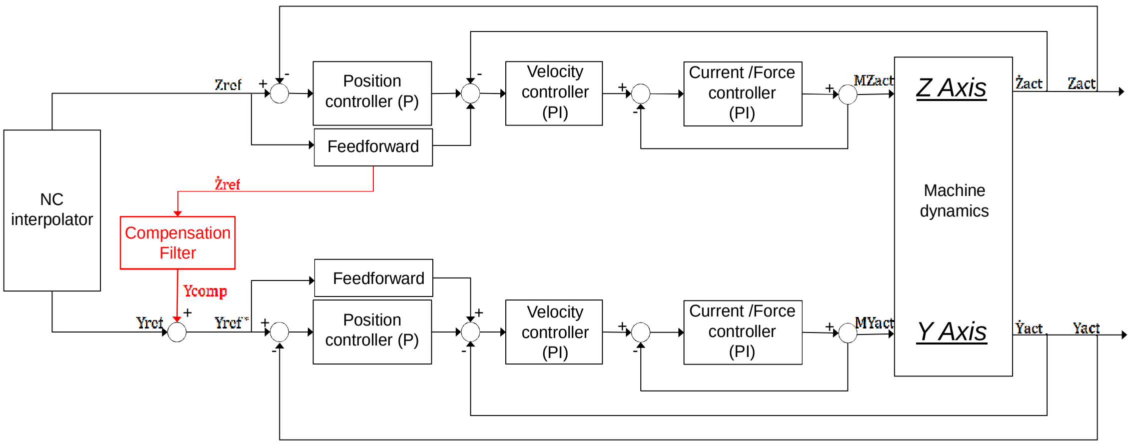

Fig. 7. Concept scheme of dynamic feed-forward Tilting Compensation Filter

The robustness of the solution can be improved taking into account the variation of the machine dynamic for different positions in the work-volume. For example the first resonance, mainly involved in tilting, varies about ±8% in frequency and ±14% in amplitude for different positions along the Y-axis stroke, due to the considerable mass of the spindle head. This issue could be tackled by different approaches (like gain scheduling [12]): in the present paper, devoted to analyse the applicability on an industrial controller, the filter parameters have been considered as constants.

Another relevant aspect that is worth to mention is related with dynamic nonlinearity. Friction in mechanical trasmission as well as in the guideway carriages can considerably affect the acutal motion. This aspect, that is not modeled, can plays some role on the axis behaviours, expecially with small amplitude oscillations: in the final part of the step response both the complete mechatronic model (Fig.3) and the simple TCP observer (Fig.10) cannot fit well the experimental signal. However, considering the small absolute error that is committed in this phase, modellling of the friction, typically quite complex, has not been included in this version of the Tilting Compensation Filter.

4. FEASIBILITY ANALYSIS

To evaluate the applicability of inertial oscillation compensation on different machine tools it is necessary to evaluate both mechanical and control aspects. Only machining centers with a dominant vibration mode at low frequency (<50Hz) can be considered willing to allow the estimation of TCP oscillation with a simple compensation filter and the use of a machine axis to perform the compensation motion. This requirement is normally met by medium-large size machines with moving mass above 2 or 3 tons and with columnar configuration. The dynamical potential of the compensation axis is another important feature that must be assessed to garantuee the requested tracking performance. Generally today's drives configuration as gantry or tandem in conjuction with the absence of hydraulic supporting system of the machining head can satisfy this need.

4.1. LIMITS ASSESSMENT

To succeed in the compensation no limit of the compensation axis must be activated by the compensation command. Limits related to bandwidth and saturations have been identified and formalized quantitatively through the use of the mechatronic model: they are influenced respectively by the structural dynamical behaviour and by the considered motion quickness (Fig. 8).

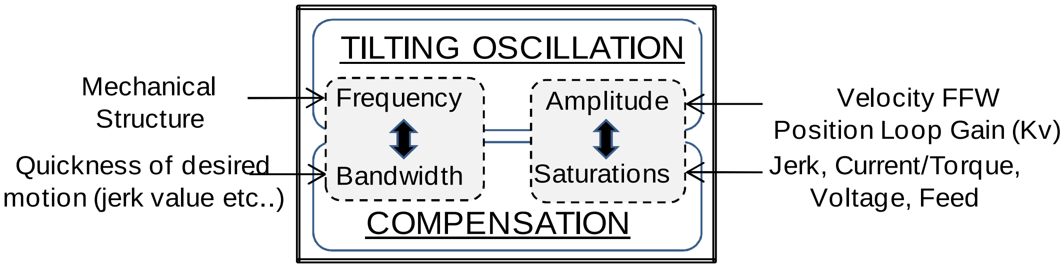

Fig. 8. Parameters to be considered in evaluation of limitations of the compensation motion

The frequency content of the compensation command is related to the dominating mechanical resonance excited by the spectral content of the Z-axis command. Obviously this parameter may vary with the position of axes X and Y but in this case it was always at about 20Hz. Simulations prove that good compensations are obtained if the position loop bandwidth of the Y-axis is higher than the resonance value by at least 2.5 times. For the studied case the required bandwidth can be achieved only using the Velocity Feed Forward function “FFW”, already available in the real controller, and with an appropriate tuning of the velocity loop. The FFW infact plays a key role to ensure the required tracking performance, more than the position loop gain. A second analysis has been performed to verify all possible drive saturations as jerk, torque/current and velocity (Table 1). For this assessment highly demanding compensation cycles have been considered (Fig. 5a): 10mm longitudinal motion steps in the Z direciton with reference jerk of 25m/s^3 and 100 m/s^3 respectively. These motion steps are assumed to be continuosly repeated after the end of every induced oscillation: hence for the calculation of the required RMS torque, which must verify the thermal S1 limit characteristics of the Y motors, a period of 0.5s has been used (Fig. 5c). Instead the jerk limit has been calculated considering only the phases inductance of the axis motors, because the torque variation induced to the axis (that is basically proportional to the jerk) is generated by a current variation on the motor phases, limited by the combination of the available bus voltage and motor inductance. For the tested Mandelli Machine none of these restrictions affects the compensation capacity.

Table 1. Saturation Limits of the compensation during motion steps with jerk= 25m/s^3 and jerk= 100m/s^3

Parameter	*Max Value*	*Units*	*Required Value J25 / J100*	*Saturation level J25 / J100*
Jerk (electrical)	25088	$\left[\frac{m}{s^3}\right]$	28 / 128	0.11% / 0.51%
Torque (Peak)	90	$[Nm]$	0.69 / 3.15	0.76% / 3.5%
Torque (Rms_0.5s)	46	$[Nm]$	5.46 / 27.47	11% / 59%
Velocity	50	$\left[\frac{m}{\text{min}}\right]$	0.077 / 0.33	0.15% / 0.66%
Position	1400	$[mm]$	0.032 / 0.093	0% / 0%

4.2. EXPERIMENTAL PARAMETERS IDENTIFICATION

One of the most important feature of the compensation strategy is its generality and easy applicability to different machines. While the feasibility of the proposed approach has been studied by the mechatronic model, the practical identification of the Tilting Filter has been fully based on experimental procedures, to avoid the need, in industrial applications, for complex models and to get the best accuracy.

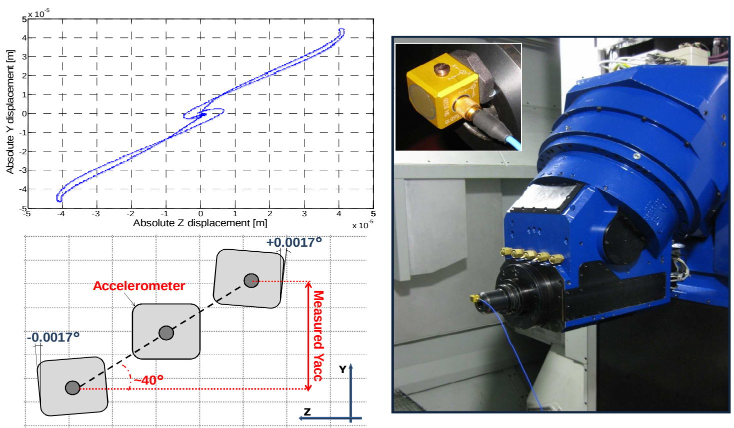

Fig. 9. Simulation of the absolute TCP oscillation trajectory and experimental setup with an accelerometer

The experimental identification has been done placing an accelerometer near the machine TCP, to avoid more complex setups with geometrical references and displacement sensors. The measured acceleration has been processed, to minimize drift due to numerical integration, obtaining a proxy of the TCP displacement, in the Y direction. The rotational effect on the sensor are insignificant (as proved in simulation, Fig.9), and hence the absolute TCP vertical displacement can be estimated from the single sensor channel which is oriented in vertical direction.

Some jerk limited step motions have been performed in the Z direction acquiring both the accelerometer signal and the Z-axis velocity reference command generated by the NC unit. A minimization function, using the simplex search method, is applied on the fitting error in post-processing in order to find the optimum filter parameters values that allow the best fit between the estimated ant the actual TCP trajectory.

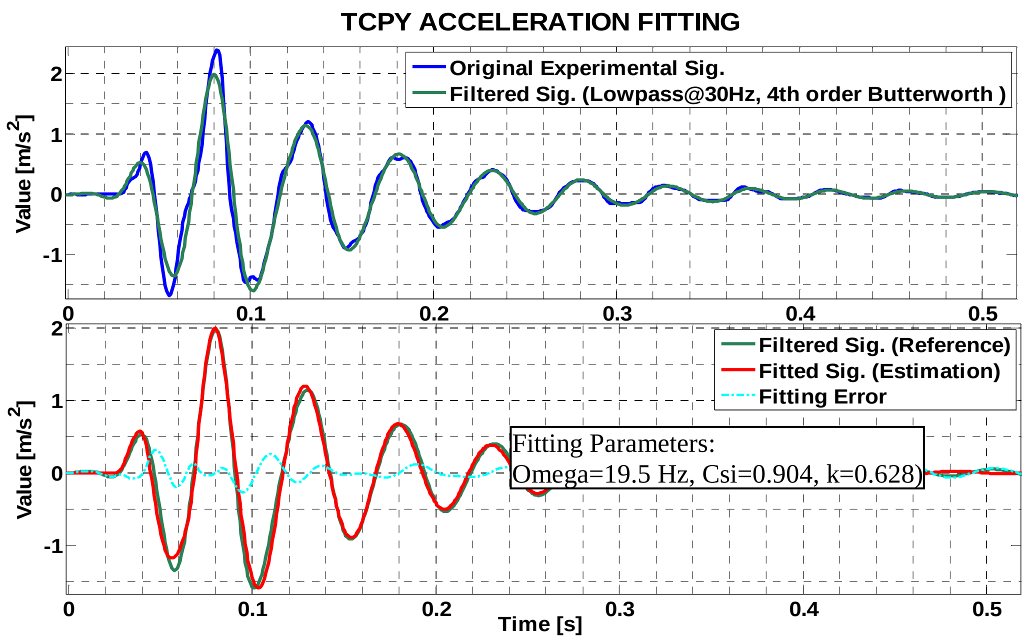

Fig. 10. 1 DOF Acceleration Fitting on the Experimental signals (Amplitude Z=1mm, Jerk Z=100m/s^3)

4.3. IMPLEMENTATION ON THE REAL SYSTEM

Another important feature of the developed strategy is its full integration with the Numerical Control of the machine. It is often possible to exploit the customizability of Industrial machine tool Numerical Controls (e.g. Siemens 840D), to implement the described algorithm into the NC Unit. Special functions called “Sinchronous Actions” [17] allow the user to realize a real-time elaboration, sinchronous with the interpolator (IPO) cycle, of a digital compensation filter (FIR) to create the feed-forward command.

During some preliminary tests on the real machine control the following effects have been investigated [Table 2]:

Factor	*Correlation*	*Investigation*	*Relevance*
Computational Effort required	NC performance	FIR Tilting Filter ask for a very limited computational power in the 1-DOF version. More challenging is the case where small IPO times, multi-axis machining cycles with high complexity and complex auxiliary functions are used togheter with the compensation.	**Low**: usually doesn’t limit performance
Computational and communication delays	Digital control, NC-Drive communication	Processing of the Compensation Synchronous Actions is done acting on the reference vel. signal and introduces some delays. This is not a problem since all the ref. signals, sent to the drives, are delayed by the same amount.	**Medium**: In the studied case the introduced fictitious delays do not limit the compensation at all. However in other cases this can represent a serious problem, expecially for strategies closing loops on measured signals [7].
Discretization of the compensation waveform	Digital control, NC numeric resolution	A high IPO sampling time togheter with a low NC numerical resolution can lead to bad compensation command due to discretization errors, depending on the amplitude and frequency of the compensation waveform.	**Low**: for all tested cases this has not represented a problem. In other cases this would imply a limit of the compensation amplitude and then a limit on the Z jerk command.

Table 2. Factors of the real industrial controller that can disturb the compensation strategy

5. SIMULATION RESULTS

Several simulation tests have been performed to evaluate the performance of the compensation algorithm, using a continuos-time filter that do not consider any system delay.

In Fig. 11 different combinations in terms of amplitude and jerk value of the longitudinal motion step are compared. Both cases show how the Y displacement amplitude at TCP level decreases drastically with the compensation action. In Fig. 11b the counter-action observed by the drive sensor (linear encoder) is perfectly superimposed to the reference command (which is nearly the opposite of the un-compensated TCP position) thanks to promptness of Y-Axis. It can then be asserted that the residual TCP oscillation is mainly caused by unavoidable modelling errors.

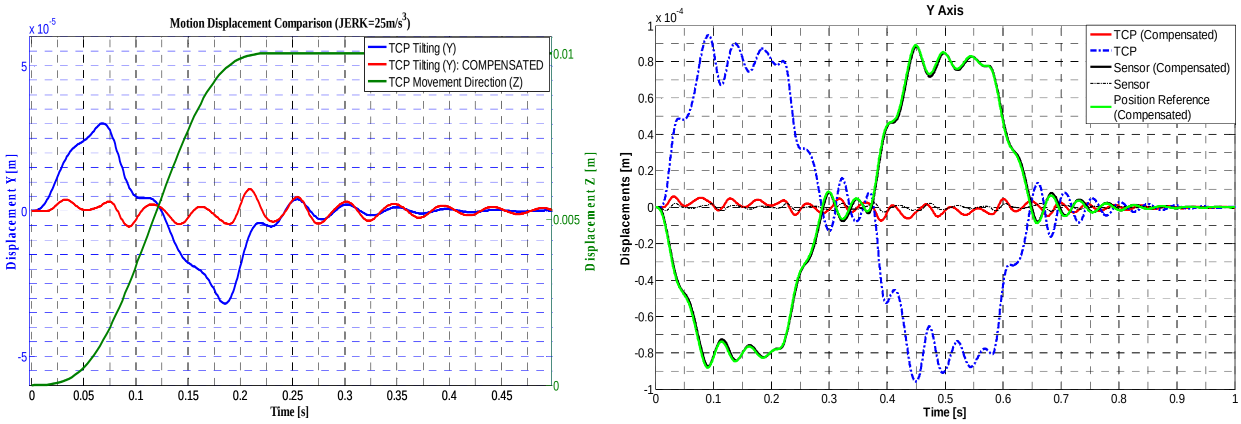

Fig. 11. Tilting compensation for different motion commands on Z-Axis

In the first case (Fig. 11a: Amplitude=10mm, Feed=20m/min, Acceleration=4 m/s^2 and jerk=25 m/s^3) the attained improvement is about 80% while in the second case (Fig. 11b: Amplitude=300mm, Feed=20m/min, Acceleration=4m/s^2 and jerk=50m/s^3) is about 93% due to the fact that higher motion amplitude means longer acceleration time and therefore greater static components (see also Fig.4a) which are easier to compensate than the dynamic contributions.

In addition, a test along a square trajectory in the XZ plane is proposed in Fig. 12. The different jerk values, used to execute the reference trajectory, affect both the cycle time and the TCP displacement along Y. It has been considered only the negative Y direction because, in face milling, the workpiece surface is irretrievably ruined by vibrations only when the tool plunges into the machined surface, going in the negative Y direction.

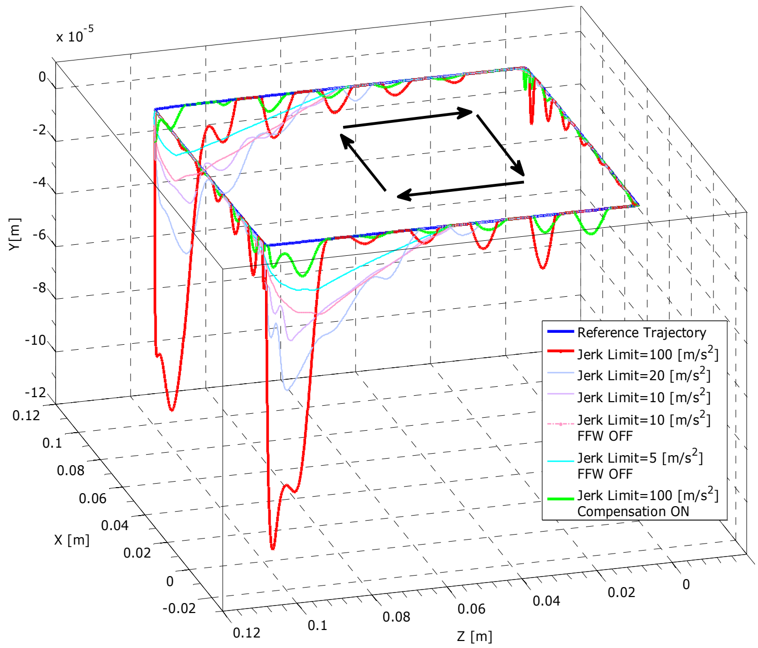

Fig. 12. Tilting Compensation for Square Trajectory on ZX Plane. (L=100mm, Target Acceleration=5m/s^2)

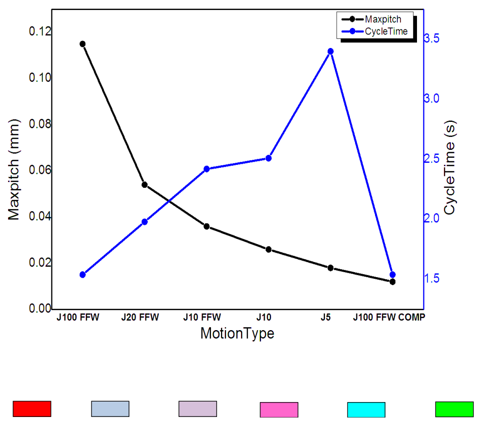

Fig. 13. Time and Pitch Error reduction permitted by the Tilting Compensation of Fig. 12

Considering a maximum pitch of about 0.02mm, the cycle time can be reduced significantly (more than 55%) activating the compensation (Fig. 13). More complicated trajectories and machining setup will be tested on the real machine, taking into account technological and other practical aspects.

6. CONCLUSIONS

The paper presents a study on the active compensation of machine tools inertial deformation based on a feed-forward controller, that can be implemented on the standard controllers and drives. An exhaustive feasibility analysis has been conducted on an integrated mechatronic model of a five-axis machining center and partly on the real system. Simulations have shown the effectiveness of the solution: more than 80% of reduction in the TCP oscillation and respectively more than 55% in cycle time saving have been achieved. The proposed approach can be adopted for all machines that have a simple vibrational behaviour with one or few dominant resonances at low frequency and that have a compensation axis with powerful motors and high dynamic performance. The proposed strategy is now being implemented on the studied machine to test the achievable performance in presence of all the uncertainties that characterize a real systems. If such trials will be satisfactory, it will be necessary to evolve the proposed feed-forward approach to take into account the unavoidable variability of the machine dynamic behaviour in the workspace exploiting also recursive identification methods for on-line system identification.

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