ECCOMAS Thematic Conference on Multibody Dynamics
July 24 – 28, 2023, Lisbon, Portugal

Integration of Flexible Multibody Systems Dynamics and Virtual Commissioning Simulations of a Machine Tool

Paolo Gioviti[1], **Alberto Martini**[1,2], **Nicolò Vincenzi**[3], **Marco Troncossi**[1,2]

[1] Center for Industrial Research
ICIR MAM
University of Bologna
Viale del Risorgimento, 2, 40136,
Bologna, Italy
paolo.gioviti@unibo.it

[2] Department of Industrial Engineering
University of Bologna
Via Fontanelle 40, 47121, Forlì, Italy
[alberto.martini6@,
marco.troncossi]@unibo.it

[3] GIULIANI
Bucci Automations S.p.A.
Via Granarolo 167, 48018, Ravenna, Italy
n.vincenzi@bucci-industries.com

ABSTRACT

The digitalization of production processes is a crucial aspect of Industry 4.0, where digital simulations are increasingly utilized to develop and test new ideas to enhance competitiveness in the marketplace. Digital Twins, which replicate the functioning of complex devices, machines, and industrial plants, can optimize mechanical design, virtual implementation of control procedures, real-time operations management, and in-service failure predictions. The application of Digital Twins for elastodynamic simulations can be highly relevant in the field of machine tools, which are often susceptible to vibrations and dynamic loads that can adversely affect their operative performance. Such simulations can help to optimize mechanical design and properly tune control programs to enhance machining precision and guarantee high quality standards. Another potential application of Digital Twins is Virtual Commissioning that enables part programs verification on virtual machine models, regardless of the availability of the physical systems (differently from the traditional procedures of control testing), thereby minimizing the occurrence of critical issues in working conditions. The combination of advanced simulations of mechanical phenomena and part program virtual verification is supposed to provide a comprehensive approach for overall analysis, design optimization, and working efficient management of automatic machines. Nowadays, to achieve such a goals, a number of challenging tasks must be afforded step by step. This paper focuses on the development of a flexible multibody model of a machine tool to analyze its mechanical behavior and dynamic response under different operating conditions. The model predicts the elastodynamic behavior of the machine at high speeds, taking into account the compliance of flexible parts and contacts. The simulation results are to be subsequently implemented in a Virtual Commissioning software environment for an integrated virtualization of the mechatronic system.

Keywords: Elastodynamic simulation, Digital Twin, , Virtual Commissioning, Mechatronic system, Machine tool.

1 INTRODUCTION

In the era of Industry 4.0, the employment of digital simulations is progressively surging to develop and test concepts intended to enhance companies' competitiveness within the marketplace. To address this issue, Digital Twins (DTs), virtual replicas of real systems, are utilized to emulate the overall functioning of complex devices, machines, and even industrial plants. Machine tools represent a perfect test bench where conceiving and analyzing methodologies aimed at improving manufacturing processes. As complex mechatronic systems, machine tools play a crucial role in manufacturing industries, and their optimization is necessary to improve product quality and production efficiency. Accordingly, DTs of machine tools can be developed as an effective tool for supporting the entire design and implementation process of machine tools. From the early-stage optimization of mechanical design and virtual implementation of control procedures [1,2] to real-time operations management and in-service failure predictions [3], DTs have the potential to revolutionize the machine tool industry. However, highly advanced techniques of modeling and simulation are not widely used in machine tools yet, due to the significant effort required in the early design stages [4], specific industrial requirements, and – to the best authors' knowledge – the lack of commercial software to integrate the simulation tools required for optimization of both mechanics and automation. Nonetheless, ongoing research efforts and the development of more integrated and user-friendly simulation tools may help to overcome these barriers and to promote wider adoption of advanced modeling and simulation techniques in machine tool design.

Within this context, this paper some preliminary steps of a medium-long term project conceived to propose an innovative methodology that effectively addresses both the mechanical and automation aspects by exploiting elastodynamic simulations and utilizing Virtual Commissioning (VC) simulations, respectively. Both these aspects are treated separately, then integrated sequentially, using purpose-built software whose corresponding outcomes will be exploited iteratively to achieve the desired goal.

In general, elastodynamic simulations are crucial for predicting the dynamic behavior of complex systems and optimizing mechanical design in various engineering fields, including manufacturing. In the context of machine tools, elastodynamic simulations can predict the dynamic behavior of the machine when operating at high speeds, taking into account the compliance of flexible parts and other dynamic phenomena that can significantly impact the machining process.

Regarding VC simulations, they represent an additional means to leverage DTs, enabling, among other things, the anticipation of control logic design phases, the verification of different operating conditions and the prediction of collisions, downtime, and bottlenecks. VC is particularly crucial within machine tools because it allows for the verification of part programs on virtual models of machines before implementation, minimizes the likelihood of encountering critical issues during the actual functioning of the physical system, thus enhancing the overall efficiency and quality of the production process.

2 CASE STUDY

A specific *transfer* type of machining center (Fig. 1) employed for the production of locks' components was used as case study. The transfer machine, designed and constructed to meet the customer's exact requirements, comprises a series of stations responsible for carrying out a sequence of machining operations on workpieces clamped on a rotating table.

Each station houses one or two operative units, which are CNC milling machines featuring a 3-axis design. The operative units have a nearly identical architecture with only slight variations based on the specific machining operation required. The three linear axes feature ball screw servo-drives and rolling bearing slide-rails guides. The Y and Z screws are connected to the actuators' shaft through elastic couplings, whereas for the X-axis a synchronous belt transmission is used.

The use of 3-axis CNC milling machines offers exceptional accuracy and precision, with the ability to achieve complex geometries and tolerances. This design is particularly advantageous for high-volume production environments, where consistency and speed are critical.

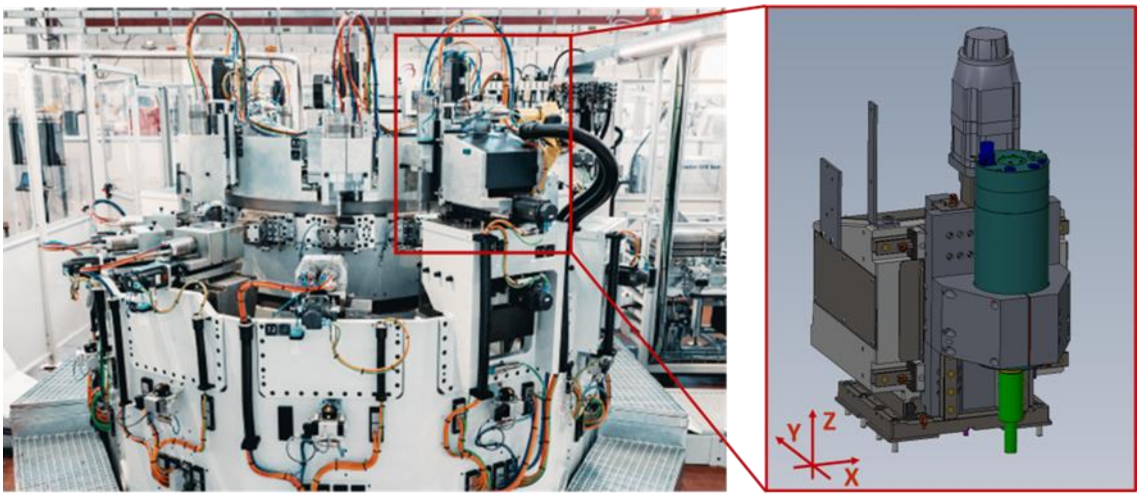

Figure 1. The studied transfer machine and a single 3 axis CNC operative unit.

3 MATERIALS AND METHODS

3.1 Methodology

DTs have garnered considerable attention in the manufacturing industry for their potential to optimize processes and reduce costs. They enable manufacturers to test various scenarios and make adjustments before implementing changes in the physical system, leading to better decision-making and improved performance. However, a complete and effective solution that considers both mechanics and automation aspects of complex mechatronic systems is not available yet. In this study, it has been proposed a methodology that addresses these two aspects sequentially (Fig.2). Specifically, the aim is to develop a flexible multibody model of the transfer machining center described in Section 2 to analyze its mechanical behavior and dynamic response under different operational conditions. The goal is to predict the elastodynamic behavior of the machine and to provide a comprehensive and effective tool for supporting the design process by further combining such model with VC simulations.

Due to the impractical computational burden of incorporating the entire transfer machine into the analysis, a more feasible approach involves focusing on individual operative units. While these units may vary in terms of the specific tool and spindle employed, as well as the length of the Y axis, they all share the same functional components. An elastodynamic model has been developed for the typical operative unit to provide a representative analysis of the system.

This approach enables a more detailed examination of the dynamics within each operative unit, facilitating a deeper understanding of the physical and functional characteristics that distinguish each component. Furthermore, by breaking down the system into its constituent parts, it becomes possible to identify specific areas where optimization or customization could yield significant improvements in performance or efficiency.

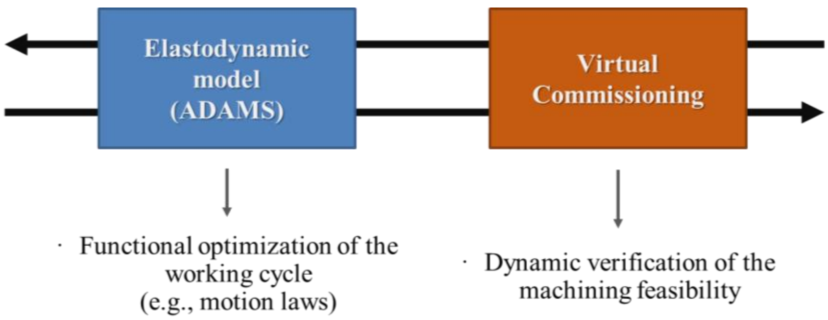

Figure 2. Scheme of the proposed methodology.

A flexible multibody model of the single operative unit was developed through the software ADAMS (Hexagon – MSC Software, USA) to simulate its mechanical behavior, including elastodynamic effects. A validated elastodynamic model is crucial for predicting the dynamic behavior of the machine when operating at high speeds, taking into account the compliance of flexible parts and possible backlash in joints (e.g. resulting from wear). To this aim, some ideal joints have been replaced with general forces to account for stiffness, damping, and the relative positions of the components involved in motion transmission.

Experimental tests have been conducted to validate the model. For such tests, two part programs were used that differ by two parameters: feed rate (F) and maximum jerk value (J). Both programs were written so that each move starts with the specific axis moving from one end of the guide to the other, with equal distance covered in all three axes, according to the following sequence: (1) only the X-axis moves; (2) only the Y-axis moves; (3) only the Z-axis moves; (4) the X and Y axes move simultaneously; (5) the Y and Z axes move; (6) the X and Z axes move; (7) all three axes move. The two part programs were run 10 times each with the following values of feed rate and maximum jerk: the first one with F=15000 mm/min and J=1000 m/s^3 and the second one with F=7500 mm/min and J=500 m/s^3.

Elastodynamic simulations can be used upstream or downstream of VC simulations, as they serve different purposes. Elastodynamic simulations optimize mechanics, while VC simulations assess dynamic feasibility of machining in the overall transfer machine at different points in time, and thus are conducted separately. The methodology can be approached from either directions, as depicted in Fig. 2. Real part programs can be directly employed to test the machine's efficiency, and the effects of motion laws on the elastodynamic model can subsequently be evaluated (VC simulations before elastodynamic simulations). Alternatively, different motion laws can be tested in the elastodynamic model to reduce machine stress in a specific condition and assess whether these modifications allow for the expected cycle time (VC simulations after elastodynamic simulations). For VC simulations, the Eureka software (Roboris, Italy - Fig. 3), which is specifically designed for milling machines, was utilized.

By adding specific command lines at the beginning and end of part programs, Eureka generates a spreadsheet with axis positions at the desired time resolution at the end of the simulation. Once the final cycle time is confirmed to meet customer requests, this spreadsheet can be used as input for elastodynamic simulations to verify mechanical aspects. Alternatively, the position derived from elastodynamic simulations can be made available to VC software as input via an Application

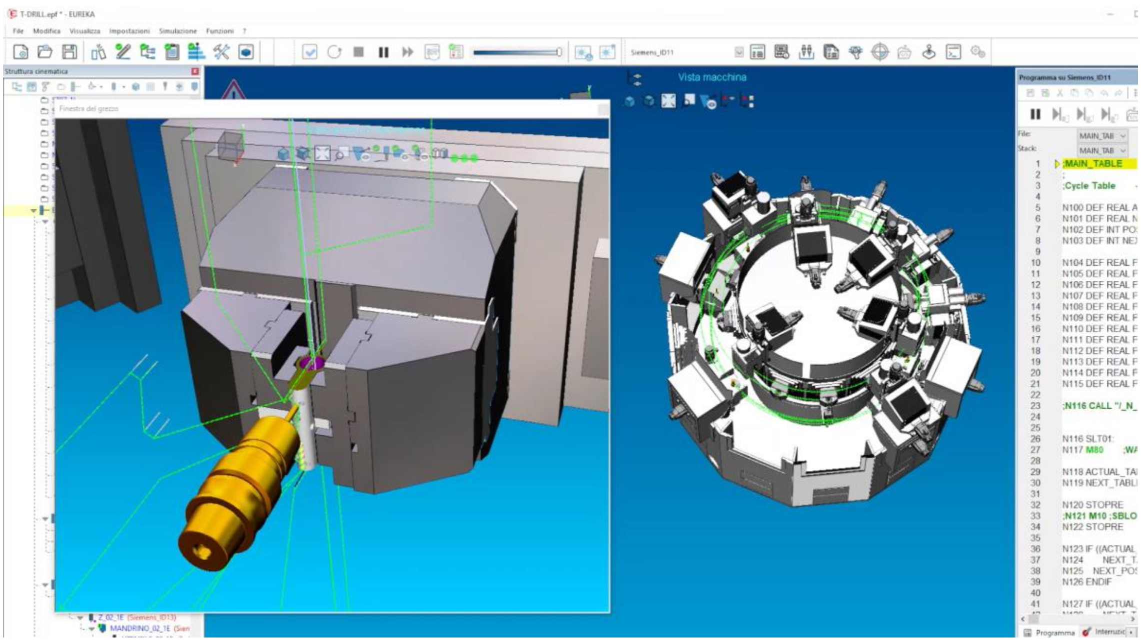

Figure 3. *Model of the transfer machine tool on the right and magnification of a drilling operation on the left.*

Programming Interface (API). This approach enables the use of actual positions that consider the milling machine's compliance as input for the VC simulation, enabling the verification of the machined workpiece's adherence to requested tolerances. The approach is intended to offer an effective and cost-efficient means of evaluating the machine's performance under different operating conditions. As a result, it will be a valuable tool in supporting decision-making processes during the early stages of the design process.

3.2 Elastodynamic modelling of the milling machine

As previously mentioned, each operative unit comprises three CNC-controlled axes. When viewed from the front, the Y-axis controls the forward and backward translation, and moves along roller recirculation guides relative to the basement. The X-axis moves horizontally from right to left with respect to the Y-axis, while the Z-axis moves vertically and is constrained to the X-axis. The transmission chain for both the Y and Z axes schematically consists of: servo-actuator, elastic coupling, ball screw, and rolling bearing slide. The slide is fixed to the translational component of the ball screw, which has a pitch of 5mm and is supported through recirculating roller guides. As mentioned, the X-axis has the same kinematic structure, except for the connection between the driving shaft and screw via two pulleys and a toothed belt instead of the elastic coupling (Fig.4a). Given the similarity between the various axes, this paper focuses on the realization process of the ADAMS model for a single axis, the X-axis, and highlights the differences between this axis and the others when necessary.

Following the order of the transmission chain described above, the model comprises the following constraints:

- a fixed joint for the motor,
- a revolute joint between the motor shaft and the motor,
- a cylindrical joint between the shaft and the driving pulley (or the elastic joint for the Y and Z axes),
- a general force between the shaft and the driving pulley (or the elastic joint for the Y and Z axes),
- a revolute joint between the bearings and a component integral to the motor (not present for the Y and Z axes),
- a coupler between the revolute joint of the shaft and the revolute joint of bearing which support the screw (not present for the Y and Z axes),
- a fixed joint between the driven pulley, the screw, and the bearings (only the last joint is present for the Y and Z axes),
- a screw joint between the screw and the ball recirculating component,
- a fixed joint between the ball recirculating component and the slide,
- one bushing between each of the four linear roller blocks and the slide (for the Y-axis, they are between the four linear roller blocks and the basement),
- an in-plane joint for each linear roller block (to stop the translation with respect to the direction of the relative velocity of the guides),
- four translational joints between the linear roller blocks and the two rails (and activated the relative friction tool),
- two fixed joints between the two guides and a component integral to the motor (for the Y-axis, they are between the two guides and the slide).

In particular, the general force with bushing-like characteristics was used to take into account the axial (adding the force component on the translational degree of freedom that remains unconstrained within the cylindrical joint) and torsional stiffness (adding the torque component of the translational degree of freedom that remains unconstrained within the cylindrical joint) of each component. The axial and torsional stiffness values of the components were obtained from datasheets or by calculations (for example, the motor shaft has been assimilated to a round section beam) and were added as stiffness of springs connected in series, and as regard the screw, were left parametric with respect to the distance between the bearings and the sliding ball recirculating component. For bushings, stiffness and damping values found in catalogues were added in the directions shown with springs symbols in Fig. 4b.

To improve the model's replication of real-world phenomena, a second, more refined, model was implemented by adding contacts in the linear guides. In particular, contacts between rails and blocks were introduced by deactivating the four translational joints and adding rollers, with dimensions equivalent to those inside the recirculating blocks. The rollers were designed as cylinders within the ADAMS environment to reduce computational load and positioned at the ends of the tracks where they would roll. This was done individually for each block, resulting in eight rollers per block, as shown in Fig. 5. Finally, the rollers were fixed to their respective blocks.

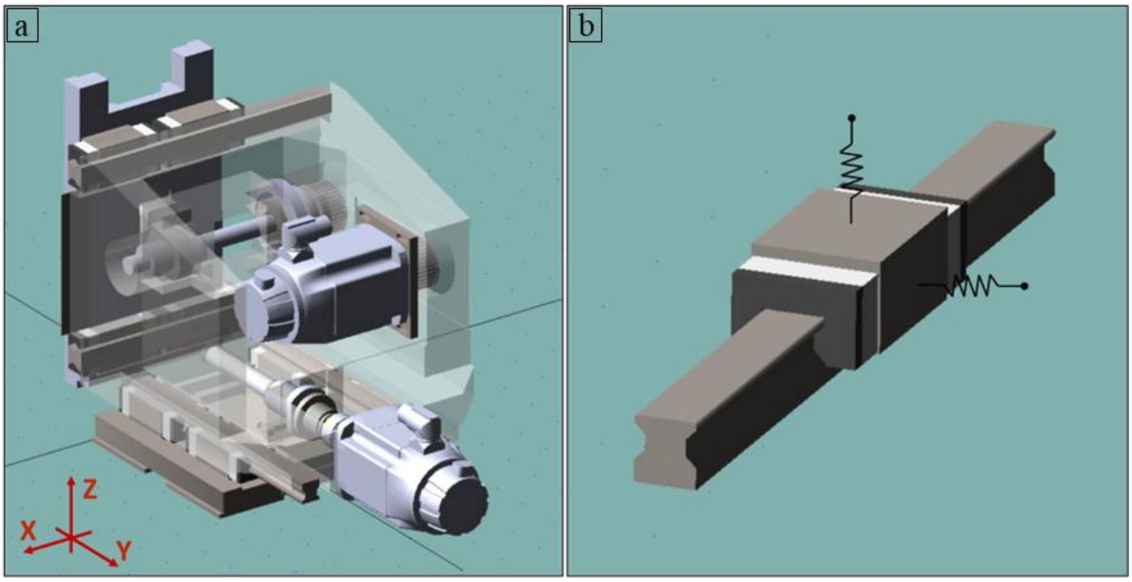

Figure 4. *(a) Mechanical components of the X and Y axes; (b) schematic of the bushings in linear roller blocks.*

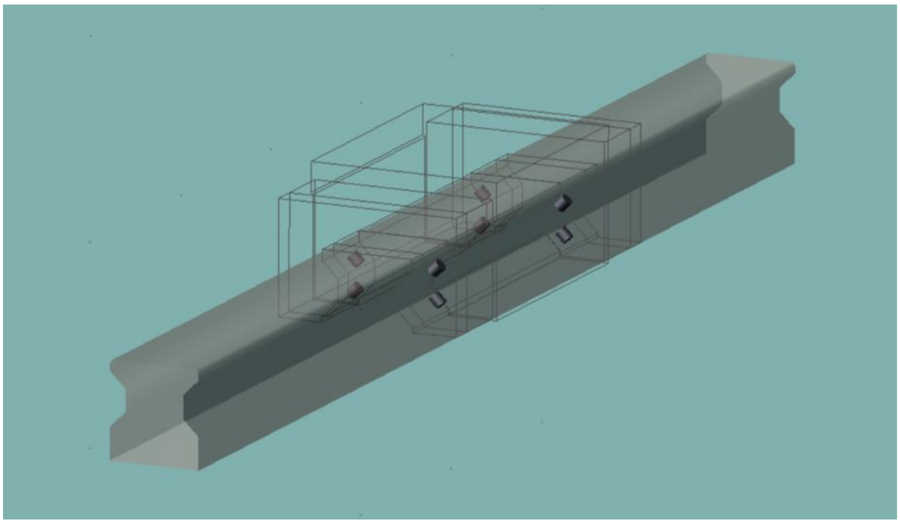

Figure 5. *Rail and wireframe of the recirculating roller block with its eight rollers.*

4 RESULTS AND DISCUSSION

The following section presents the results of the first round of validation utilizing the measured motor torques. The obtained results are compared with those of the ADAMS base model, referred hereafter as "model without contacts," and subsequently with those of the ADAMS model with the introduction of contacts, referred to hereafter as "model with contacts". As described in Section 3.2, as similar trends were observed for the other axes, only results for the X-axis are presented.

In each figure, torque curves required to ensure the movements described in Subsection 3.1 of the X-axis were plotted, along with the curve of the measured feed rate. It should be noted that the feed rate set in the part program is the feed rate in the 3D space, therefore, if only one axis is driven, the feed rate will remain the same, if two axes are driven, the axial feed rate will be divided by the square root of two (planar movement) and if all three axes are driven, the axial feed rate will be divided by the square root of three.

In Fig. 6, the torque curves for the X-axis were generated with a feed rate of 15000 mm/min and a jerk of 1000 m/s^3. It is evident that the magenta curve obtained from the model without contacts deviates from the black curve obtained from experimental data. The torque values obtained from simulations in constant feed rate zones were found to be lower than those obtained from experimental data by 64% in Zone A, 56% in Zone B, 61% in Zone C, and 54% in Zone D, on average. Some discrepancies are also observed for peak values.

The figure shown in Fig. 7 demonstrates similar behavior to the previously plotted curve, but with a feed rate of 7500 mm/min and a jerk of 500 m/s^3. Also in this case, even if with a minor percentage error, the torque values obtained from simulations in constant feed rate zones were found to be lower by 46% in Zone A, 26% in Zones B, 31% in Zone C, and 24% in Zone D than those obtained from experimental data.

In contrast, the model with contacts produced significantly improved results both in constant speed zones and peaks. The torque values for movements with a feed rate of 15000 mm/min and a jerk of 1000 m/s^3 obtained from simulations in constant feed rate zones were compared to experimental data and were found to be lower by 6% in Zone A, higher by 8% in Zones B, by 4% in Zone C, and 15% in Zone D, as shown in Fig. 8.

Figure 9 shows the torque values for movements with a feed rate of 7500 mm/min and a jerk of 500 m/s^3. The values obtained from simulations in constant feed rate zones were found to be higher by 13% in Zone A, by 26% in Zones B, by 17% in Zone C, and 6% in Zone D compared to experimental data. Although the torque values obtained through simulations seem to overestimate the torque values of the experimental data, it should be noted that where the percentage values are higher, the curve obtained from simulations is overlapped with the maximum values of the curve of experimental tests, which presents more ripples. Indeed, the percentage error previously reported was calculated based on the mean values of the experimental curve and the simulation curve. The latter exhibits a more stable behavior and is essentially superimposed on the maximum values of the former curve, which exhibits significant oscillations.

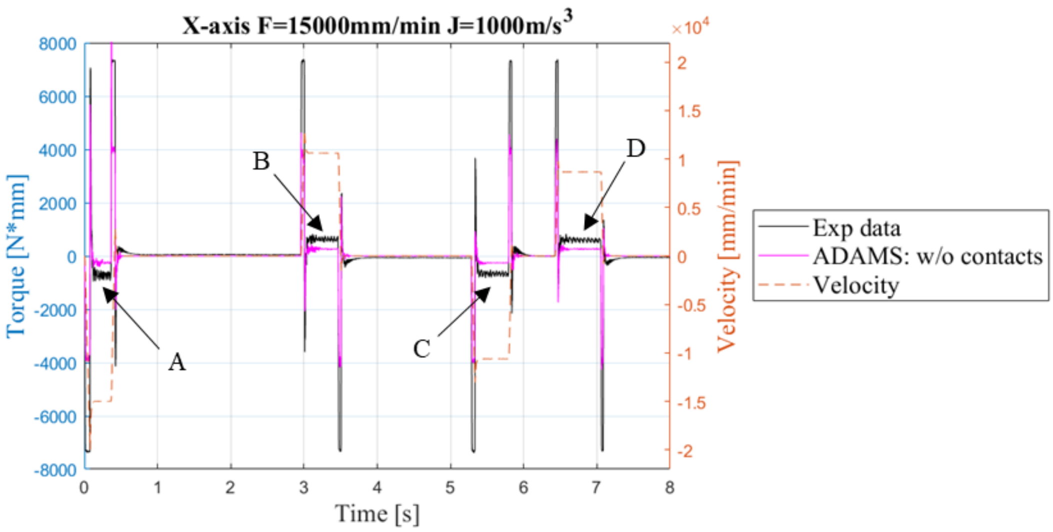

Figure 6. *Comparison between the torque curves obtained from the model without contacts and the experimental data, with a feed rate of 15000mm/s and a jerk of 1000 m/s^3 of the X-axis.*

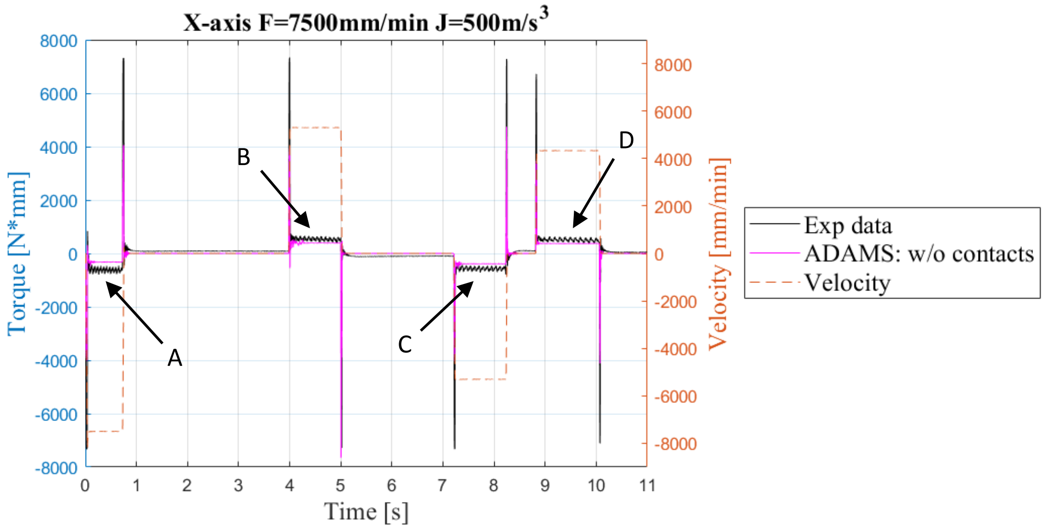

Figure 7. *Comparison between the torque curves obtained from the model without contacts and the experimental data, with a feed rate of 7500mm/s and a jerk of 500 m/s^3 of the X-axis.*

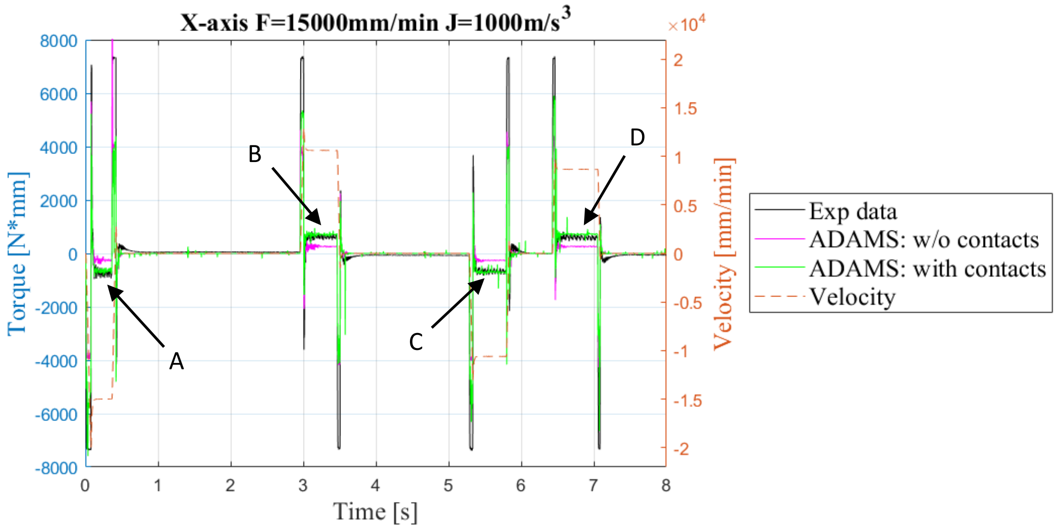

Figure 8. *Comparison between the torque curves obtained from the model without contacts, the model with contacts and the experimental data, with a feed rate of 15000mm/s and a jerk of 1000 m/s^3 of the X-axis.*

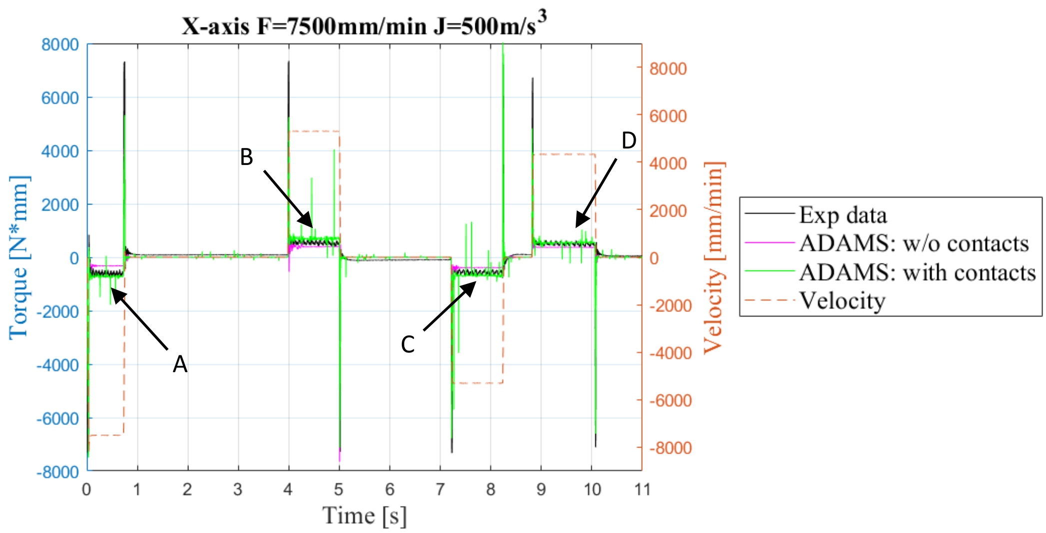

Figure 9. *Comparison between the torque curves obtained from the model without contacts, the model with contacts and the experimental data, with a feed rate of 7500mm/s and a jerk of 500 m/s^3 of the X-axis.*

Both models necessitate further enhancements, particularly the model lacking contacts, as it is important for the lower computational load, to achieve a better fit in the transient phases, characteristic not yet considered in the paper. To address this issue, a new formulation of the stiction phenomenon is currently under development and a second round of model updating will be performed by exploiting additional experimental data, e.g. accelerometric signals.

5 CONCLUSION

This paper presents the first steps of a research project intended to propose an integrated approach for optimizing the design of mechatronic systems, with a particular focus on machine tools. Advanced modeling and simulation techniques are used to improve companies' competitiveness. A sequential methodology is conceived to highlight and address specific industrial requirements for implementing such techniques, which can simultaneously consider both mechanical and automation aspects, due to the unavailability of a comprehensive software solution in the market.

To this aim, an elastodynamic model of a milling machine (which is the focus of this paper) has been developed and then improved with the introduction of contacts between the sliding components involved in the motion transmission. The updated model provides results that are in agreement with experimental data. This model can be used to enhance the dynamic response of the milling machine and the quality of the overall production process when employed in combination with VC simulations.

Ongoing and future research studies are aimed at (i) further improving the accuracy of the flexible multibody model and (ii) implementing the integration with VC simulations in order to achieve a comprehensive virtualization of the entire CNC machine tool.

REFERENCES

[1] Oppeltand, M. and Urbas L., “Integrated virtual commissioning an essential activity in the automation engineering process: From virtual commissioning to simulation supported engineering”, in Proceedings of IECON 2014, Dallas (TX, USA), 2014, pp. 2564-2570.

[2] Reinhart, G. and Wünsch, G., “Economic application of virtual commissioning to mechatronic production systems”, Production Engineering, Vol. 1(4), pp. 371-379 (2007).

[3] Tao, F., Zhang, H., Liu, A. and Nee, A.Y., “Digital twin in industry: State-of-the-art”, IEEE Transactions on Industrial Informatics, Vol. 15(4), pp. 2405-2415 (2018).

[4] Drath, R., Weber, P. and Mauser, N., “An evolutionary approach for the industrial introduction of virtual commissioning” in Proceedings of IEEE-EFTA, Hamburg (Germany), 2008, pp. 5-8.