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Design optimization of tool holder extension for enhanced chatter stability by using component mode tuning method

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Abstract

Tool overhang length tuning by matching the modes of tool-extension-holder-spindle (TEHS) assembly components creating dynamic absorber effect is one of the rare chatter suppression techniques. In some milling applications, a tool holder extension may have to be used to increase the overall tool overhang length for reaching far sections of the workpiece. In this work, design parameters of the holder extension are optimized to maximize the dynamic absorber effect resulting from modal interactions among the components of the TEHS assembly. Improvements in the dynamic rigidity and chatter free Material Removal Rate (MRR) are demonstrated through simulation results.

Keywords: Chatter stability; Machine tool dynamics; Milling

1. Introduction

Chatter, or self- excited vibrations of the cutting tool, is a major problem in machining processes which results in poor surface finish and decreased life of the tool and mechanical components, thus limited productivity. Recent advances in the aerospace and automotive sectors demanded more powerful, precise and rigid machine tools. As a result, suppression of chatter vibrations remains as an essential problem in machining industry [1].

Chatter free conditions can be predicted and avoided by using stability lobe diagrams which provide stable depth of cut and spindle speed combinations. It is well known that critical depth of cut, b_{lim}, is inversely proportional to negative real part of the oriented tool tip FRF [2]. Thus, b_{lim} can be increased by decreasing the amplitude of the tool tip FRF. One way to decrease it is to create dynamic absorber effect among machine tool's components [3].

Tool overhang length tuning to create modal interaction between spindle-holder and tool assembly was first studied by Schmitz [3]. He matched the fundamental natural frequency of the cantilevered tool with one of the natural frequencies of the spindle-holder subassembly. As a result, the tool tip FRF was split into two smaller modes around the cantilevered tool mode, similar to the absorber effect created by tuned mass dampers. Similarly, Ertürk et al. [4] showed that it was possible to create dynamic absorber effect among the components of the tool-holder-spindle (THS) assembly by varying their dimensions. However, these procedures required trial and error. Later, Mohammedi et al. [5] developed a systematic and generalized methodology to optimize the tool length for maximized dynamic~~s~~ rigidity. They also included the tool holder dimensions, and their optimum selection, into the analysis since in some machining applications, it is not possible to shorten the length of the tool to create sufficient dynamic interaction resulting in increased dynamic rigidity. For those cases, on the other hand, standard tool holder geometries can be used in the same algorithm to determine the one which creates sufficient dynamic interaction, and thus increased chatter stability [5]. Similarly, spindle dimension tuning can be employed in the design stage to increase dynamic rigidity of the spindle assembly [5].

In milling applications where long tool length is necessary due to far sections on the workpiece, an extension piece can be used between the tool holder and the tool in order to increase the overall effective length without increasing the tool cost due to long length which often requires custom made tool. In this paper, length of the holder extension is optimized to create dynamic absorber effect among the components of the TEHS assembly. Moreover, both holder extension length and tool overhang length are optimized simultaneously to maximize the dynamic rigidity of the assembly as a result of modal interaction. The holder extension can be implemented and tuned in a machine tool assembly where it is not possible to find a standard holder which creates dynamic interaction among the components or when it is not practical to change spindle design.

2. Dynamic Modeling of TEHS assembly for prediction of tool tip FRF

Dynamic response of the TEHS assembly (see Fig. 1) can be predicted by using Receptance Coupling Substructure Analysis (RCSA) [6]. In this method, FRFs of the individual components can be calculated analytically by using free-free Timoshenko beam formulation and then they can be coupled elastically by considering interface dynamics. Furthermore, bearing dynamics can be added to the spindle at the bearing locations by using the structural modification method [6].

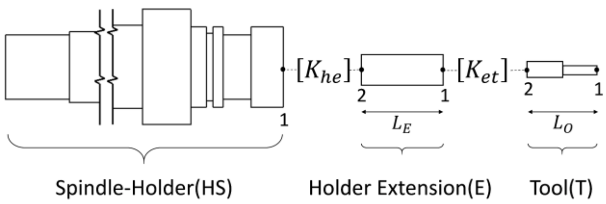

Fig. 1. TEHS assembly.

The receptance matrix of a free-free Timoshenko beam can be formulated as [6]:

$$[B_{ij}] = \begin{bmatrix} H_{ij}^{B} & L_{ij}^{B} \\ N_{ij}^{B} & P_{ij}^{B} \end{bmatrix} \tag{1}$$

where H, L, N, P are the receptance functions defined in the following relationships [6]:

$$H_{ij} = \frac{y_i}{F_j}; L_{ij} = \frac{y_i}{M_j}; N_{ij} = \frac{\varphi_i}{F_j}; P_{ij} = \frac{\varphi_i}{M_j} \tag{2}$$

where y_i and φ_i are the translational and rotational displacements at coordinate i, respectively. F_j and M_j are the applied force and moment at coordinate j, respectively.

Using RCSA, the receptance at the holder extension tip can be calculated as follows:

$$[EHS_{11}] = [E_{11}] - [E_{12}]\left([E_{22}] + [K_{he}]^{-1} + [HS_{11}]\right)^{-1}[E_{21}] \tag{3}$$

Here, $[HS_{11}]$ represents the FRFs of the spindle-holder subassembly at the holder nose, $[E_{11}]$, $[E_{12}]$, and $[E_{22}]$ represent the direct and cross FRFs of the holder extension between holder and tool, respectively. Also, $[K_{he}]$ .represents the joint dynamics between holder and holder extension.

Next, tool tip FRFs can be calculated as follows:

$$[TEHS_{11}] = [T_{11}] - [T_{12}]\left([T_{22}] + [K_{et}]^{-1} + [EHS_{11}]\right)^{-1}[T_{21}] \tag{4}$$

Similarly, $[T_{11}]$, $[T_{12}]$ and $[T_{22}]$ represent the direct and cross FRFs of the tool respectively. $[K_{et}]$ is the interface dynamics between holder extension and tool. Details of the interface dynamics are provided in [6].

3. Tool tip FRF optimization

Objective of this study is to create a modal interaction between the substructures of the spindle system. Various optimization techniques have been used for design of dynamic vibration absorbers. In this study, the goal is to find a suitable set of design parameters which minimizes the maximum amplitude of the tool tip FRF (defined by transverse displacement and force) in the selected frequency range. Optimum design parameters are searched in the practical lower and upper limits. The objective function θ used in the optimization can be expressed as follows:

$$\begin{gathered} \theta(x) = \max\left(\left|H_{11}^{TEHS}\right|\right) \\ x_{\min} \le x \le x_{\max} \end{gathered} \tag{5}$$

Here, x represents the design parameters to be optimized. The optimization problem can be either one or multi variable case.

3.1. Case 1: Holder extension dimension tuning

In applications where the tool holder cannot be changed and the tool should be kept long enough, holder extension dimensions can be tuned to reduce the peak FRF at the tool tip. Similar procedure in [5] can be utilized as the tuning algorithm. Firstly, one has to choose the effective mode at the tool holder nose which is capable of being matched with the dominant mode of the assembly [5]. Later, the length of the holder extension is tuned to shift the fundamental natural frequency of the cantilevered holder extension and tool (TE) subassembly to the effective mode at the holder nose. As a result, sufficient modal interaction between the spindle-holder subsystem and holder extension is created and the critical mode of the assembly is damped.

3.1.1. Simulations and numerical results

A TEHS assembly described in Table 1 and 2 is considered herein. The dimensions of the spindle and holder are the same as the ones considered in the analytical case study in [6]. The materials of all the components are steel with mass density of $\rho = 7800\ kg/m^3$, Young modules $E = 200\,GPa$, Poisons ratio $\upsilon = 0.3$ and the material loss factor $\gamma = 0.003$. The tool tip FRF H_{11}^{TEHS} is calculated and a dominant mode at around 1700 Hz is observed (see Fig. 2. a.). Later, the effective modes at the holder nose are investigated. As it can be seen from Fig. 2b., there are several effective modes, which are at 1010, 1930 and 3620 Hz and they can be used to create modal interaction within the assembly.

Table 1. Dimensions (in mm) of the assembly components

	Extension Dimensions	Tool Dimensions	
Segment Number	1	1	2
Length	40	24	26
Outer Diameter	16	6.5	8
Inner Diameter	8	0	0

Table 2. Dynamic properties of the interfaces

	SH interface	HE interface	ET interface
Translational stiffness$[N/m]$	8×10^7	2×10^7	0.75×10^7

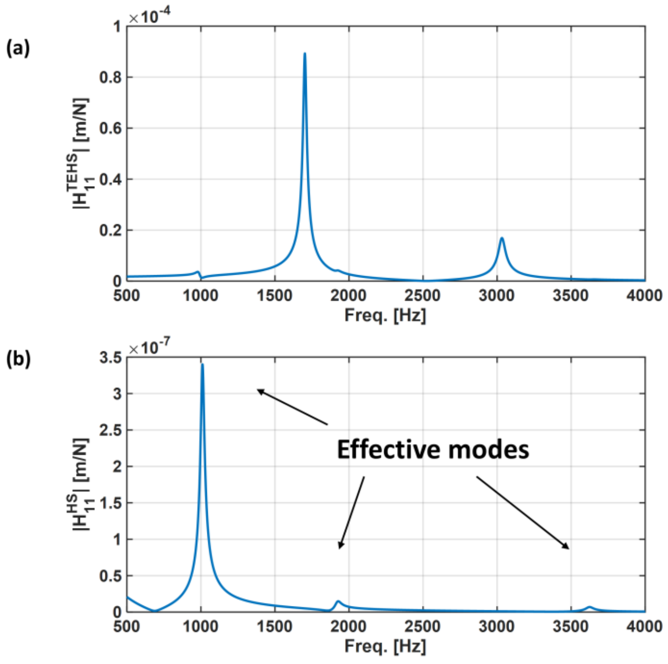

Fig. 2. Calculated tool tip FRF with 40 mm long holder extension and 50 mm tool overhang (a) Effective modes at the holder nose while holder extension and tool are not clamped(b).

The optimum holder extension length is searched between 20 and 40 mm. The maximum amplitude of the tool point FRF is minimized when the length of the holder extension is 24 mm. Using the tuned-holder-extension length, tool tip FRF split into two smaller peaks (see Fig. 3). Blue dashed line shows the FRF of the TE subassembly elastically coupled to the wall with connection dynamics $[K_{he}]$. The figure shows that a modal interaction between the spindle-holder and holder extension-tool subsystems are created damping the tool tip FRF.

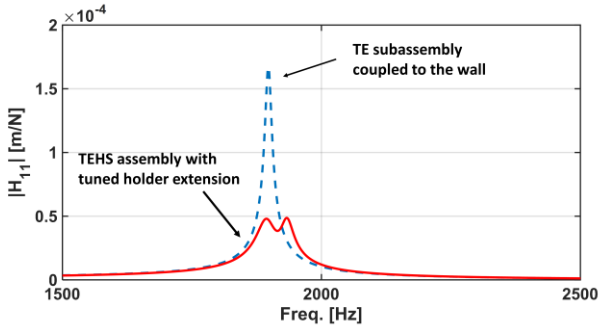

Fig. 3. Calculated tool tip FRF with optimized holder extension length.

3.2. Case 2: Holder extension and tool dimension tuning

It is possible to tune the fundamental frequency of the holder extension and tool subsystem by using different combinations of the dimensions of the holder extension and tool.

The procedure in the previous case can be used here as well. This time, the overhang length of the tool is also considered as an optimization parameter.

3.2.1. Simulations and numerical results

The same assembly used in case 1 is considered here. The peak of the tool tip FRF is calculated and plotted for different holder extension length (L_E) and tool overhang (L_O) combinations in Fig. 4. The optimum parameters which created a modal interaction found to be 47 and 39 mm, respectively.

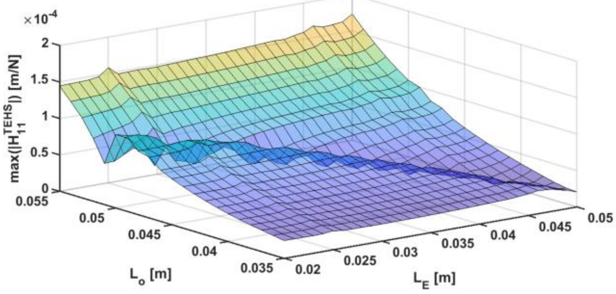

Fig. 4. Change in the peak tool tip FRF due to design parameters.

Fig. 5 shows the optimized tool tip FRF with multi-component dimension tuning which is obtained by simultaneous optimization of L_E and L_o in this case. As it can be seen, the peak amplitude of the tool tip FRF is decreased from $0.5 \times 10^{-4}\ m/N$ to $0.2 \times 10^{-4}\ m/N$ compared to the previous case. Consequently, the tool tip FRF is decreased more than two times by using multi-component dimension tuning.

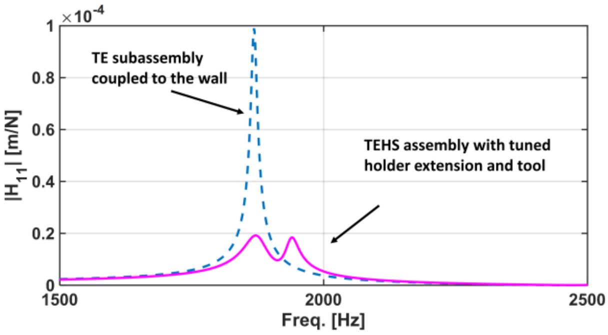

Fig. 5. Calculated tool tip FRF with simultaneously optimized holder extension length and tool overhang length.

4. Comparison of dimension tuning for THS and TEHS assemblies

In this part, the optimized tool tip FRFs and corresponding stability lobe diagrams for THS and TEHS assemblies are compared.

To tune tool tip FRF for THS assembly, the same spindle and holder used in the analytical case study in [6] are used here. Then the tool specified in Table 1 is clamped to this spindle-holder assembly. The interface dynamics between the holder and the tool connection is assumed to be the same as the interface dynamics between the holder extension and tool connection.

The optimization problem becomes:

$$\theta(L_O) = \max\left(\left|H_{11}^{THS}\right|\right)$$
$$L_{O_{\min}} \leq L_O \leq L_{O_{\max}} \tag{6}$$

The procedure used in [5] is used here as well to find the optimum tool overhang length. The peak at 1930 Hz at the holder nose is selected as the effective mode for damping purposes. The optimum overhang length of the tool is found to be 55.5 mm. The tool tip FRF for the THS assembly with arbitrary tool overhang and tuned tool overhang lengths are compared in Fig. 6. The tuned tool tip FRFs for TEHS assemblies in the previous cases are also included in the figure. The corresponding stability lobes are given in Fig. 7. There is a noticeable improvement in the limiting stability due to modal interactions in the existence of extension piece. This result shows that there is a considerable potential to suppress dominant mode, hence improve stable MRR, by adding a tuned component between the holder and the tool. It is shown that, the tool tip FRF and limiting stability of the milling applications can further be improved by multi-component dimension tuning in TEHS assemblies.

5. Conclusion

In this paper, peak FRF of the tool tip is suppressed by creating dynamic absorber effect in TEHS assembly by matching the fundamental frequency of TE subassembly with the effective mode at the holder nose. First, length of the holder extension is optimized to create modal interaction. Then, both holder extension length and tool overhang length are included as design parameters in the optimization problem. By multi component dimension tuning, the peak FRF of the assembly is suppressed further. Finally, tool overhang length is tuned for THS assembly in which dimensions of the components are the same with the ones in TEHS assembly. The reductions in peak FRF at the tool tip and the improvements in the limiting stabilities are compared. It is observed that existence of holder extension and proper tuning of holder extension and tool dimension result in a dynamically stiffer system. As a result, a considerable improvement in the stable MRR is achieved.

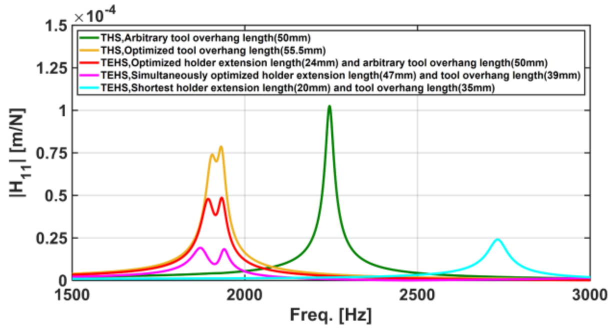

Fig. 6. Tool tip FRFs with optimized component dimensions.

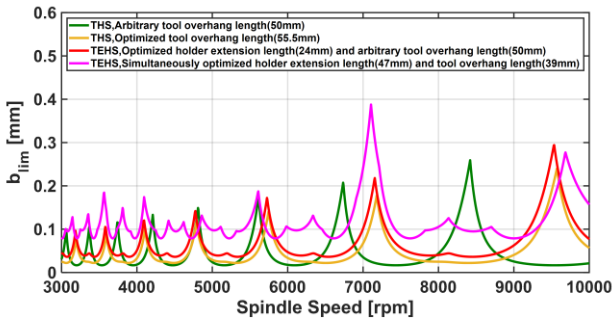

Fig. 7. Stability lobe diagrams.

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