# Improving Machining Performance of In-use Machine Tools with Active Damping Devices

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*Keywords: Machine tool, Dynamics, Active Damping*

## Abstract

Machine tool productivity is governed by machining stability of the system, which varies as a function of the position of the machine in the workspace. To enhance machine tool performance over the entire workspace, this paper presents an approach that integrates an active damping device to increase the dynamic stiffness of structural modes that limit productivity. The actuator is sized and selected based on defined performance objectives of an increase in productivity. Device performance is simulated and validated experimentally, and it is demonstrated that improvements of up to 22% in productivity and part surface quality is possible with such devices.

## 1 Introduction

Machine tool performance characterized by its ability to produce a part of required quality in the minimum time possible is significantly influenced by its dynamic stiffness at the tool center point. For in-use machine tools, lack of this dynamic stiffness may lead to unstable chatter vibrations which limit the achievable stable material removal rates, and may also result in poor quality of part surface [1]. An increase in the dynamic stiffness, which is a function of the modal stiffness and damping of the machine, will evidently result in an increase in machine tool performance. Traditional methods to modify machine tool dynamic stiffness include

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structural level modification (stiffening through optimization); selection of structural materials with higher damping constants; or involve the use of passive vibration absorbers, tuned or otherwise [2]. However, these conventional methods are more favorable at the design stage and are less practical to implement on in-use machine tools, if (when) their performance is found wanting. Moreover, passive systems can damp only specific modes and are not feasible in applications where the dynamics of the system change as a function of position, and an active system is often needed.

Active means of damping based on control of an actuator integrated with a structure has shown to be rather effective; see [3] for an excellent summary of various active damping strategies. Several recent successful examples of integrating active systems within machine tools can also be found in [4-8]. Though effective, in most cases integrated active systems have seldom been more than merely concept proving, with little consideration paid to design/selection of active systems based on force requirements to meet defined performance objectives. Moreover, the position-varying dynamics against which the active systems are meant to be tested have also been less investigated in the available literature; [8] being one example. Furthermore, the use of active systems in finish milling applications wherein by reducing the tool vibrations levels they can improve the part surface quality has also not received much attention in the available literature.

The above issues are addressed in the present article by describing methods that consist of five main steps that will effectively improve the dynamic performance of in-use machine tools by means of integrating an active damping device. At first, position-varying dynamics of a 5-axis machine tool are evaluated from its finite element (FE) model - discussed in Section 2; following which position-varying machining stability is evaluated for representative rough milling operations. For a desired improvement in performance levels, an active damping device is integrated and tested in the virtual environment in Section 3. Section 3 also presents a systematic methodology to size and select inertial actuators to meet the performance objectives when subject to external disturbances. This is followed by experimental validation of the device in Section 4. Simulation driven investigations in Section 5 demonstrate how the target productivity and improvement to part surface quality may be achieved; which is followed by the main conclusions in Section 6.

# 2 *Machine tool dynamics and stability*

As an example of a machine whose performance is to be evaluated and modified as necessary, a representative 5-axis machine tool is considered, as shown in Figure 1, which shows only its virtual, finite element model equivalent.

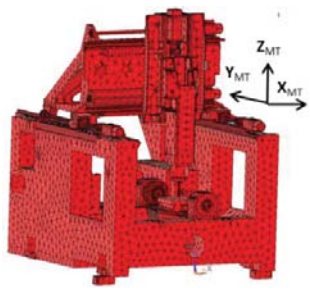


***Figure 1:*** *Virtual (FE model) of the 5-axis machine tool under consideration*

## 2.1 Position-dependent dynamic behavior

For the machine in Figure 1, it is evident that dynamic response will be more influenced by the movement of the tool along the machine Z axis than the X/Y axis. Hence, the position-dependent behavior shown in Figure 2 is evaluated at four different Z levels of the tool: when the tool is near the table (bottom position); and when it has moved in the +Z direction by 200 mm; by 300 mm; and, by 400 mm respectively. For the given kinematic configuration, the Z-directional response is an order of magnitude stiffer than the X and Y directions, and is not further considered.

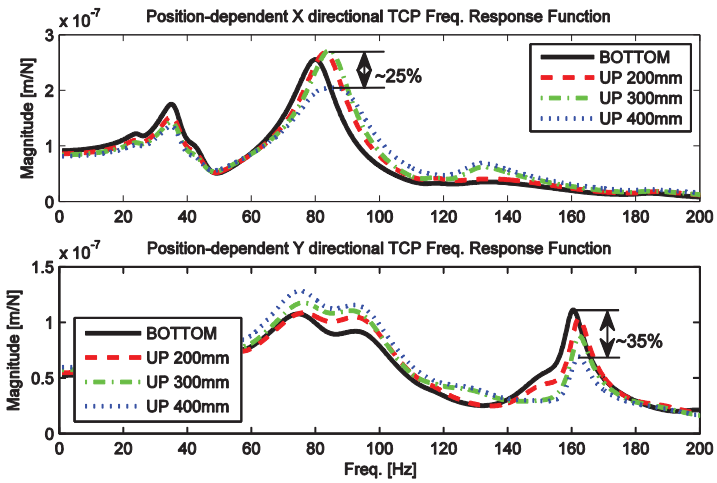


***Figure 2:*** *Simulated position-dependent TCP response in X and Y directions*

Simulated tool center point (TCP) frequency response functions (FRFs) are all updated with modal damping estimates from measurements carried out at the bottom position. It is assumed that modal damping estimates remain position-independent. Only the low-frequency global/local structural modes are affected by change in tool position, hence position-dependent comparisons in this study are limited to 200 Hz. The mid-high frequency behaviour corresponds to the spindle-tool-tool holder combinations, and does not exhibit strong position-varying behaviour.

As evident from comparisons, response varies by as much as ~35% over the Z stroke and is non-monotonic, being stiffer sometimes at the end of stroke than in the middle. The lower frequency modes between ~30-70 Hz correspond to the global structural bending modes in the X and Y directions respectively, whereas the modes between ~70-200 Hz correspond to the local bending modes of the Z-slide. To investigate the influence of these varying dynamics on machining stability a representative machining operation of heavy duty milling of steel is chosen; which is carried out at lower cutting speeds and which tends to generate excitation of the global structural modes below 200 Hz [5].

## 2.2 Position and feed direction dependent machining stability

The position-dependent stability is determined using a modal model of the machine and the following characteristic equation [1]:

$$det([\mathbf{I}] + \Lambda[\boldsymbol{\Phi}_{PD}(i\omega_c)]) = 0 \tag{1}$$

where $\Lambda = \Lambda_R + i\Lambda_I = -\frac{1}{4\pi}\mathrm{N_t K_t} a\left(1 - e^{-i\omega_c T}\right)$ (2)

is the complex eigenvalue of the characteristic equation; $\Lambda_R$ and $\Lambda_I$ are its real and imaginary parts; $\mathrm{N_t}$ is the number of teeth on the cutter; $\mathrm{K_t}$ is the cutting force coefficient of the material being cut; $a$ is the axial depth of cut; $\omega_c$ is the chatter frequency; and, $T$ is the tooth passing period. $\boldsymbol{\Phi}_{PD}$ within Eq. (1), which is a function of the directional factors, $\boldsymbol{\alpha}_0$, and the TCP transfer function matrix in the machine tool principal directions, $\boldsymbol{\Phi}_{xy}$, is known as the oriented position-dependent transfer function matrix $\left([\boldsymbol{\Phi}_{PD}] = [\boldsymbol{\alpha}_0][\boldsymbol{\Phi}_{xy}]\right)$. The limiting depth of cut, described by the parameters in Eq. (1) and Eq. (2), may be analytically determined as [1]:

$$a_{lim} = -\frac{2\pi\Lambda_R}{N_t K_t}\left[1 + \left(\frac{\Lambda_I}{\Lambda_R}\right)^2\right]. \tag{3}$$

Position-dependent stability is evaluated, as shown in Figure 3 for 75% immersion down-milling of DIN 34CrNiMo6 (EN 1.6582) steel with a 20 mm diameter tool with $N_t = 4$; $K_t = 3000$ MPa; and the radial coefficient, $K_r = 0.24$.

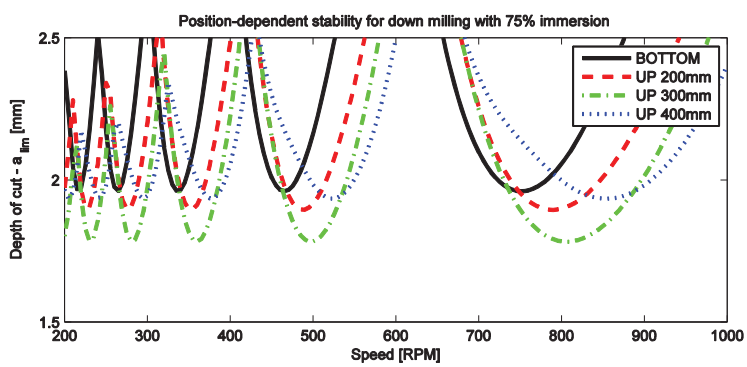


***Figure 3:*** *Comparison of simulated position-dependent stability lobes. Feed direction in +X direction*

The region above the lobes in Figure 3 is unstable and the one below is stable. Absolute minimum stability limits, as well as the stability pockets are

observed to be strongly position-dependent. Moreover, in the low speed region, it will be difficult to select cutting parameters which are stable at all positions. Moreover, besides being influenced by the position-dependent directional compliances, stability also changes as a function of feed direction due to the effect of modal projections on these feed directions [9], and before implementing an active system to improve machining performance; it becomes necessary to evaluate the changing stability (machining performance) over the entire work volume.

This is accounted for by projecting the vibrations $(\boldsymbol{\Phi}_{xy})$ of the tool in machine tool principal directions $(xy)$ into the feed $(uv)$ directions when the tool is travelling at an angular orientation $\phi$ with respect to the $x$ axis. The feed-plane transfer function matrix at the TCP, $\boldsymbol{\Phi}_{uv}$, hence becomes [9]:

$$[\boldsymbol{\Phi}_{uv}] = [\boldsymbol{R}][\boldsymbol{\Phi}_{xy}][\boldsymbol{R}]^{-1} \tag{4}$$

where $\boldsymbol{R} = \begin{bmatrix} cos\ \phi & -sin\ \phi \\ sin\ \phi & cos\ \phi \end{bmatrix}$ is a rotational operator.

Solution to Eq. (1-3) updated to account for different feed directions (0-360˚) by use of Eq. (4) results in position-dependent speed-independent absolute minimum stable depths of cut which vary across feed directions in proportion to the magnitude of projections of the modes in that direction – as shown in Figure 4.

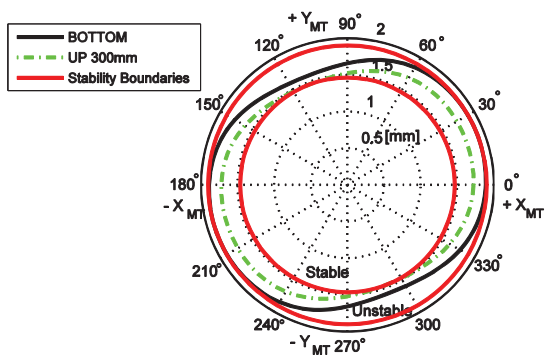


***Figure 4:*** *Simulated feed direction dependent speed-independent stability for two positions*

Results in Figure 4 only apply for the dynamically most and least stiff positions and with the same machining parameters as before. The regions inside the stability envelopes are stable. The absolute limiting depth of cut is plotted radially, while the machining (feed) directions are plotted circumferentially. These

feed direction dependent stability charts are a deviation from the conventional stability lobes in a way that they represent only the absolute minimum stability at a particular position and feed direction. The curves are symmetric about 45º and 135º respectively, i.e. in directions where the mode under consideration halves the angle between the principal directions. The shape and envelope of the stability boundaries are a strong function of the engagement conditions in up/down milling, and the position-varying strengths of directional modes. The absolute depth of cut of 1.46 mm occurs at the 110° feed orientation for when the tool is at a distance of +300 mm from the bottom position, whereas the absolute maximum depth of cut of 1.9 mm happens at the 200° orientation for when the tool is at the bottom position. Figure 4 also shows the absolute minimum/maximum circular stable boundaries.

### 2.3 Defining target performance levels

Position-varying stability of the system will result in changing productive cutting conditions over the work volume and may require planning dynamically changing machining trajectories; or, alternatively, it may result in selection of cutting parameters below the lowest of all possible stability thresholds, thereby resulting in a slower material removal process. To avoid these issues it is desirable to have a near uniform stable depth of cut within the whole working range of the machine; and it is targeted that at the least a speed-independent position and feed direction independent 2 mm stable depth of cut be attained by integrating active damping devices (ADDs). The target 2 mm depth of cut corresponds to a ~30% increase from the absolute minimum 1.46 mm depth of cut. All other parameters remaining constant, since the limiting stability is a function of the dynamic stiffness of the most flexible mode, increasing this with the ADD by ~30% should result in an equivalent increase in the stability of the system.

## *3 Integration of an active damping device*

To meet the performance objectives for the given mechanical system when subject to external disturbances, structural control with an integrated actuator is a function of the type of the actuator, its controller, and the actuator/sensor mounting locations [3]. The performance objectives in the present case are a desired ~30% increase in dynamic stiffness of the structural modes, i.e. the operating bandwidth

of the device should be > 200 Hz. In addition, the device should act against disturbances representative of the cutting force levels during heavy-engagement milling of steel. In the present case, a proof-mass actuator is selected as the actuator type since it has been previously shown to be effective in a broad range of structural vibration control problems, including in machine tools [3-8]. The working principle of the actuator, its control strategy, a generalized mathematical modelling strategy and sizing considerations for structural control are discussed below.

## 3.1 Proof-mass actuator and its control

A proof-mass actuator (Figure 5) is based on accelerating a suspended mass that results in a reaction force on the supporting structure [3]. A reaction mass $m_p$ is connected to the support structure by a spring $k_p$, a damper $c_p$ and a force actuator $f_a$ – which in the present case is electromagnetic. In the electromagnetic actuator discussed here, the force actuator consists of a voice coil transducer of constant $T$ (in N/A) excited by a current generator $i$. Combining the oscillator dynamics with the Lorentz force law, the transfer function in the Laplace domain between the total force applied to the support and the current applied to the coil is [3]:

$$\frac{F}{i} = \frac{-s^2 T}{s^2 + 2\zeta_p \omega_p s + \omega_p{}^2} \tag{5}$$

wherein $\omega_p$ is the natural frequency of the spring mass system in Figure 5.

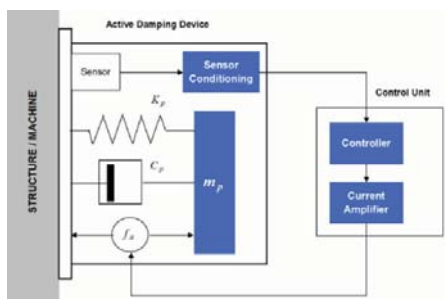


***Figure 5:*** *Schematic of the proof-mass active damping device with a controller [10]*

This inertial actuator is controlled by a collocated vibration sensor and a controller implementing a direct velocity feedback (DVF), which has been reported to have better performance as compared to direct acceleration, or direct position

feedback controllers [8]. The combined effect of the DVF controller and the inertial actuator is to add viscous damping into the system. The only selectable parameters for the control system are the actuator/sensor location and the feedback gain. In the present case, the actuator is mounted on side surface of the Z-slide, close to the tool. Since the ADD exerts a force in proportion to the gain setting based on the state feedback, the achievable damping is a function of the feedback gain, and of the ADD system parameters ($m_p$, $k_p$, $c_p$, and $T$). To establish operational guidelines and prescribe selection and sizing considerations for the ADD, a generalized mathematical model is developed in the next Section.

## 3.2 Generalized machine mechatronic model

Since the ADD can be represented using a transfer function description, and since it comes with its own controller, it is most convenient to decompose the modal model of the machine to the state-space (SS) domain for analyses as:

$$\begin{aligned} \dot{\boldsymbol{x}} &= \boldsymbol{A}\boldsymbol{x} + \boldsymbol{B}\boldsymbol{u} \\ \boldsymbol{y} &= \boldsymbol{C}\boldsymbol{x} + \boldsymbol{D}\boldsymbol{u} \end{aligned} \tag{6}$$

where $\boldsymbol{x}$, $\boldsymbol{u}$, and $\boldsymbol{y}$ are the state, input and the output vectors of the system. The state vector consists of the modal vector of generalized displacements and its derivatives, i.e. its velocities. The $\boldsymbol{A}$, $\boldsymbol{B}$, and $\boldsymbol{C}$ matrices can be described as:

$$\boldsymbol{A} = \begin{bmatrix} \boldsymbol{0} & \boldsymbol{I} \\ -\boldsymbol{\Lambda} & -\boldsymbol{C_q} \end{bmatrix}_{2n\times 2n}; \boldsymbol{B} = \begin{bmatrix} \boldsymbol{0} \\ \boldsymbol{V}^T \boldsymbol{F}_{physical} \end{bmatrix}; \ \boldsymbol{C} = [\,\boldsymbol{V} \quad \boldsymbol{0}] \tag{7}$$

wherein

$$\boldsymbol{\Lambda} = \begin{bmatrix} \omega_{n_1}^2 & 0 & 0 \\ 0 & \ddots & 0 \\ 0 & 0 & \omega_{n_n}^2 \end{bmatrix}_{n\times n}; \boldsymbol{C_q} = \begin{bmatrix} 2\zeta\omega_{n_1} & 0 & 0 \\ 0 & \ddots & 0 \\ 0 & 0 & 2\zeta\omega_{n_n} \end{bmatrix}_{n\times n}; \ \boldsymbol{V} = [\boldsymbol{V_1} \quad \dots \quad \boldsymbol{V_m}]_{n\times m}; \boldsymbol{F}_{physical} = [\boldsymbol{I}] \tag{8}$$

where $n$ – is the number of degrees of freedom; $m$ – the number of modes; $\omega_n$ and $\boldsymbol{V}$ – represent the eigenvalues and the mass normalized eigenvectors output from the FE environment – for the machine model. Dynamically, the model expressed in Eq. (6-8) along with the actuator and controller (an integrator with a high pass filter) is modeled in MATLAB® Simulink, as shown in Figure 6.

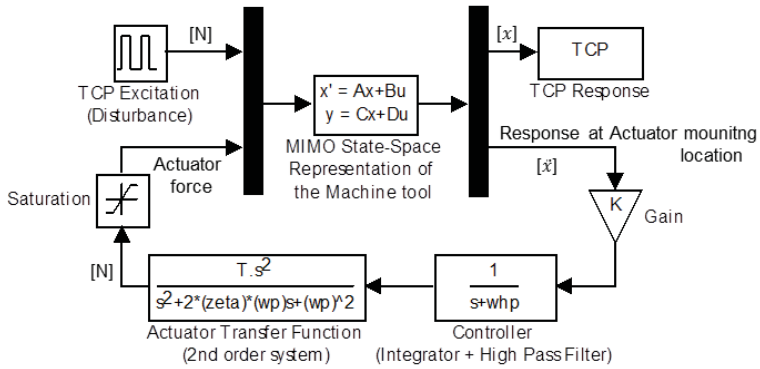


***Figure 6:*** *Simulink model of the machine tool integrated with the active damping device*

## 3.3 ADD sizing, selection and force requirements

The limiting (defining) criterion for the actuator is the maximum force that it can output. At first, it is important to decide what the force output of the device should be to achieve a target ~30% increase in dynamic stiffness at the TCP subject to a 500 N level of disturbance force, $f_d$. This disturbance is representative of the cutting force levels for a depth of cut of 1.5 mm, feed of 0.15 mm/tooth, for down milling DIN 34CrNiMo6 steel with 75% engagement [11]. To make the force output independent of ADD system parameters, all parameters are initially set to unity, i.e. $m_p = k_p = T = 1$, and $c_p = 0$; the saturation block and the high pass filter is turned off and the gain is adjusted iteratively to find the force required to meet the target increase in dynamic stiffness. It is found that for a non-dimensional gain of $K = 5 \times 10^4$, the dynamic stiffness for the dominant modes increases by ~30% for an ADD force output of ~54 N. Similarly, for a 100% increase in dynamic stiffness, the required ADD force output was found to be ~160 N for a gain of $K = 1.65 \times 10^5$.

The above procedure presents a systematic methodology for the design/selection of inertial actuators based on defined performance objectives and disturbance specifications. Since the focus of this exercise is not to develop a new ADD, but to prescribe guidelines to select from available commercial ADDs based on performance requirements, the ADD45 device from Micromega Dynamics [10] has been selected. This device can output a maximum of 45 N, and has system parameters of: $m_p = 2.2\ kg$; $k_p = 6.13\ {}^{kN}/_{m}$; $\zeta_p = 0.15$; $and\ T = 20\ N/A$ [10]. The 45 N force output leads to a ~25% increase in dynamic stiffness (for high gain

settings), as shown in Figure 7. Though this is slightly less than the target ~30% increase, the ADD was found to saturate for higher gain settings.

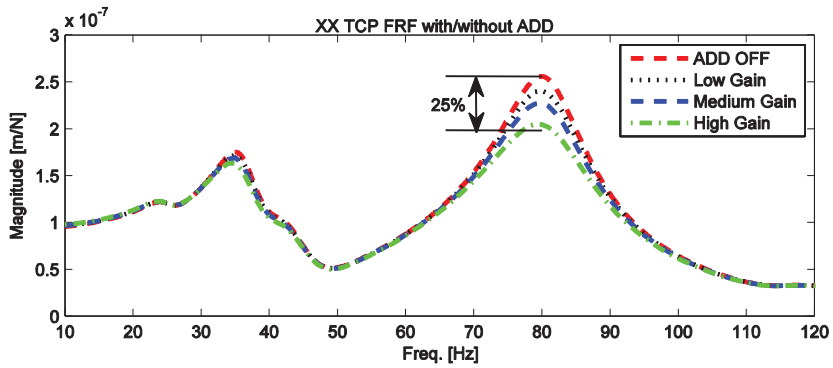


***Figure 7:*** *Simulated X-directional TCP response with/without the ADD for different gain levels*

If the disturbance input is of a greater magnitude (> 500 N), the ADD will be forced to operate with lower gain settings, possibly reducing the achievable level of damping. For the selected device, the magnitude of the steady-state actuator force required per magnitude of steady-state disturbance force is best described by Figure 8 – which shows the ratio of the actuator force to the disturbance force for different excitation frequencies at different gains. This information is useful for sizing the actuator if the magnitude of disturbance is known [12]. For structural modes having low natural frequencies, this force requirement is often feasible.

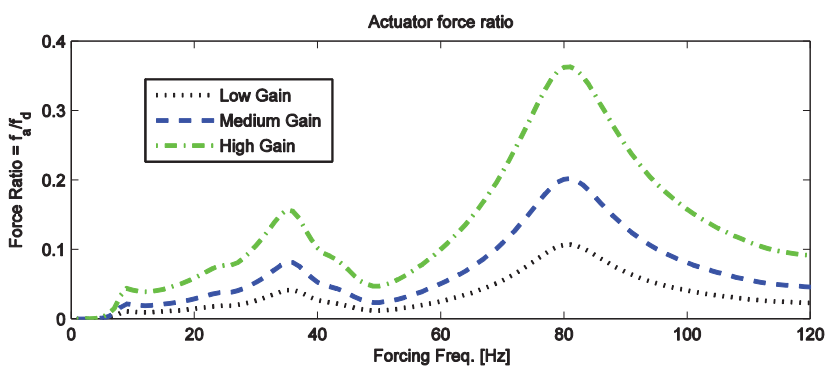


***Figure 8:*** *Magnitude of the steady-state actuator force required per magnitude of steady-state disturbance force for different excitation frequencies for the ADD45 device, in X direction only*

## 4 Experimental validation with ADD

The measurement setup is as shown in Figure 9. Two ADDs are mounted (one each in the X and Y direction, though Figure 9 shows ADD mounted only in X direction) with magnets on the side surface of the Z-slide. The device is tested by providing disturbance with an instrumented impact hammer in place of cutting forces against which the ADD must act; and experimental modal analysis is conducted with the ADD being Off/On. Modal measurements were made with CUTRPO® [11], and the ADD was controlled using its own control unit [10]. Measured and simulated response with/without the ADD is compared in Figure 10.

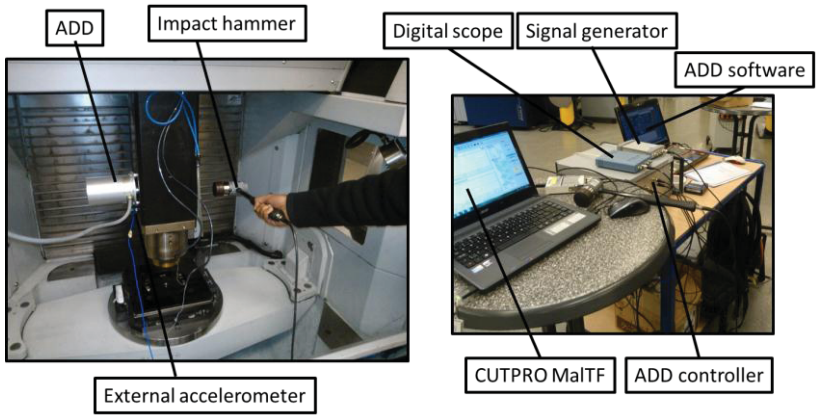


***Figure 9:*** *Measurement setup to test the machine tool dynamics with the ADD*

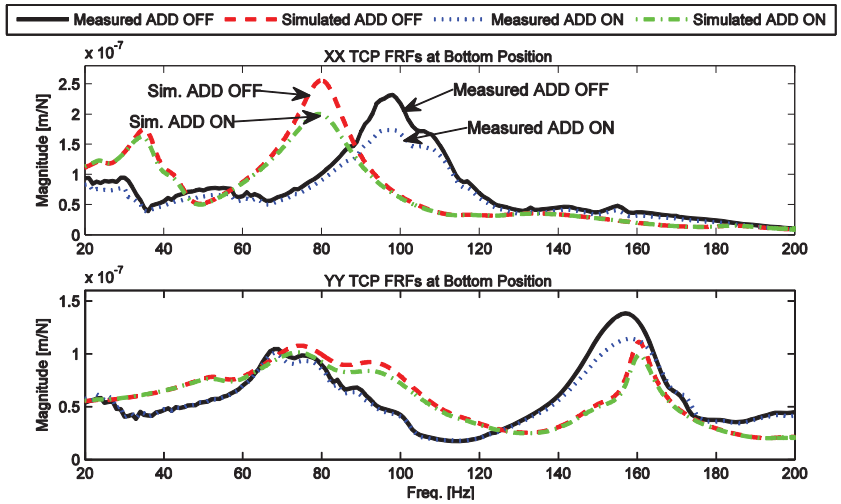


***Figure 10:*** *Comparison of measured and simulated (FE) TCP response when ADD is off/on*

Simulated behavior is able to achieve levels of damping similar to those attained with measurements. Discrepancies observed in natural frequencies between the measurements and simulation results may be attributed to: modeling simplifications; and, to a greater extent due to the difficulties with correctly modeling the machine tool joints. As evident from comparisons of response with/without the ADD, the X-directional response is damped more than the Y-directional response, with a 25% increase in dynamic stiffness for dominant mode in the X direction at ~80 Hz, and a maximum of 12% increase in the dynamic stiffness for the mode at ~160 Hz in the Y direction.

## 5 *Improved machining performance*

Further investigations into how the improvement in dynamic stiffness translates to potentially realizing the target position and feed direction independent speed-independent 2 mm stable depth of cut is discussed below using simulations. Also investigated is the potential of using the ADD during finish milling operations to enhance surface quality, as discussed in the next Section(s).

## 5.1 Improved machining stability and productivity

Feed direction dependent speed-independent stability for the dynamically most and least stiff positions is compared in Figure 11 with/without the ADD. Tool, workpiece material and engagement conditions remain the same as before. As evident from comparisons, there is both an increase in the stability envelope as well as an increase in the absolute minimum stability limit by ~22%. The increase in stability envelope due to the increase in damping made possible by the ADD helps achieve the target 2 mm productivity at almost all feed directions at the bottom position, even exceeding the target 2 mm at certain feed directions. At the +300 mm position (which was the dynamically least stiff position), the target 2 mm depth of cut is attained only at certain feed orientations, a considerable improvement over the case without the ADD. Similar increase in stability limits and envelopes at other positions are also observed, exceeding the target 2 mm at most feed directions.

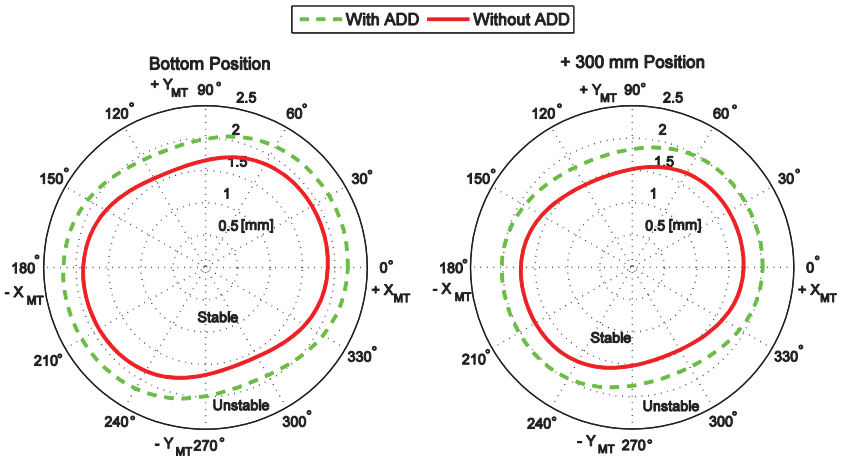


***Figure 11:*** *Simulated feed direction dependent absolute stability with and without the ADD for two different positions, left – bottom position, right – tool has moved +300 mm from bottom position*

## 5.2 Improved surface quality

Even during stable machining operations, the tool vibrations occurring at a forcing frequency near one of the structural modes can lead to a poor surface quality. This is especially consequential in the case of finish milling operations, in

which cutting parameters are generally well inside the stability threshold of the machine, and surface quality is often more paramount. To demonstrate the effect of enhanced dynamic stiffness on the surface finish during stable finish milling, numerical investigations are carried out to investigate three different cases: (i) when the tooth passing frequency is close to the dominant X-directional mode; (ii) when tooth passing frequency is close to the dominant Y-directional mode, and, (ii) when the tooth passing frequency is far from any of the dominant structural modes of vibration. Tool vibrations in principal machine directions, their spectrums, and the resulting surface finish for each of the cases with/without the ADD are shown in Figure 12. Results are obtained using the milling simulation module of CUTPRO® [11]. Cutting parameters selected are: up milling with 25% engagement with a stable depth of cut of 0.5 mm, and a feed of 0.15 mm/tooth. Tool-workpiece parameters remain the same as earlier. It is assumed that only the machine tool system is flexible and the workpiece is rigid.

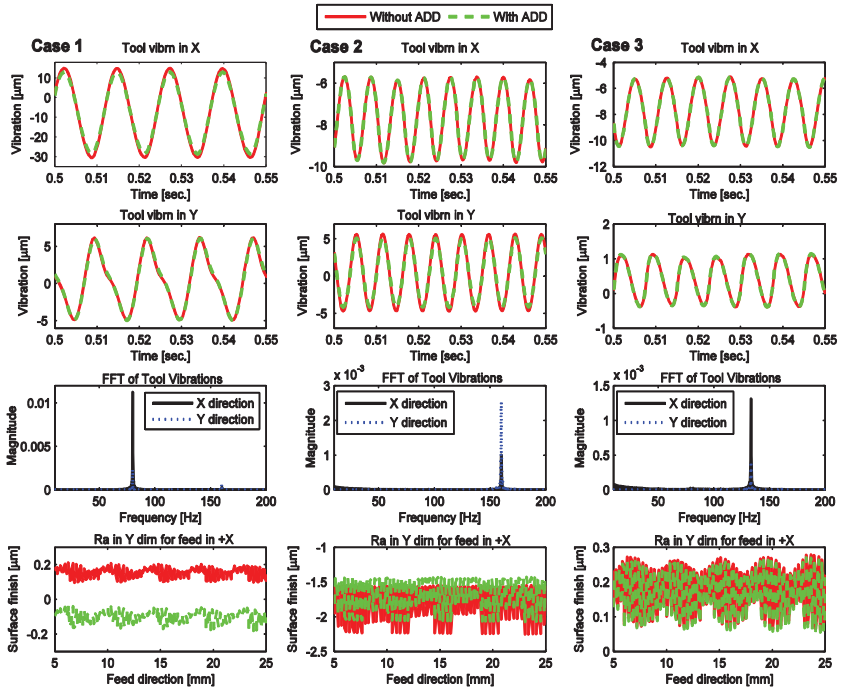


***Figure 12:*** *Simulated tool vibrations, vibration spectrums and surface roughness with/without the ADD for three different cases. Case 1 – left column; Case 2 – middle column; Case 3 – right column*

For Case 1, i.e. for spindle speed at $N$ = 1200 RPM, the tooth passing frequency of 80 Hz ($f_c = N \times \mathrm{N_t}/60$) – is close to the dominant X-directional mode. The use of the ADD in this case leads to a reduction of the X-directional vibrations by ~12%, as well as the RMS surface roughness values by ~28%. Excitation being limited to the fundamental forcing frequency with negligible harmonic components, the Y-directional dominant mode at 160 Hz does not get excited and damping in the Y-directional mode due the ADD has no effect on minimizing the Y-directional tool vibrations. For Case 2, i.e. for spindle speed of 2400 RPM, the tooth passing frequency of 160 Hz – is close to the dominant Y-directional mode. In this case, the fundamental frequency is greater than the structural modes of the structure in the X-direction; hence only Y-directional tool vibrations are damped. In this case, the ADD reduced Y-directional vibrations by ~11%, and improved the surface quality by ~6%. Interestingly, though the X-directional vibration amplitude for the first case is significantly higher than the Y-directional amplitude in the second case, the RMS values for the surface roughness are greater in the second case. This is due to the fact that surface finish results in Figure 12 are shown for the case of surface heights measured in the Y direction for feed in the +X direction; hence if the X-directional mode is excited, it has little effect on the overall surface finish, when measured in the Y direction. For Case 3, since the forcing frequency occurs at 133 Hz (2000 RPM), none of the structural modes are excited; hence the ADD has no effect in minimizing tool vibrations or improving surface finish in this case.

## 6 Conclusions and future work

To demonstrate how machining performance of in-use machines may be enhanced, a systematic approach by integrating an active damping device was presented in this paper, including a methodology for the design/selection of inertial actuators based on defined performance objectives and disturbance specifications.

Performance of the device was simulated and validated experimentally on a 5-axis machine tool. It was further demonstrated that the productivity improved by up to 22% when using such devices. In the case of finish milling, it was shown that use of the ADD may improve the surface quality significantly. Such integration may renew functionality of a machine if it may have changed (degraded) over time, or may also expand its functionality beyond its originally designed function. Further experimentation for testing the device under variable disturbances is planned and forms part of the future work.

## Acknowledgments

We acknowledge support by The Cluster of Excellence “Energy-Efficient Product and Process Innovation in Production Engineering” (eniPROD®) which is funded by the European Regional Development Fund and the Free State of Saxony.

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