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Machine tool design based on dynamics of design-for-assembly (DFA)

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Abstract

Machine tools is a resilient system which response to the dynamic cutting forces in the form of vibrations during a typical cutting operation. Presently, machine tool design is limited to the development of the optimum design for individual components such as base, column, milling head, etc. Sometimes, analysis tools such as static/dynamic characteristics and topology optimization are used shaping up the final designs of these components. However, it is commonly observed that the component designed based on individual dynamic analysis change its frequency of vibration, stiffness and, damping as it becomes part of a machine tool assembly. Therefore, in present work, a vertical machining center has been chosen for the development of design-for-assembly considering dynamics of assembled machine components instead of individual ones. The methodology consists of an experimental analysis of tooltip Frequency Response Function (FRF) of a prototype machine. It is followed by experimental modal analysis. Successively, the sensitive FRF frequencies which directly affect the machining quality are correlated to the corresponding modal shapes of the assembled components. The finite element (FE) based topology optimization then carried out for the selected components to modify the dynamic characteristics of the complete machine. Further, the joint dynamics were optimized in the form of boundary conditions for the mating surfaces of selected components towards the development of assembly guidelines. The components then manufactured and assembled based on the design and assembly guidelines to achieve a dynamically superior topology optimized machine tool system.

Keywords: machine tool, design-for-assembly, dynamics, modal analysis, FE analysis, FRF, topology optimization

1. Introduction

A set of standards ISO 14955 has issued by International Organization for Standardization (ISO) to guide machine tool designers to design high-efficient machine tools (ISO 14955-1, 2017). The stiffness of machine components were given utmost importance designing the machine tool system. A traditional theory of machine design has been developed based on static performance analysis of the machine components. However, the vibrations associated due to the dynamic cutting attracted researchers to analyze its dynamic characteristics. The design and development of machine structures based on static analysis proves inadequate. The optimum balance between the mass and the stiffness of the component along with the stiffness of the joints between various machine components decide the dynamic behaviour of a typical machine tool. Therefore, the progressive trend in machine tool design seems to be transited from the static analysis towards the mass-stifness based dynamic analysis and thereby optimising the topology and shape of the machine tool components. The recent advancements in the FEA tools made it possible to design the machine tools considering the periodic cutting loads and their effect on the natural vibrational frequencies of the machine components. Therefore, the present machine design approach consists of designing individual machine tool components considering their dynamic response to the periodic cutting loads by minimising the mass and enhancing the structural stiffness.

The recent studies show a significant numbers of research focused on the design of machine tools based on its dynamic response rather depending only on its static structural behaviour [1-9]. Li *et. al*. [10] were one of the most recent groups to develop a dynamics based structure optimization approach for lightweight moving parts of the hobbing machine. An interesting biologically inspired topology optimization methodology was proposed by Li *et. al*. [11] where the column ribs were optimized to about 20% improvement in the dynamics of a grinding machine.

The present research methodology involves a combination of experimental and predictive methods to enhance the dynamic performance of a typical machine tool. The experimental analysis of tool-tip frequency response function (FRF) was performed to define the significant natural frequencies of vibrations along with their associated amplitudes and damping factors. An FEA model was developed to predict the natural frequencies of vibrations and verified with the experimental tooltip FRF results. Further, the experimental modal analysis of the complete machine structure resulted into the physical manifestation of the mode shapes associated with the dominating natural frequencies obtained earlier. It helped define the machine tool component in need of the design modification. Finally, the topology optimization technique has been used for achieving the improved stiffness-mass ratio of the target component and thereby enhancing the performance quality of a complete machine tool system.

The rest of the paper is organized as follows: Section-2 discusses the experimental and predictive techniques such as tool-tip FRF, modal analysis and FE based frequency analysis to define the dynamic characteristics of a machine tool. Section-3 represents the design optimization of spindle head structure as a function of natural frequencies of structural vibration. The design analysis of the modified spindle head structure is represented in terms of its static and dynamic characteristics in Section-4. Finally, the Section-5 incorporates the concluding remarks and the future scope of the research.

2. Dynamic analysis of the existing machine tool

2.1. Tool-tip Frequency Response Function (FRF)

A Bharat Fritz Werner India Ltd. BMV-45 series vertical machining center (VMC) has been selected for performance analysis subjected to its demand in Indian machine tool market. This machine targets the Indian die and mold industries for its application. An impact test has been performed with a 16 mm diameter 4 fluted coated carbide flat end mill cutter having 35 mm tool overhang as shown in Figure 1*a* inset. The experimentally obtained tool-tip FRF plot represents a combined dynamic behaviour of a machine with all its components and joints. Figure 1*a* represents a frequency v/s compliance plot as a result of an impact test performed at the tooltip. Frequency response of the machine tool is represented in the form of linear or logarithmic plots of transfer function (m/N) against natural frequency (Hz). A sharp peak is observed at 142 Hz which is dominating in X-X direction. The peaks within the range of 0-500 Hz were considered significant based on their direct contribution to the machining operation. However, the other peaks were neglected based on their relatively lower magnitudes of compliance. Further, a 3D model of spindle head was subjected to the FE analysis using COMSOL software. The details of input boundry conditions to the analysis are presented in Table 1. A dynamic analysis was run for the frequencies over the range of 0-500 Hz using frequency analysis. The results were plotted in the form of total displacement v/s frequency as shown in Fig. 1*b*. The predicted frequencies found matching with the experimental FRF peaks.

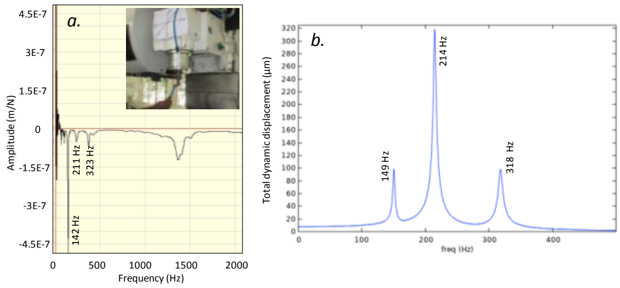

Figure 1. *a*. Experimental tool-tip FRF plot with weaker frequency mode at 142 Hz; inset: a photograph of tapping test at tool –tip, *b*. Predicted FRF at spindle head surface using dynamic analysis using COMSOL

Table 1 Details of boundry conditions to analyze spindle head

Specification	Value
Spindle motor weight (Kg)	45
Declamp assembly weight (Kg)	30
Spindle weight (Kg)	40
Maximum cutting forces at tool tip considering machining of HRC 48 workpiece material (N)	$F_x = 540$
	$F_y = 480$
	$F_z = 25$

2.2. Modal analysis of VMC

In order to know the dynamic behaviour of machine at 142 Hz which was detected as the weakest natural vibrational frequency, a modal analysis test was performed. A certain reference point on VMC base was excited by impact hammer having force transducer. The vibration spectrum in Cartesian coordinate system was measured by positioning tri-axial accelerometers at around 250 various locations on the entire-

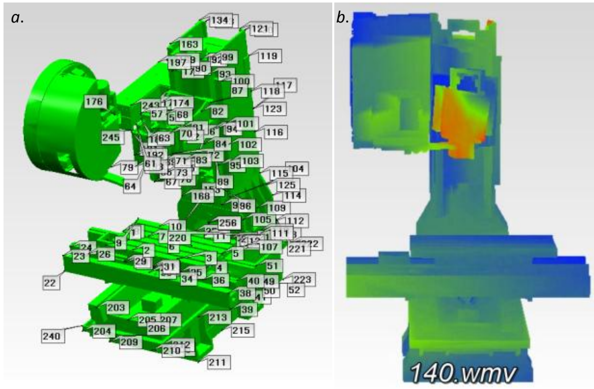

Figure 2. *a*. Accelerometer positioning for Modal analysis of VMC, *b*. A spindle head swing in X-X direction associated with 140 Hz

machine tool surface, see Figure 2*a*. The analysis of 3D model of a machine tool helped define the mode shapes and the associated machine tool components. A dominating mode shape at 140 Hz is recorded and found to be associated with the swing of the spindle head in X-X direction in Figure 2*b*. It nearly matches with the 142 Hz frequency obtained at tool-tip FRF (Figure 1a).

The tool tip FRF and the modal analysis of the existing machine resulted into the problem definition. The weakest modal frequency which was observed during the experimental and FE analysis was found to be associated to the swing of the spindle head in X-X direction. Most of the other modal shapes were found less significant and were neglected as design inputs. A combination of FRF feedback and modal analysis along with other design tools such as topology optimization and design applicability was used to modify the existing spindle head.

3. Design of spindle head

The tool-tip FRF and the modal analysis of VMC has defined the need of modification in the design of spindle head. Furthermore, it is observed that the weakeness of the spindle head in X-direction is resulted into the swing mode of vibration. Furthermore, it is observed that the swing starts from the mid-portion of the spindle head keeping the spindle head-column joint unaffected. This confirms the source of the mode as spindle head structure rather than the joint between the spindle head and the column. Nevertheless, the spindle head is found stiffer in Y-direction in absence of any vibrational mode observed in experimental modal analysis. Therefore, "stiffness enhacement in X-direction" is set as an objective for the design of spindle head. The enhanced stiffness-mass ratio of the component has also been set as a second goal towards cost-effectiveness. A topology optimization technique has been used to maintain desired stiffness-mass ratio. A spindle head structure without assembled components is shown in Figure 3*a*. A set of boundary conditions imitating the practical loads and pressures was applied to the 3D CAD model of a grey cast iron (G4) made spindle head. A frequency based FEA simulation for modal analysis is designed and run. The FEA results show a proximity with the practical mode shapes of the spindle head structure. Similar to the experimental observations, a swing mode in X-direction was observed at 140 Hz which ensures the confidence level of the FEA model.

3.1. FRF based design modification of spindle head

The definitive objective towards strengthening the spindle head in X-direction leads to the design modification of the-

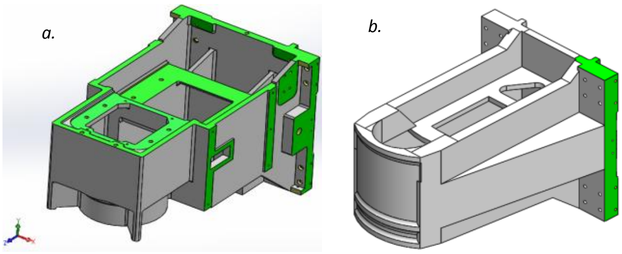

Figure 3. *a*. Existing 3D model of a spindle head, *b*. Version-1: modified spindle head design based on experimental and predicted FRF feedback

spindle head. A set of design guidelines were used while incorporating the design modifications in the spindle head structure considering its applicability.

- The present two step design is transformed into a single continuous structure from LM guide pads to the spindle mounting.
- The structural symmetry is maintained along X-axis.
- All ribs were designed with varying thicknesses i.e. thicker at the fixed joint near LM guideways and thinner towards the hanging part near the spindle mounting ring.
- A pair of ribs with varying thickness is provided both sides from the fixed to the hanging point of the spindle head structure to restrict X-directional swing.

A version-1 modification is depicted in Figure 3*b* where the X-directional ribs were provided. The version-1 was then simulated for FEA based modal analysis and thereby the purpose of introducing the ribs was justified. As a result, the amplitude of X-directional swing was reduced, however, except the first modal frequency (swing along X-axis) all other frequencies have got shifted to relatively lower frequencies as compared to the analysis of the existing spindle head.

3.2. Topology optimization of spindle head

The design guidelines have enhanced the stifness of the spindle head structure, however, the higher natural frequencies found to be lowered due to addition in the existing weight of the structure. The material addition only for the sake of improvement of the static stiffness might not prove as a right strategy for a spindle head subjected to the dynamic cutting loads. In this situation, the weight addition might cause the poor dynamic performance of the spindle head. Moreover, the assembled components such as, counter balance cylinder, encoder mounting, spindle motor, etc. exert direct weight on the spindle head further worsening its performance. Therefore, it is necessary to design the spindle head with lightweight and stiffer structure. This ultimately will help shift the structural modes of vibrations towards higher frequencies thereby, improving the structure dynamics and ultimately, the machining quality.

The version-1 after following the design guidelines mentioned in the previous sub-section has been analyzed for its topology optimization. A principle of minimum strain energy was followed while optimizing the structure for its mass and stiffness using optimization module in the FE analysis. An optimized version-2 is shown in Figure 4*a* and 4*b* where the FEA suggests the regions with the necessity for removal of material.

A final version of spindle head structure was designed based on the suggestion from the topology optimization analysis. A few design modifications were introduced to accommodate the assembly requirements on the spindle head structure. The final version which is shown in Figure 5*a* is a combination of the-

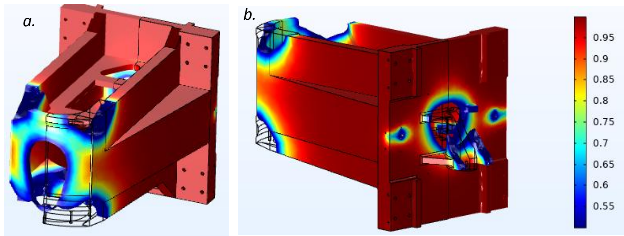

Figure 4. Version-2: Topology optimized spindle head structure *a*. Front and *b*. Back view

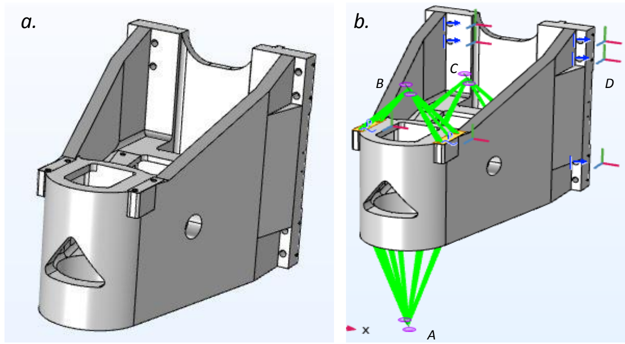

Figure 5. Version 3: A final modified spindle head *a*. 3D model and *b*. 3D model with the boundry conditions due to assembly

stiffness-mass optimization and the design-for-assembly approach.

Figure 5*b* represents the boundry conditions on the spindle head structure. The corresponding weights and forces of the assembly components were applied at the tool-tip *A*, encoder mouting bracket *B*, spindle motor mounting surface *C* and the LM pads butting surface *D* (see, Figure 5*b*).

4. Analysis and comparison of modified spindle head

A summary of static and dynamic characteristics of the existing, design guidelines based version-1, and topology optimization plus design-for-assembly based version-3 is presented in Table 2. An improvement in the dynamic characterstics was observed after following the design guidelines discussed earlier. Furthermore, a gradual trend of improved static and dynamic characteristics was observed sequencially from existing to version-3 design models. Figure 6 represents dynamic displacement plots as a function of modal frequencies for all the versions of the spindle head design. A clear distinction between the dynamics of existing structural design and the improved modified design was observed.

Table 2 Predicted dynamic characteristics of spindle head structures

Design version	Weight (Kg)	Static disp. (μm)	Freq. (Hz)	Dynamic disp. (μm)
Existing	160	12	149	95
			214	315
			318	100
V-1	185	9	168	55
			237	180
			360	70
V-3	175	5	267	12
			310	160
			475	60

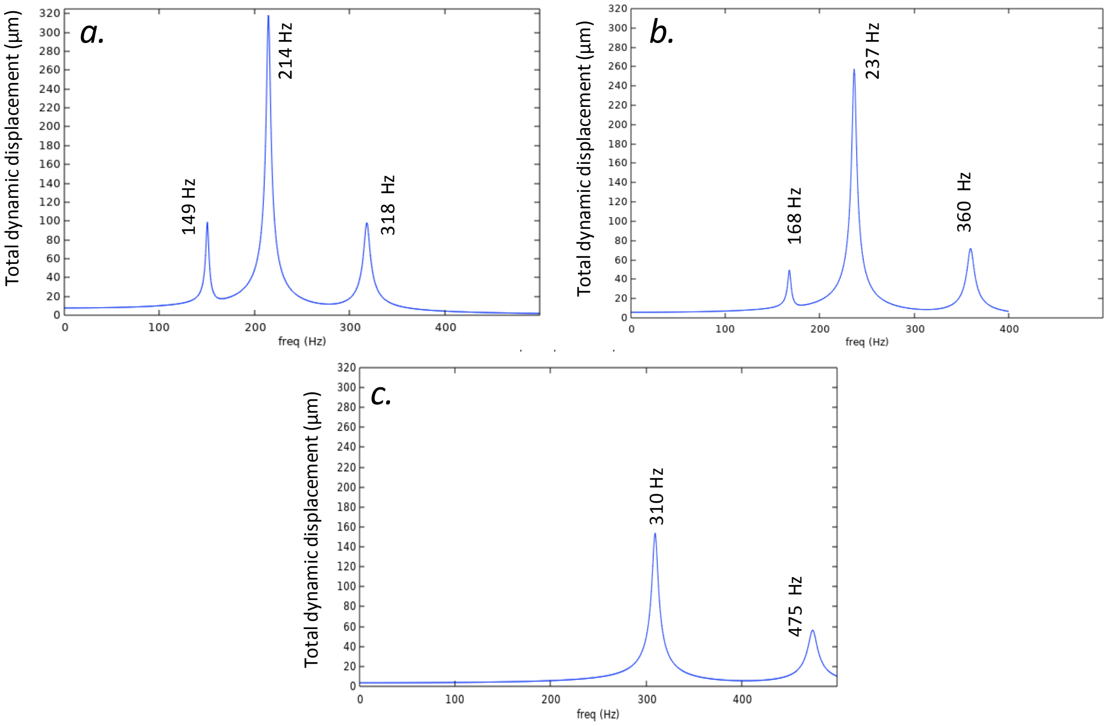

Figure 6. A comparison among the spindle head designs a. Existing, b. Version-1 and, c. Version-3

As compared to the existing spindle head design, the FRF oriented version-2 indicates the shifting of all the natural frequencies towards the higher side. However, the total dynamic displacement seems to be increased at all the frequencies which is not acceptable as a stable machining point of view. Nevertheless, the final version V-3, which is a product of topology optimization, shows higher frequencies at lower dynamic displacements. Therefore, the final design version of spindle head confirms a significantly improved design considering all three aspects such as, FRF feedback, modal analysis, design guidelines, topology optimization and the design-for-assembly.

5. Conclusion

The paper presents a reverse methodology where the existing machines can undergo the series of experimental and predictive design analyses to achieve the enhanced operational quality. A tool-tip FRF of an existing machine setup was obtained using a practical tap test. Further, the FE model has been developed and analyzed to validate with the experimental results. All the first three modal frequencies obtained using FE model show a good match with the experimentally obtained FRF results. This has confirmed the authenticity of the predictive FE model. Further, the experimental modal analysis test revealed the source of the vibrations in terms of swing of the spindle head in X-X direction. This helped decide the modification of the spindle head design. Nevertheless, the topology optimization analysis was used for further improvement in the stiffness to mass ratio. A final version was designed incorporating the assembly aspects of the spindle head. A significant improvement with 60% (12µm to 5µm) in static displacement, 80% (149Hz to 267Hz) in the first modal frequency and 90% (95µm to 12µm) in the dynamic displacement was observed in the modified version of the spindle head structure as compared to the existing one.

The author aims to develop a comprehensive design guideline for the machine tool designers and manufacturers incorporating the experimental and FEA aspects towards enhancing the dynamic performance of the existing machine tools. It can also be used for predicting the dynamics of new products and thereby designing the new machines from scratch as the FEA results matches well with the experimental tool-tip FRF. Furthermore, the introduction of predictive stability lobe diagram in the present design methodology has been planned. This ultimately will help designers look ahead into the performance of the machine tool even at its design stage.

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