Mitigating Chatter in Micro and Mesoscale Milling by Tuning Fixturing Dynamics: A Feasability Study

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KEYWORDS : Fixturing, Microscale Milling, Mesoscale Milling, Regenerative Chatter, Frequency Response

Chatter in micro and mesoscale milling adversely affects the surface quality of machined parts and increases the occurrence of tool breakage. Due to the relatively small sizes of micro and mesoscale tools with respect to that of the machine, this phenomenon is typically driven by the tool dynamics. In this paper, the effectiveness of a fixturing device, designed with the aim of mitigating chatter in micro and mesoscale milling, is assessed for various commercially available tools.

In determining the feasibility of developing such a platform, various criteria were taken into account. The dynamics of the tools were affected not only by the tool dimensions but also their position in the tool holder. Given that the tool is the limiting factor of the stability of the system, two criteria must be met for the fixture to improve the stability: (1) the real dynamic compliance of the fixture must be positive at the same frequency as the tool's minimum real dynamic compliance, and (2) the minimum real dynamic compliance of the fixture must be greater than that of the tool.

The feasibility study discussed in this paper used a combination of numerical and analytical modeling. By comparing the dynamic response of the tool with that of the fixture, a technological window for the applicability of this approach was determined.

NOMENCLATURE

$D =$ tool diameter
$E =$ Young's modulus
$f =$ transverse force
$I =$ second moment of area
$\boldsymbol{K} =$ stiffness matrix
$L =$ tool length
$m =$ mass distribution
$\boldsymbol{M} =$ modal matrix
$P =$ axial tension acting on tool
$\boldsymbol{Q} =$ force matrix
$t =$ time
$x =$ length along cutting tool
$y =$ transverse deflection of cutting tool
$\eta =$ harmonic function

1. Introduction

Today's technological trends call for the capability to manufacture ever smaller parts and feature sizes, with increased precision, at lower costs. This has led to the development of mesoscale machine tools (mMTs). One of the most common mesoscale machine tools today is the miniaturized milling machine. Having benefited from a long history of research and development in the field of macroscale milling, this type of machine is constantly being pushed to new limits. Feature sizes are being decreased into the microscale, while material removal rates and spindle speeds are constantly increasing to new heights. However, with these advancements, many problems have arisen.

Due to the scaling effects of micro and mesoscale endmilling, it can be very difficult to achieve the tolerances required of small parts. One of the most common problems that occur during milling is the self-excited vibration of cutting tools, also known as chatter, which adversely affects surface finish, dimensional accuracy, tool life and machine life [1].

One way to combat regenerative chatter is to optimize the overall stiffness of the machine tool, usually by increasing it as much as possible. However, it is difficult to increase machine tool stiffness without also increasing the size, weight and cost. In miniaturized milling machines, all three of these factors are reduced as much as possible. Furthermore, the weakest link in combating regenerative chatter in micro and mesoscale milling is often the cutting tool, which is far less stiff than the rest of the machine structure and of that the workpiece. Alternatively, stability lobe plots can be used to find the maximum stable depths of cuts for a given spindle speed [2]. The spindle speeds of milling machines can then be tuned in order to increase the maximum depth of cut for a given machining operation.

In some cases, it is possible that regenerative chatter can be mitigated by decreasing the overall stiffness of the machine tool [3,4]. This can be accomplished using a fixturing mechanism that is tuned in such a manner that it attenuates the vibrations of the cutting tool. However, there are several critical design challenges associated with the development of such a platform, not the least of which is the size of the mass of a fixture relative to the modal mass of the tool. In this paper, a tunable milling fixture will be presented and these challenges and many more will be discussed.

In Figure 1, the fixturing mechanism is shown with the top plate removed so that the rest of the mechanism can be more easily seen. The machining workpiece can be fixed to the center bar, which is supported by two fixed-fixed beams. By applying tensile loads along the length of support beams, the stiffness and natural frequency of the mechanism can be tuned. The tension is applied by two screws that, when tightened, pull the left bar toward the long bar. Additionally, springs were added to increase the number of turns needed, thus allowing for more precise stiffness tuning.

In Figure 2, two section views are given that show the effect of turning the screws. In the first figure, the springs are uncompressed so the tensile load is zero, while in the second picture, the screws are fully compressed resulting in maximum tensile load. It should be noted that while the length of the beam does in fact change as a result of the tensile load, the maximum deflection is just 7.15 µm for a beam of 1 mm in thickness.

In the next two sections of this report, the platform design shown in Figures 1 and 2 will be analyzed. However, for the fourth section of this paper, the mass of the center bar will be neglected. With the mass of the center bar shown in the figures, the dynamic compliance was found to be too low to produce significant changes in the dynamic compliance. The decision to omit the center bar mass was therefore made in order to find the best-case scenario for the technological window. However, a redesign of the fixture focused on reducing the modal mass of the center bar will be necessary in order to fully validate the capability of such an approach to mitigate chatter.

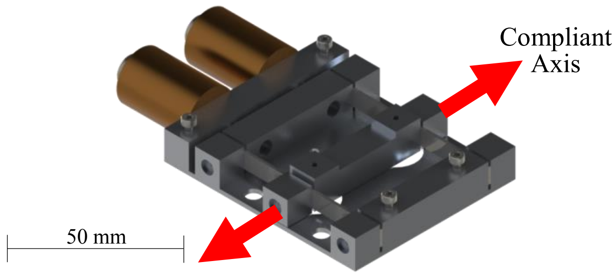

Fig. 1 Rendering of one-DOF compliant fixturing platform with top plate removed for better visibility, showing the compliant axis of the center bar

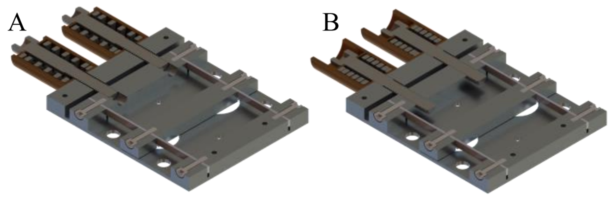

Fig. 2 Section views of compliant platform showing (A) the uncompressed springs applying zero tensile load and (B) the fully compressed springs applying maximum tensile load

2. Tool Dynamics

In micro and mesoscale milling, the dynamics of the cutting operation are often highly dependent upon the dynamic stiffness and natural frequencies of the cutting tool. As such, the most critical part of determining the dynamics of the overall system is to characterize that of the tool. In this paper, a survey of several micro and mesoscale cutting tools will be discussed and trends found in the resulting dynamics of these tools will be determined.

Since micro and mesoscale tools are typically very small in comparison to common modal analysis equipment, the mounting of an accelerometer onto the tool tip for modal testing would be very difficult and would be highly biased by the accelerometer mass. Instead, the modal mass and stiffness of cutting tools is typically determined using finite element methods or beam theory. Usually to determine the overall relative compliance between tool and workpiece, the tool dynamics are coupled to the rest of the machine dynamics, that are measured, by using analytical techniques such as receptance coupling [5]. In this paper, the calculated tool dynamics were verified using an electromagnetic probe. The transverse vibration along the two orthogonal axes in the cross-sectional plan of the beam can be calculated using the Euler-Bernoulli beam equations, as follows [6].

$$E\frac{\partial^2}{\partial x^2}\left[I(x)\frac{\partial^2 y(x,t)}{\partial x^2}\right] - P\frac{\partial^2 y(x,t)}{\partial x^2} + f(x,t) - m(x)\frac{\partial^2 y(x,t)}{\partial t^2} = 0 \quad (1)$$

Diagrams depicting the forces acting on the beam is shown in Figure 3. While tension in the cutting tool, P, will not be specifically addressed in this paper, the downward force acting on the tool increases the natural frequency of the tool and thus changes the stability limit. It was included in the characteristic equation for completeness. The general solution for this equation is given by:

$$Y(x) = A\sin\beta x + B\cos\beta x + C\sinh\beta x + D\cosh\beta,\ 0 < x < L \quad (2)$$

where A, B, C, D, and β are constant coefficients. By applying the boundary conditions for a cantilever beam, the solution can be simplified and the frequency equation can be found.

$$\cos\beta L\cosh\beta L + 1 = 0 \quad (3)$$

By solving for β and plugging into the simplified solution, the mode shapes or eigenfunction of the beam with respect to x can be found. These eigenfunctions are then multiplied by a sinusoidal function with respect to time, t, normalized and then superimposed to form the matrix representation of the characteristic equation.

$$\boldsymbol{M}\ddot{\eta}_r(t) + \boldsymbol{K}\eta_r(t) = \boldsymbol{Q}(t) \quad (4)$$

The modal mass matrix is, by definition, positive semi-definite and the stiffness matrix is assumed to be either positive semi-definite or positive definite. Therefore, the modal parameters can be found by solving the following eigenvalue problem.

$$\boldsymbol{K}\boldsymbol{u_r} = \omega_r^2\boldsymbol{M}\boldsymbol{u_r} \quad (5)$$

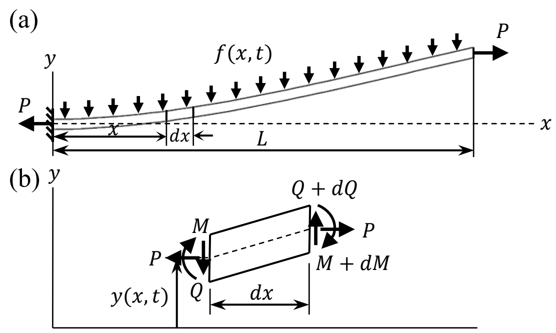

Fig. 3 Diagrams of a fixed-fixed beam under applied tension load, showing (a) the general case of forces acting on the beam, and (b) the forces acting on a discretized element of the beam

In Figure 4, a diagram of the cutting tool is shown, where D_1 is the cutting edge diameter, D_2 is the shank diameter, D_3 is the minor shank diameter, L_1 is the overall tool length, L_2 is the cutting edge length, L_3 is the minor shank length, and L_4 is the taper length. The tool shank length is the distance from the taper to the collet.

$$L_{shank} = L_1 - L_4 \quad (6)$$

This theory was used to find the modal mass, stiffness and natural frequency of different micro and mesoscale tools, such as those shown in Figures 5 and 6. Tools with very slender shafts were selected since they are the most compliant and therefore have lower stability, allowing for the compliant platform to provide the most benefit.

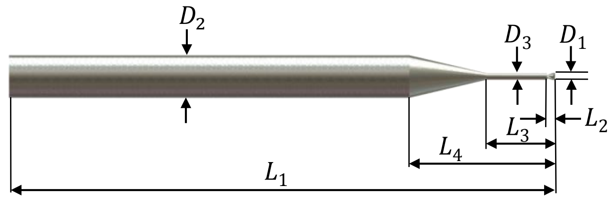

Fig. 4 Cutting tool dimensional variables

Following the FEM analysis of the tools, Figures 4 and 5, shows the natural frequencies of twenty one tools. It can be seen that the natural frequencies decreased as the shank length increased. Next, COMSOL simulations were performed in a batch that included 33 different tools. These simulations were carried out as in the above simulations over the length of the tool from the taper all the way up the longest possible shank length. For a 3 mm diameter shank tool, at least 14.5 mm of shank were required to hold the tool with a standard collet. In Figure 6, plots of natural frequencies with respect to the overall tool length, L_1, are shown for 33 different micro and mesoscale tools.

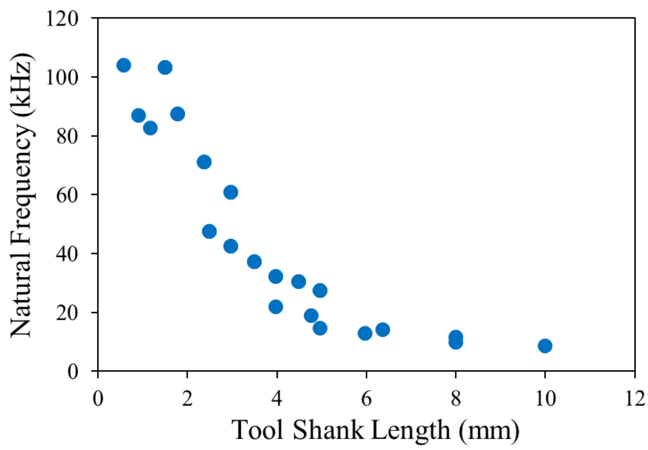

Fig. 5 Plot of first natural frequencies with respect to tool lengths (up until the taper) for a selection of 21 micromilling tools

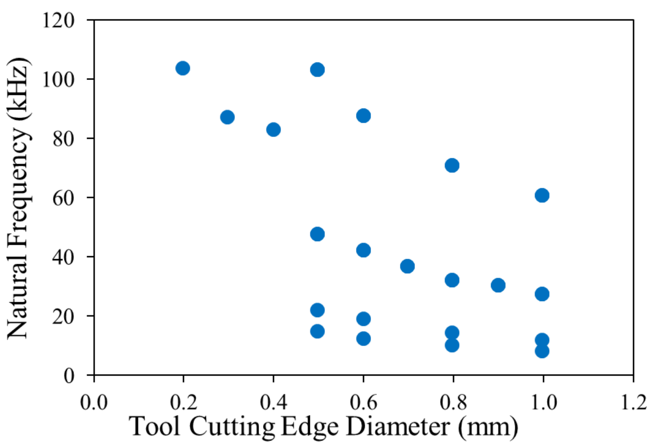

Fig. 6 Plot of first natural frequencies with respect to the cutting edge diameters for a selection of 21 micromilling tools

In Figure 7, it can be seen that the first natural frequencies of the tools converge into a relatively narrow path as the tool length increases regardless of the cutting diameter. When the tools were held close to the tapers, the tools exhibited unique dynamic characteristics that were based more on the specific geometries of the tools, including cutting diameter and minor shank length. As they were held farther away, there was a steep drop-off and the dynamics of the shank began to resemble that of a cantilever beam with approximately the same diameter as the tool shank.

The tool with the lowest natural frequency was a 1 mm tool, with a natural frequency of just 1351 Hz when it was held at its maximum shank length of 47 mm. This piece of information was found to be very important in later steps of the research since the natural frequency of the fixture design can be severely limited, especially if the tunable fixturing design incorporates a movable platform to support the workpiece. Since the 1 mm tool had the lowest natural frequency, as well as a relatively large modal mass, it was the best candidate for finding a technological window for the application of the fixture tuning approach using the proposed tunable fixturing design shown in Figure 1. Alternate tunable fixturing approaches will yield different technological windows.

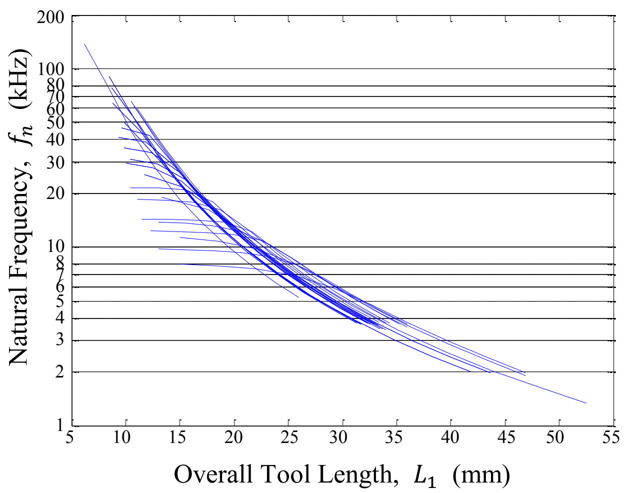

Fig. 7 Plot of natural frequencies with respect to the overall tool length, L_1

While the previous analyses determined only the natural frequency of the first mode of the tools, for the purpose of determining the stability of the system, under the hypothesis of a stiff workpiece, it is necessary to calculate the dynamic compliance of the tool, including all of its natural frequencies. After all, it is theoretically possible also for the higher order modes to be dominant in some cases. However, for all of the tools tested, it was found that the first mode was, in fact, dominant.

In order to find the dominant mode of the tools, frequency response functions (FRFs) were found for each tool over a wide range of natural frequencies. The frequency response function for a one degree-of-freedom system with linear stiffness, constant mass and viscous damping is given below.

$$\frac{X}{F} = \frac{1}{k}\left(\frac{1-r^2-2\zeta r}{(1-r^2)^2+(2\zeta r)^2}\right) \tag{7}$$

In this equation, the frequency ratio, r, is the spindle speed divided by the natural frequencies and ζ is the damping ratio.

$$r = \frac{\omega}{\omega_n} \tag{8}$$

$$\zeta = \frac{c}{2\sqrt{km}} \tag{9}$$

In these analyses, the damping value was chosen to be 0.015, based on the isotropic structural loss factor of steel. The equivalent diameter of the cutting tool due to the helical geometry of the cutting edge was taken into account. The equivalent diameter of endmill tool at cutting end was approximated to be 0.80 times the cutting diameter [7]. The frequency response functions for each natural frequency were then added to produce the true frequency response function.

The frequency response function for each tool was then determined for a range of different tool lengths, as in the previous analysis but with the dynamic response instead of just the first natural frequency. For each of these frequency response functions, the minimum real value, as the most important parameter with regard to the regenerative chatter condition, was determined. In order to understand why the minimum real value is the most important parameter, one must refer to the equation for the critical depth of cut.

$$b_{lim} = \frac{-1}{2K_s \mathrm{Re}[FRF_{orient}]N_t^*} \tag{10}$$

In this equation, b_{lim} is the limiting chip width, FRF_{orient} is the frequency response function oriented perpendicular to the average tooth angle and then rotated by the force angle, K_s is the specific force of the material being cut, and N_t^* is the average tooth per cut. In the above equation, it can be seen that at the minimum of the real part of the frequency response function, the limiting chip width will also be at a minimum. This point is called the critical depth of cut, $b_{lim,crit}$, and one method of maintaining stability is to simply stay below this value at all spindle speeds. By increasing the minimum real value of the FRF, the critical depth of cut can also be increased.

To demonstrate what the FRF of a micro or mesoscale tool looks like, a plot of the one found for a 1 mm tool is shown in Figure 8 below. The findings for 33 different tools were then plotted with respect to the overall tool length in Figure 9. As can be seen in the figure, the minimum real receptance of the tool follows a similar trend to the previous analysis of the first natural frequency, where the tools follow a narrow curve regardless of the cutting diameter and minor shank length. This plot shows that overall tool length is a significant factor, if not the most significant factor, in determining the stability of the system. It should also be noted that the curve in the plot that extends furthest down is again the 1 mm tool.

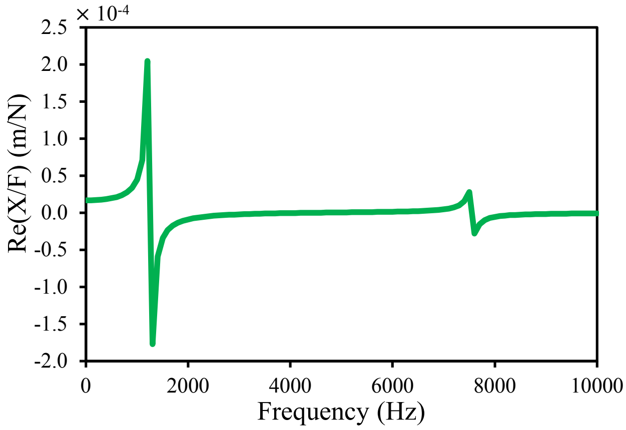

Fig. 8 Real part of the frequency response function of the 1 mm cutting tool

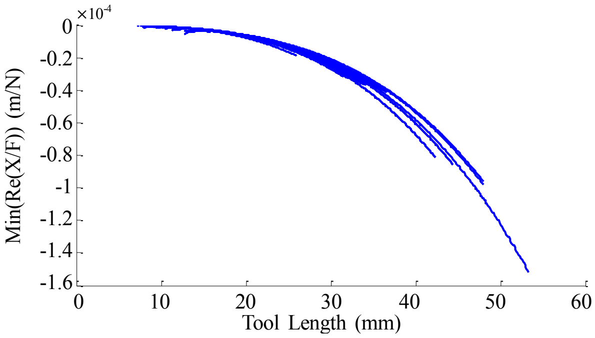

Fig. 9 Plot of minimum real values for a set of 33 tools

3. Experimental Testing

3.1. Electromagnetic Probe Tests

Validating the theoretical models for the cutting tool dynamics was complicated by the fact that the tools were too small to accommodate accelerometers. Even if a small enough accelerometer was available, it would cause a slight bias due to its mass. This problem was addressed by using an electromagnetic probe to measure the frequency response.

An electromagnetic probe detects the change in velocity of a ferromagnetic substrate the same way a guitar pickup detects the vibration of a metal guitar string. However, the probe is designed to be used on large, flat plates with well-known, magnetic properties. For an oddly shaped component, like a cutting tool, with unknown magnetic properties, the probe can only tell you the frequency of vibration, not the magnitude. However, other methods, such as the use of laser Doppler vibrometers, can measure neither the frequency nor the magnitude of the vibrations. The advantage of using an electromagnetic probe is that it has zero contact forces, it causes only minimal damping due to electromagnetic eddy currents, and imposes no mass loads on the cutting tool. This allowed for an accurate measurement of the natural frequency of the tool to be taken, which was used to validate the finite element methods.

In the electromagnetic probe experiments, a 2 mm tool (R216.32-02030-AC60P 1630) was clamped with a total length of 27 mm. A PCB 086E80 impact hammer with a hard tip was used to strike the tool just above the taper and a JP/1412 SN 5977 probe was used to pick up the vibrations. The resulting vibrations were given in volts and were normalized to the impact hammer force, giving units of 1/N. A picture of the experimental apparatus is given in Figure 10. The magnitude of the frequency response function was plotted over a range of zero to 20 kHz.

In Figure 11, the plot of the resulting FRF is given. It can easily be seen that the dominant mode has a natural frequency of about 12,950 Hz. This was compared to a COMSOL Multiphysics eigenfrequency analysis of the cutting tool, which yielded a first natural frequency of 12,829 Hz. Two other peaks in the electromagnetic probe response were present at 3430 Hz and 5350 Hz. However, it is unknown what caused them. The similarity in the simulated and experimental results shows that the first natural frequency of a cutting tool can be accurately predicted using finite element models.

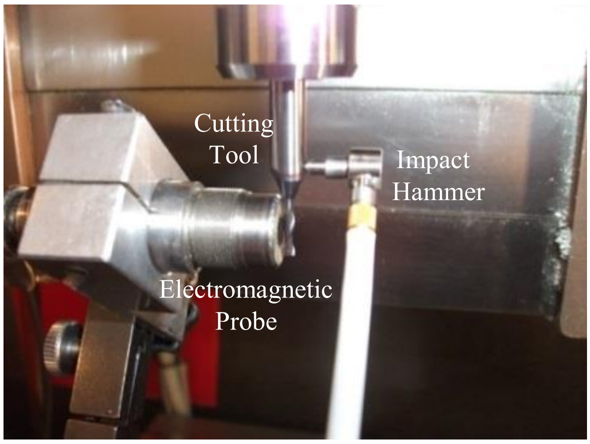

Fig. 10 Experimental apparatus for electromagnetic probe testing of mesoscale tool frequency response

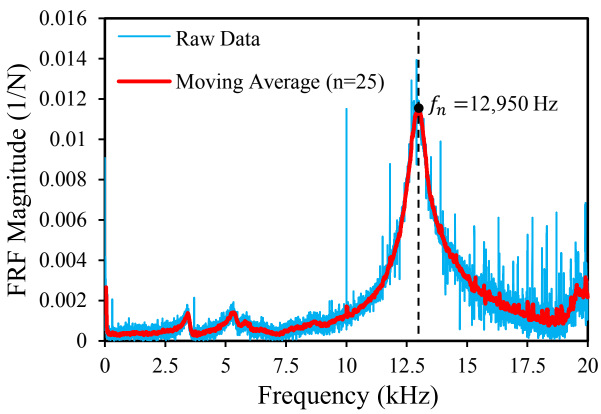

Fig. 11 Response of mesoscale cutting tool using an electromagnetic probe and impact hammer, with moving average of sample size 25 (sample range of 50 Hz)

3.2. Platform Dynamics

Using COMSOL Multiphysics, the natural frequencies of the platform for three different beam thicknesses were calculated. In Figures 1 and 2, these beams are shown supporting the center bar of the platform. The springs, also shown in Figure 2, can each apply 266.5 N of tensile force when fully compressed, so the natural frequencies were calculated for both 533 N and zero tension.

Additionally, impact hammer tests were performed for the stacked platforms with 0.500 mm thick beams at zero and 533 N of tension. The bottom platform was tested with zero tension using a PCB 086C01 hammer with medium tip covered by vinyl cover. The bottom platform was also set to have zero tension and it was assumed that the axial compliance of the bottom platform was not significant enough to affect the natural frequency of the top in the transverse direction. The hammer struck the stacked platforms on the side of the top platform with an accelerometer placed on the opposite side of the top platform. The top platform was then tested by striking the center bar end cap with a smaller hammer, PCB 086E80, with a vinyl cover. The tensionizing screws were then fully tightened to provide 533 N of tensile force in the beams and the tests were repeated. The magnitude and phase of the responses are shown in Figures 12 and 13 below.

It can be seen that the first natural frequencies of the platform are clearly visible in the plot. These natural frequencies were compare to those found using beam theory as well as finite element analyses in COMSOL Multiphysics. The results, shown in Figure 14, show that the theoretical methods slightly overestimated the natural frequency when no tension was applied.

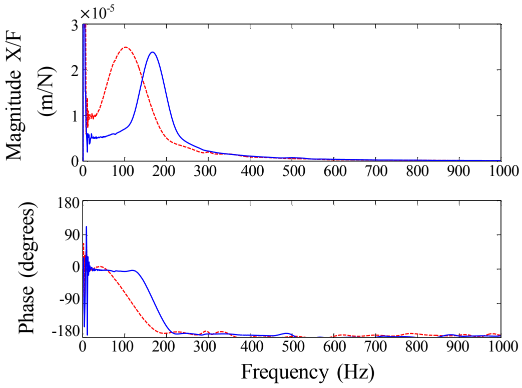

Fig. 12 Magnitude and phase of the frequency response of the bottom platform in the stacked configuration for both zero (dashed) and 533 N of tension (solid)

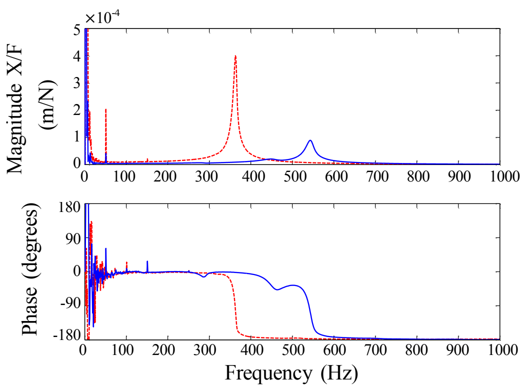

Fig. 13 Magnitude and phase of the frequency response of the top platform in the stacked configuration for both zero (dashed) and 533 N of tension (solid)

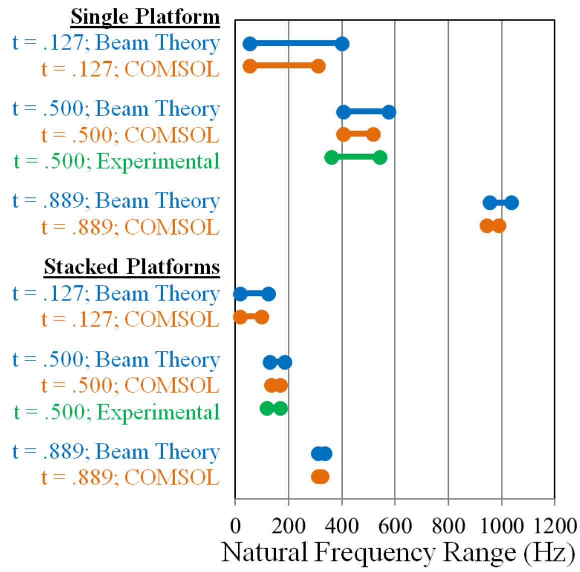

Fig. 14 Comparison of COMSOL, beam theory and experimental results for first natural frequency of the fixturing platform

Another finding from the above plot was that the range of stiffness values was very low for thick beams. In order to tune the platform over a wide range of frequencies, a thin beam is preferred. However, this must be balanced with the need to achieve a natural frequency higher than the frequency at the minimum real value of the frequency response function for the machine tool. Furthermore, the frequencies of the platform were too low since the lowest natural frequency of any of the tools was 1351 Hz while the highest natural frequency found in the platform analyses was 1035 Hz. In order to address this issue the mass of the platform would have to be significantly reduced.

4. Technological Window

Having found the modal properties of a range of different micro and mesoscale tools, the design of the fixturing platform can now be evaluated. During the preliminary analyses, it was found that the beam would have to be far thicker than what was designed for in the experimental apparatus. As a result, a modified apparatus with reduced center bar mass was developed and the resulting frequency range was determined. A side-by-side comparison of the unmodified and modified center bars is shown in Figure 15. By removing material, the part mass was reduced by more than 50% while the structural integrity was still maintained.

A parameter sweep of beam thickness was applied to determine how thick the support beams in the fixture would need to be to produce a higher natural frequency than that of the tool for a fixture with the modified center bar. The results of the analyses of the modified platform and the 1 mm tool were then plotted together in Figure 16, which shows that the window of opportunity exists for a platform with beam thickness of greater than 1 mm.

For optimal performance, the maximum real value of the fixture FRF must be located at the same frequency as the minimum real value of the tool FRF. Furthermore, it is desired that the maximum real value of the fixture FRF is as large as possible. While the modified platform was able to achieve natural frequencies higher than that of the tool, and could thus mitigate regenerative chatter, the effects were insignificant.

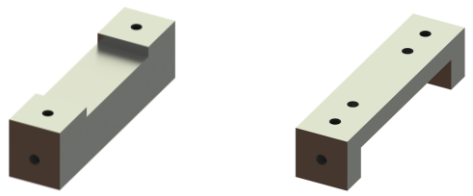

Fig. 15 Comparison of unmodified (left) and modified (right) center bars, showing the material that was removed in order to reduce the mass of the part

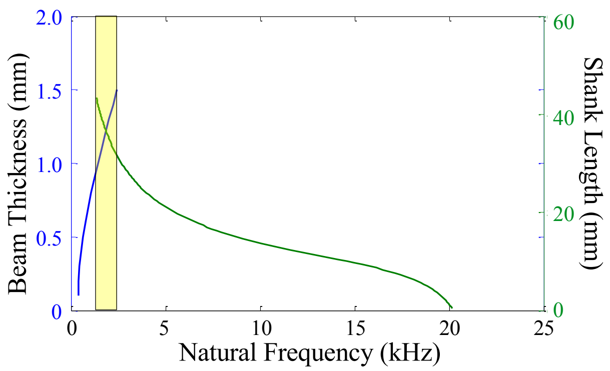

Fig. 16 Window of opportunity where natural frequency of platform is higher than that of tool, between 1351 and 2436 Hz

In order to address the lack of dynamic compliance in the platform, one final analysis was performed in which the center bar mass and workpiece mass were set to zero. This scenario represented what would happen if a workpiece with close to zero mass was attached directly to the support beams, in place of the center bar. In Figure 17, plots of the real FRFs for the 1 mm tool from the previous sections, an idealized fixture, as well as the combined FRF are shown. It can be seen that if the center bar mass was somehow reduced to zero and the workpiece was arbitrarily small, a significant improvement in the minimum real FRF can be achieved.

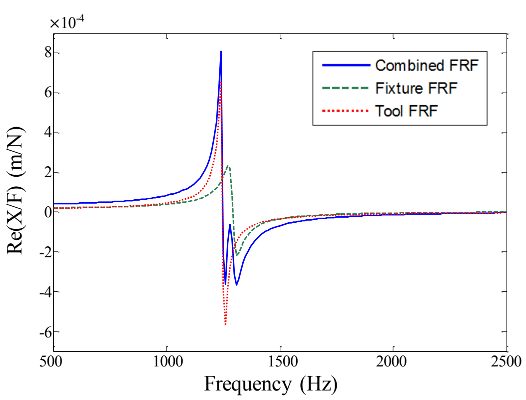

Fig. 17 Plot of tool, fixture and combined receptance with respect to frequency for an idealized fixture and the 1 mm tool

5. Discussion

While the goal of mitigating regenerative chatter using stiffness tuning has yet to be accomplished, this research has provided some useful insight on how it could be achieved. By modifying or redesigning the fixturing device, with a focus on minimizing its modal mass, it is possible that this approach could be validated. Furthermore, the information gathered about micro and mesoscale cutting tools will provide a useful foundation for the next iteration of tunable fixture design.

One interesting approach for a future design would be to mount the workpiece on a pin and tune the dynamics of the pin by modifying its length. This approach would completely negate the problem of center bar mass since the workpiece would be mounted directly onto the compliant member. If this approach is to be attempted, it is likely that the pin should have a diameter and material properties similar to that of the cutting tool. Then, the length of the pin should be made slightly shorter than that of the tool. This would allow the pin to have a higher natural frequency than the cutting tool with similar levels of compliance.

ACKNOWLEDGEMENT

This work was made possible by a generous travel grant from CNR-ITIA, and with guidance and assistance from Francesco Cacciatore, Massimiliano Annoni, Massimo Goletti and Marco Leonesio.

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