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Development of a vibration free machine structure for high-speed micro-milling center

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Abstract The demand of ultra-precision micro-machine tools is growing day by day due to exigent requirements of miniaturized components. High accuracy, good dimensional precision and smooth surface finish are the major characteristics of these ultra-precision machine tools. High-speed machining has been adopted to increase the productivity using high-speed spindles. However, machine tool vibration is a major issue in high-speed machining. Vibration significantly deteriorates the quality of micro-machining in terms of dimensional precision and surface finish. This paper describes a design methodology of a closed type machine structure for vibration minimization of a high-speed micro-milling center. The rigid machine structure has provided plenty of stiffness and the damping capability to the machine tool without utilizing vibration absorbers. The models of the machine structures have been generated and assembled in AutoCAD 3D. The performances of the integrated micro-milling machine tools were determined by finite element analysis. The best model has been selected and proposed for manufacturing. Additionally, simulation results were validated by comparing with experimental results. Eventually, after manufacturing and assembly, experiments have been performed and determined that the amplitude of vibration was approaching towards nanometer level throughout the working range of the high-speed spindle. The machine tool was capable to fabricate miniaturized components with fine surface finish.

Keywords High-speed micro-milling · Machine structure · Modal analysis · Frequency response analysis · Dynamic stability

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1 Introduction

Micro-machining technologies have versatile applications in several industries like electro-optics, automotive, biotechnology, aerospace, information technology etc to fabricate high accuracy miniaturized components. The growing demand of micro-machining technology has facilitated the requirement of high-performance and efficient ultra-precision machines tools [1]. Highly precise complex 3D shapes with mirror finish on different materials can be fabricated in those ultra-precision machine tools in a expeditious and cost-effective way [2,3]. Micro-milling is an adaptable technology for generating miniaturized components with complex geometric features and mirror finish on difficult-to-machine materials [4,5], even on silicon [6]. The major limitation of the micro machining is low tool stiffness and low MRR. High speed micro-machining has been adopted to improve MRR and reduce the chip load. The mechanism of chip formation and the grade of surface finish have made high-speed micro-milling a distinctive one from other traditional material removal processes [7]. The static and dynamic performances of the high-speed micro-machine tools have directly influenced the machining performances. However, determination of the dynamic performances of those high-speed machine tools is a challenging issue by creating real prototype [8]. Finite element analysis has been adopted as a comprehensive mean for that issue [9].

High-speed micro-milling machine tools have required high speed spindles to maintain the machining speed and efficiency. But at high spindle speeds, acute vibration takes place in the machine tools [10]. This jeopardizes the dimensional accuracy; precision and surface finish of micro size products. Apart from this, chatter may be developed due to vibration and that can lead to tool wear and breakage of the cutting tool [11]. Chatter may also affect the spindle and the machine tool [12]. Therefore, vibration isolation is necessary to maintain the accuracy and surface finish of the machined surface and also to maintain the tool life. The stiffness of the machine structure plays a very crucial role for machining efficiency and vibration isolation of a machine tool [13]. Hence, the high-speed micro-milling machine structure must have good static, dynamic stiffness and damping performance for quality machining performance [1].

Several approaches have been reported for development of the ultra-precision machine tools since 1980's [14]. However, micro machining has been incorporated with ultra-precision precision machining since last two decades. Luo et al. [15] developed a bench-top UPM machine tool which was capable to machine miniaturized components with high accuracy. The accuracy of the spindle and stages including the damping quality of the machine structure has significant influence on the machining accuracy in that machine tool. Huo et al. [9,16] proposed a design approach of high-speed ultra-precision micro-milling machine tool. They concluded that closed type machine structure has shown higher stiffness and damping capability as compared to open type machine structure. However, medium to high frequency vibration still exists within the range of high-speed spindle. Park et al. [17] proposed a design methodology of a meso-scale machine tool based on analytical and finite element modeling. They focused on optimization of the structure in order to achieve higher stiffness. They have found that the vibration was increased and the stiffness was reduced with the increase in height of the column. The stiffness of the machine structure has been considered as a crucial factor influencing the product quality in terms of dimensional precision and surface finish [13]. Liang et al [18] designed an ultra-precision diamond fly-cutting machine tool. For improvement in accuracy of that ultra-precision machine, they focused on minimizing the size of machine components to reduce the cantilever action and thus, decrement in deformation due to vibration. They also suggested closed type structure for better rigidity. Therefore, the most considerable factor during development of a high-speed precision machine tool is minimization of machine tool vibration which creates a major area for future research.

For structural vibration minimization, Yang et al. [19] developed a vibration isolator mechanism based on quasi zero stiffness which significantly reduced the vibration transmissibility. Therefore, it has worked as an effective mean of low frequency vibration. Additionally, Zhang et al. [20] applied an active vibration control method using nonlinear vibration absorber which resulted in lower resonant amplitude of vibration. Semm et al. [21] incorporated substructure coupling approach considering local and global damping to improve the accuracy of FEM simulation in order to determine damping performance of a machine tool. Zhang et al. [22] proposed a discrete time-delay chatter control method with closed loop chatter stability model for milling process. It stabilized the machining parameters and reduced the amplitude of chatter vibration by 78.6%. Orra and Chaudhury [23] presented a electro-magneto-rheological damping system with a closed loop feedback control system to suppress the machine tool vibration during turning process. The damper has been attached under the tool holder and generated counter force to suppress tool vibration when excited by current signal. Representing the contacts in machine tool is a challenging issue which is required for accurate prediction of machine tool behavior. For ease of computational issues, the contact forces have been directly mapped onto the FE model of machine tool topological optimization problem [24]. Mohammadi and Ahmadi [25] proposed a single degree of freedom model with nonlinear restoring force to determine the vibration response of a KUKA machining robot at tool center point. However, it cannot accurately predict the system dynamics. The design of the machine structure is significant for vibration isolation of the precision micro-machine tool. The machine structure contributes to the machine tool dynamics. The goal of the research was to reduce the vibration of the machine structure so that number of resonances with the working frequency of the machine tool has been reduced. The perfect design of the machine structure possessed very low amplitude of vibration under the resonance condition within the working range.

The current study reveals the development strategy of a gantry type machine structure for a high-speed micro-milling center to minimize the machine tool vibration. The material of the machine structure has been determined

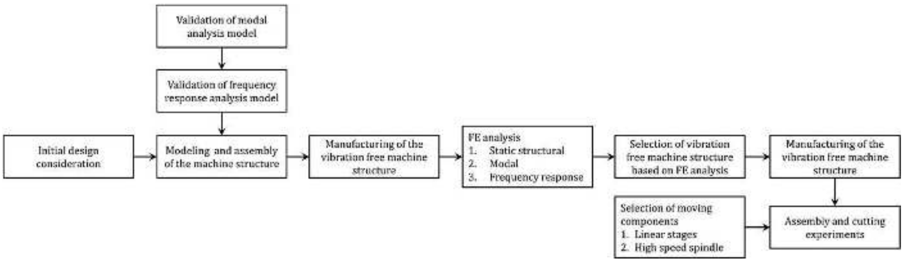


Fig. 1 Flow chart of the design methodology

based on their specific stiffness, damping property, thermal stability, and ease to fabrication. All 3D components of the machine structure have been generated in AutoCAD 3D. Finite element analyses have been performed on the assembled machine tool models in ANSYS 15.0 to evaluate their static, dynamic stiffness and damping performance. The fundamental objective was to design the machine structure so that it created less number of resonances with the working frequencies of the spindle and the amplitude of vibration during the high-speed micro-milling operation was very low. Based on the FEM results, the best model of machine structure has been selected and proposed for manufacturing. Eventually, amplitude of vibration during micro-milling operation has been determined experimentally to check the similarity with the FEM results.

2 Model validation

Figure 1 shows the flow chart of the development strategy. The design methodology is based on FEM analysis, especially modal analysis and frequency response analysis. Therefore, it has been focused primarily to validate the models before starting the design approach.

For validation, a bridge type structure has been fabricated and its CAD model has been replicated in AutoCAD 3D. Modal analysis simulation has been performed on that model in ANSYS and the results were compared with results of experimental modal analysis. The experimental modal analysis has been performed by exciting the bridge structure using an impact hammer and the signals have been collected by roving accelerometer technique. The accelerometers of Bruel & Kjaer have been used for this purpose. The FRF data was generated by photon+ dynamic signal analyzer and processed in MEScope Ves software to determine the natural frequencies experimentally. The results of experimental modal analysis of the bridge structure have shown a good similarity with the simulation result of modal analysis in ANSYS workbench with a maximum of 11% deviation which was found in the third mode. Therefore, the model was further expanded to find out the vibration free machine structure. Figure 2 represents the fabricated bridge structure and the compar-

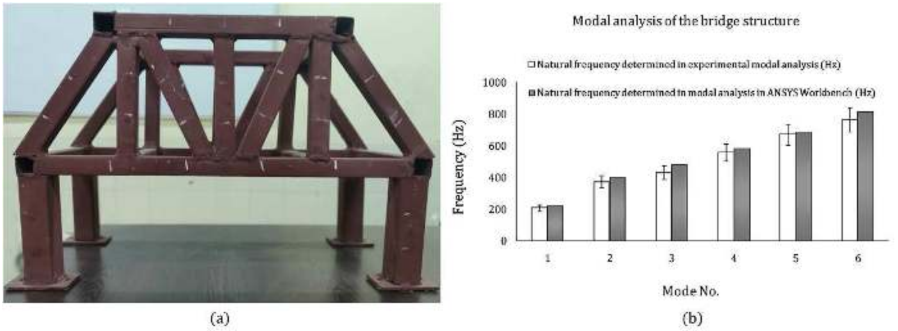


Fig. 2 (a) The bridge structure used for validation of modal analysis model in ANSYS workbench, (b) the comparison of natural frequencies of the bridge structure found in ANSYS modal analysis and experimental modal analysis

ative study of natural frequencies found in ANSYS and experimental modal analysis.

Initially, after validation of modal analysis, a model of machine structure was developed in AutoCAD for the micro-milling machine and the natural frequencies of the individual structure were determined by modal analysis in ANSYS. The material chosen for the structure was mild steel due to its easy availability. The natural frequency of 1st mode of the individual mild steel frame was 143.38 Hz, found in ANSYS modal analysis. After assembling the linear stages, spindle and spindle holding fixture with the structure, static structural analysis, modal analysis and harmonic response analysis were performed on the CAD model of the assembled machine tool. The static structural analysis over the assembled model of mild steel frame has determined maximum 7 micron deformation under constant static load. However, the natural frequency of the 1st mode of the assembled frame was 59.83 Hz found in ANSYS modal analysis. The reduction in natural frequency of the assembled frame than the individual one is due to the increase in mass because of the attachment of stages, spindle and spindle holder. The harmonic response analysis over the assembled mild steel frame was performed applying average constant cutting force on the tip of the spindle. The range of frequency was given 150 Hz to 1000 Hz according to spindle working frequency. It has determined maximum amplitude of 5.03 micron along Y axis at a frequency of 235 Hz. The maximum amplitude of vibration along Z axis was 2.59 micron at a frequency of 490 Hz.

Subsequently, the structure of mild steel was manufactured for the micro-milling machine and fixed over a honeycomb table as shown in Fig. 3 (a). Experimental modal analysis was performed over the unassembled machine structure using impact hammer and accelerometers in order to check the similarity with the simulation result. The results of experimental modal analysis have also shown good similarity with the simulation result. Maximum 17% deviation was found in first mode as shown in Fig. 3 (b). The linear stages and the spindle along with the spindle holding fixture were assembled and attached to the structure and the honeycomb table. An experimental frequency response analysis was performed to check the amplitude of vibration of that mild steel machine structure using accelerometers under machining condition at constant chip load and varying rotational velocity. The depth of cut was constant in every case and feed rate was being varied with rotational speed to keep chip load as constant. The cutter used was a two flute milling cutter. The FRF generated by photon+ dynamic signal analyzer was taken and processed in MEScope Ves software to determine the amplitudes of vibration at resonance condition. The results of experimental frequency response analysis were compared with the results of harmonic response analysis simulation and the similarity was determined. The experimental frequency response analysis of the steel structure at machining condition shows that maximum deformation along Y direction is 5.7 micron in resonance with frequency 265 Hz. However, along Z direction, the maximum deformation is 2.7 micron in resonance with frequency 484 Hz. The plot is almost similar with the plot achieved in harmonic response analysis in ANSYS. A continuous plot can be achieved in ANSYS; however, this is impractical to achieve such continuous plot in experimental frequency response analysis. Therefore, slight variation occurs in these plots. Figure 4 represents the comparative results of frequency response determined in experiment and FEM analysis. Hence the model was validated. It was observed that the amplitude of vibration reduced with increase in rotational speed while machining. This is due to lower force transmission to the structure at higher rotational speed. Above 1000 Hz, no peak was found as the maximum working frequency of the spindle was 1000 Hz. The deformation was measured in these two directions because the maximum deformation was found in Y and Z directions in the mode shapes in modal analysis and the deformation along X direction was negligible.

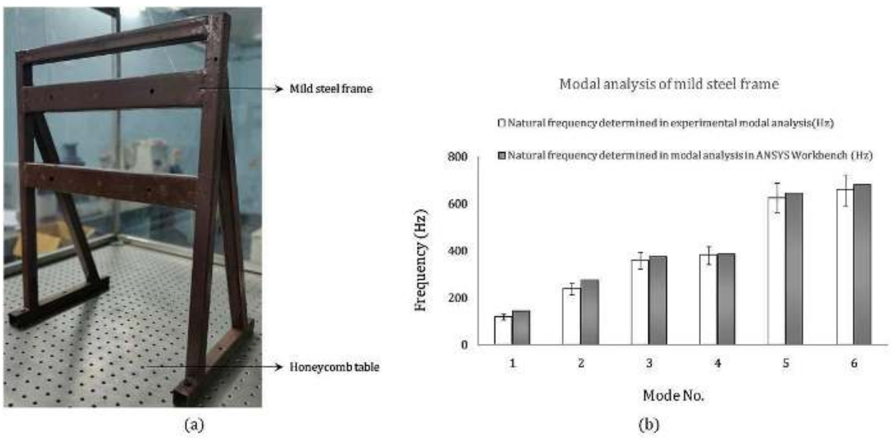
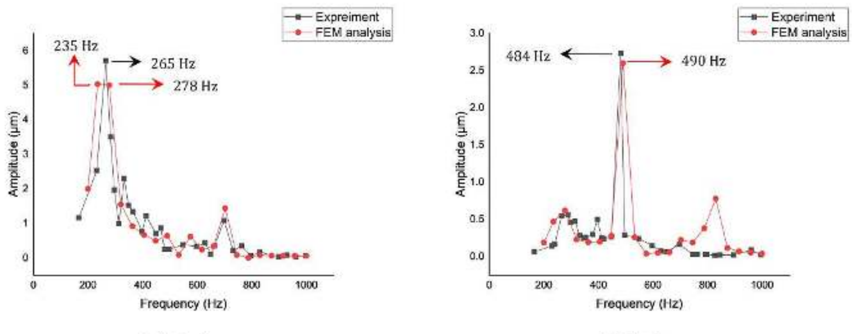


Fig. 3 (a) The mild steel frame for the micro-milling set up, (b) the comparison of natural frequencies of the mild steel frame determined in ANSYS modal analysis and experiments



(a) Y axis (b) Z axis

Fig. 4 Comparison of the results of frequency response analysis on the mild steel frame determined in experiments and FEM analysis

3 Preparation of the model and FEM analysis

3.1 Preparation of the model

The design approach is a trial and error approach that incorporated 3D modeling of the structure components, their assemblies and finally evaluation of their performances by FEM analyses. Based on the results of the analyses, the change of materials, structural configuration and the dimensional optimization have been performed to improve the performances in terms of static, dynamic stiffness and damping capability. Eventually, the best model has been selected as the vibration free structure and proposed for manufacturing. Most of the previous researches have focused on external vibration absorbers in mechanical structures to improve the damping capability [26,27]. However, this research has been focused on the internal damping capability of the structure to reduce the machine tool vibration without utilizing any external vibration absorber. The design of the machine structure of the high speed micro-milling machine has been made closed type as open type structure are more assailable to vibration [9].

3.1.1 Materials selection

The initial design criterion involved selection of the material for the machine structure. A material having higher damping capacity, good specific stiffness, and small thermal expansion co-efficient with low specific heat capacity may be considered as good structural material. Additionally, the manufacturability of materials has been considered during design. Therefore, the material selection is the primitive factor in machine tool design. Generally, metallic structures possesses low damping performance due to lower damping capacity [28]. Cast iron has been used for conventional machine structure for higher damping characteristics and load carrying capacity [29]. However, elevated processing cost and poor environmental properties have limited cast iron for

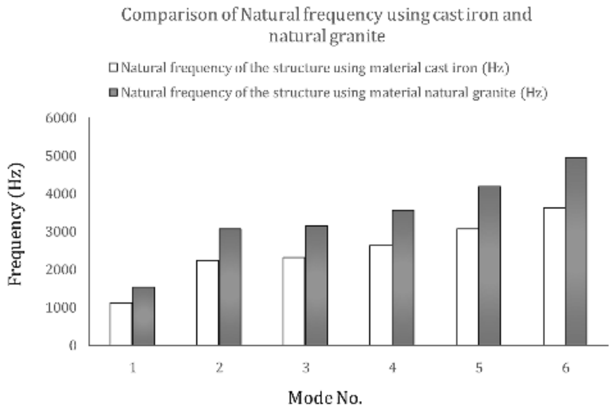


Fig. 5 The comparison of natural frequencies of the proposed machine structure using cast iron and natural granite

precision machine tool structure [30]. Natural granite has shown better performance than cast iron in terms of rigidity, damping quality, thermal stability and dynamic load carrying capacity [31,32]. Besides, good form stability and flatness have made natural granite superior for high-speed ultra-precision machine tool structure [33]. For all of those characteristics, both cast iron and natural granite were selected for the machine structure. The performances were evaluated for a single CAD model using both the materials in ANSYS modal analysis. The minimum natural frequency of the structure was 1118.8 Hz when the material was cast iron. However, the minimum natural frequency was 1528.9 Hz using natural granite as the material of the machine structure. Also natural granite is capable for higher vibration damping as compared to cast iron. Hence, for better performance, natural granite was selected as the material of the invented machine structure. Figure 5 shows the comparison of natural frequencies of the structure using both cast iron and natural granite.

3.1.2 Structural configuration

The parts of the high-speed micro-milling machine were divided into two categories. The machine structure is the stationary part and the other parts such as spindle, linear stages are moving parts. Before starting the design of the machine structure, the moving parts were selected. According to the bearing stiffness, accuracy, damping capacity and thermal performance, a spindle of model AG62-60-0.24-P-DS, manufactured by Fischer spindle was selected. The minimum rotational speed of the spindle is 10000 rpm and maximum rotational speed is 60000 rpm. Therefore corresponding minimum and maximum working frequencies of the spindle are 166.6 Hz and 1000 Hz respectively. The specifications of the spindle have been described in Tab. 1. The spindle holding fixture was made of mild steel. Further linear stages were selected according to their resolution, accuracy and repeatability. The linear stages of model LMS-

200-200 were selected for the high-speed micro-milling center. The maximum travel ranges of the stages are 150 mm and maximum base area is 200 mm × 250 mm. The specifications of the linear stages have been represented in Tab. 2. After the selection of the moving parts, the machine structure was designed for providing the accommodation of these parts and vibration isolation. The whole design approach incorporated a number of 3D models among which the best one has been selected for manufacturing based on static and dynamic stability. This description only involved the best model which has been considered as vibration free.

Table 1 Technical specifications of the high-speed spindle

Spindle speed (RPM)	Motor type	Torque (N.m)	Motor power (kW)	Cooling system	Lubrication system	Mass (kg)	Tool change
10000 - 60000	Asynchronous	0.24	1.5	Liquid	Grease	3.3	Pneumatic

Table 2 Technical specifications of the motorized linear stages

Stages	Travel range (mm)	Resolution (nm)	Accuracy (μm)	Maximum travel speed (mm/min)	Minimum travel speed (mm/min)	Load carrying capacity (kg)
X stage	150	312	±2.5	240	0.01	50
Y stage	150	312	±2.5	240	0.01	20
Z stage	150	312	±2.5	240	0.01	50

Machine bed accommodates the overall machine tool. So the stiffness and the load carrying capacity of the machine bed must be high enough. For the machine structure, the length of the machine bed is 660 mm and the width is 556 mm. The maximum height of the machine bed is 100 mm. Two slots of 50 mm depth were given at both sides of the bed. The frame columns were attached into the slots of the bed from bottom as well as from transverse side so that the transverse movement of the frame column due to vibration could be restricted. The X-Y linear stages were fixed at the center of the machine bed from the transverse side. The total distance required in X or Y direction for the travel of the X-Y linear stages is 350 mm (200 mm stage width + 150 mm travel). Therefore, the distance provided between two slots is 354 mm so that the X-Y linear stages can travel without getting obstructed between the frame columns. A through hole of 40 mm diameter were provided at the machine bed so that the wires connected to the control systems of the spindle and linear stages can be passed through that hole.

The design of the frame column was generated as per the vibration point of view and the manufacturability of granite. The thickness of the frame column is 100 mm and its height is 275 mm. The heights of the frame columns were optimized in order to minimize the natural vibration. The thickness was adjusted to provide sufficient stiffness. The frame columns are carrying the load of the upper block. The upper block is a granite block designed for the closed type machine structure. It carries the vertical linear stage along with spindle holding fixture and spindle. The thickness of the upper block is 100 mm and the maximum height is 207.5 mm. The height of the upper block has been adjusted to minimize the natural vibration of the overall machine structure. After generating each part of the machine structure they were integrated. Figure 6 represents the CAD models of all components of the vibration free high-speed micro-milling machine tool. Figure 7 (a) represents the unassembled vibration free machine structure which has been integrated with spindle and linear stages as shown in Fig. 7 (b).

3.2 Static structural analysis

Static structural analysis has calculated the influence of static loading on the machine structure neglecting the effects of vibration. The assembled CAD model of that machine structure has undergone static structural analysis in ANSYS considering the materials of each part and applying constant static load in each X, Y and Z direction at the tip of the tool holder of the spindle. The bottom of the machine bed was remained fixed. Total deformation and von-mises stress have been selected as output parameters. The maximum deformation of 2.85 micron and negligible stress were found in static structural analysis. The deformation variation in the overall machine tool is shown in Fig. 8(a). The variation of von-mises stresses in the high-speed machine tool is depicted in Fig. 8(b). The maximum von-mises stress was 27.5 MPa found in the structure. The effect of the bearings in the spindle was not considered in this analysis. Therefore, the maximum deformation as well as stress was found at the collet of the spindle. It can be seen that the proposed model of machine structure has provided superior rigidity to the assembled high-speed micro-milling machine tool. The maximum deflections in the frames were approaching below 10 µm for almost all designs. Therefore, it cannot be determined the best design based on the performance of only static structural analysis.

3.3 Modal analysis

After static structural analysis, the natural frequencies of the assembled machine tool models have been determined by modal analysis in ANSYS workbench. This analysis uses Block Lanczos mode extraction method to determine the natural frequencies. The structural configurations which created less number of resonances with the working frequency of the spindle were selected. The

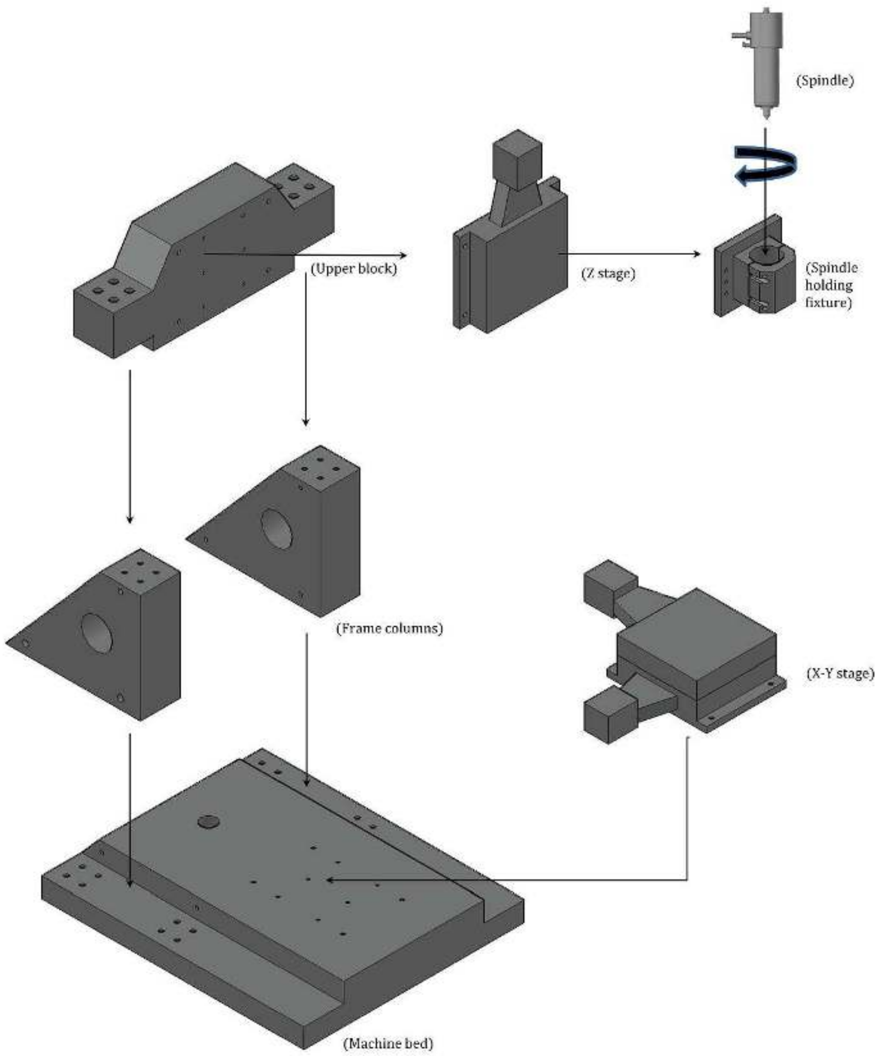


Fig. 6 The components of the vibration free high-speed micro-milling center prepared in AutoCAD

purpose of the modal analysis was to find out the corresponding rotational speeds for resonance. Maximum rotational speed of 60000 rpm has been applied at the spindle collet. The spindle frequency can be depicted from the equation f=N/60, where N and f represents the rotational speed and corresponding working frequency of the spindle respectively. Therefore, the maximum working frequency of the spindle was 1000 Hz. The natural frequency of 1st mode of the assembled model was 718.79 Hz determined in modal analysis. It was found that only four natural frequencies are there below or adjacent to

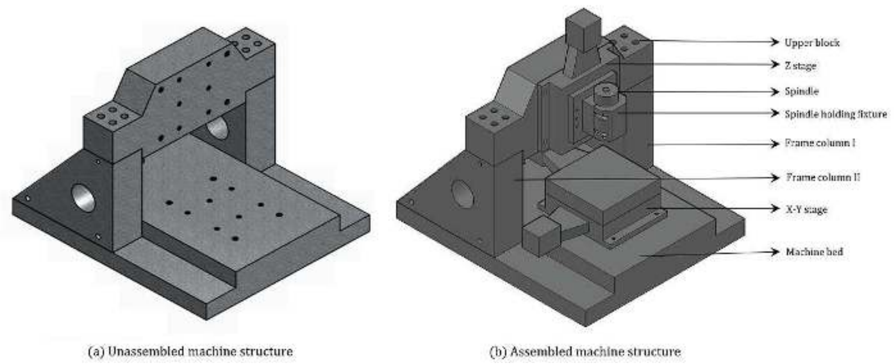


Fig. 7 CAD model of the vibration free machine structure

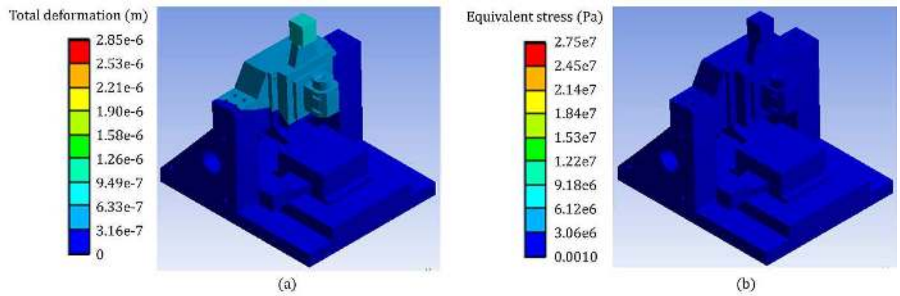


Fig. 8 The results of static structural analysis of the proposed model determined in ANSYS workbench. (a) Total deformation, and (b) Equivalent stress

1000 Hz as shown in Fig. 9. However, in the 2nd and 3rd natural frequencies, deformations were found only in the servo motors of linear stages according to the mode shapes. Therefore, only two natural frequencies are there where the structure may create resonance with the working frequency of the spindle. The first four mode shapes have been represented in Fig. 10.

The materials selections, structural configuration, and dimensional change have taken a major part to determine the mode shapes, natural frequencies, and deformations in different mode shapes under resonance. A 36% improvement has been observed for a single model, when the material has changed from cast iron to natural granite as shown in Fig. 5. Therefore, the number of resonances has been reduced when the structure material was natural granite. Similarly, a small increase in height of the upper block resulted in significant deformation as shown in the mode shapes, represented in Fig. 11. All of the issues have been taken into consideration during the design of the machine structure.

The proposed model has shown better result compared to other models considering the dimensions and spaces for movement of the stages. It was not much excited with the working frequency of the spindle in terms of creating resonances. However, the resonant frequencies have changed under the action

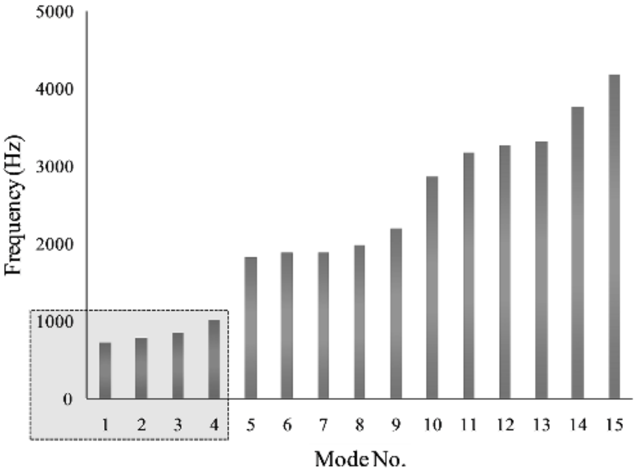


Fig. 9 Plots of first 15 natural frequencies of the proposed assembled machine structure determined in ANSYS modal analysis

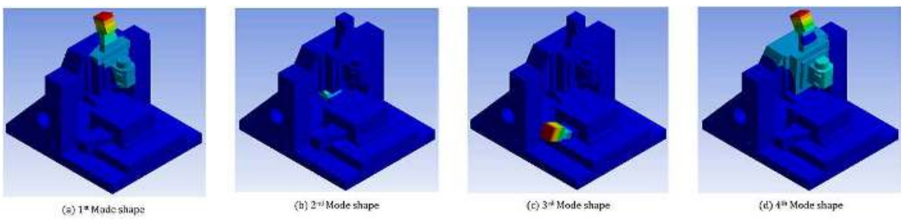


Fig. 10 First four mode shapes of the assembled machine structure, determined in ANSYS modal analysis

of forced vibration occurring due to the action of cutting force. Therefore, the effect of machining operation on the machine structure cannot be considered in modal analysis.

3.4 Harmonic response analysis

Harmonic response analysis was performed on the assembled models of the machine tools to check the amplitude of vibration under forced vibration. In this analysis, the machine structure has been excited by a series of harmonic cutting forces F, acting between the milling cutter and the workpiece. The harmonic force F can be expressed as, $[F = F_0 sin(\omega t)]$; where ω is the given frequency of the cutting operation [9]. The tip of the tool holder has been given an average cutting force F_0. The range of frequency was given according to the working cutting frequency i.e. 166 Hz to 1000 Hz. The model which showed lowest amplitude of vibration under the action of cutting force has been finalized as the optimized structure. Then that machine structure was proposed for man-

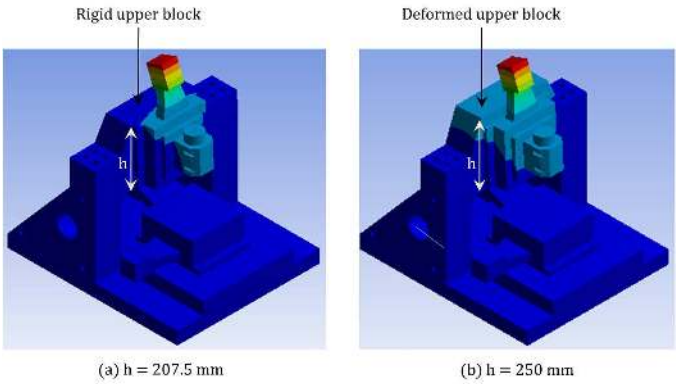


Fig. 11 First mode shapes of the assembled machine structure with different heights of the upper block

ufacturing. It has been observed that the structures made of natural granite possessed higher structural rigidity as compared to other structural materials like cast iron. The amplitude of vibration for the proposed machine structure was approaching towards nanometer level. The maximum amplitude was 0.567 micron along Y axis and the peak was found in 235 Hz. However, maximum amplitude along Z axis was 0.108 micron and corresponding frequency was 235 Hz. The frequency response plots of the assembled machine structure along Y and Z axes have been represented in Fig. 12. The deformation was measured along these two directions because the maximum deformation was found in Y and Z directions in the mode shapes found in modal analysis and the deformation along X direction was negligible. The analysis result has shown that the amplitude of vibration has been reduced by approximately 10 times for the proposed machine structure as compared to initial mild steel structure. Hence, the structure has provided good rigidity and damping performance to the micro-milling machine tool. It absorbed a huge extent of noise generated by vibration during the machining operation and resisted the displacement of the machine tool. Therefore, the structural damping contributed considerably to reduce the vibration of high-speed micro-milling center.

After evaluating the performance in ANSYS, the model was proposed for manufacturing. The maximum floor area required for the machine structure was 660 mm $\times$ 556 mm. The maximum height of the structure was 500 mm. The weight of the complete structure was 200 kg. The attachments with fasteners take a major part for structural damping as slightly loose fastening may results in enormous vibration in the machine structure. It was taken into consideration during manufacturing.

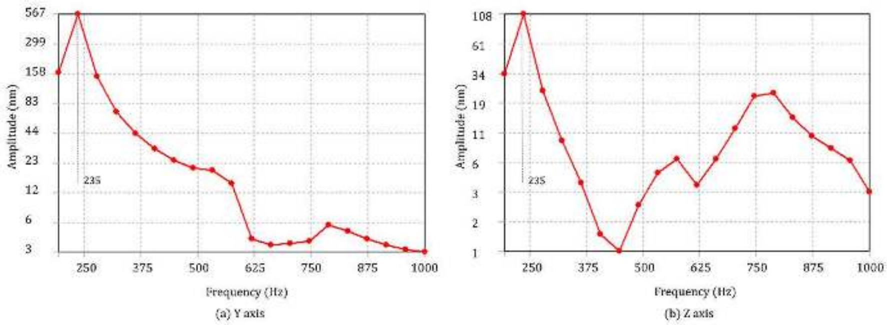


Fig. 12 Plots of harmonic response analysis of the assembled machine tool model determined in ANSYS

4 Experimental performance of the machine structure

4.1 Experimental setup

After manufacturing, the machine structure has been assembled with the spindle and linear stages. Figure 13 represents the assembled high-speed micro-milling center. Experimental frequency response analysis has been performed in the assembled machine tool during machining to check the amplitude of vibration. The machining operations have been performed under dry condition with constant chip load (Chip load= $\frac{f}{NZ}$; where f is feed rate per min, N is rpm, and Z is the number of flutes). The chip load was same as the previous experiment performed in mild steel frame. The range of rotational speed has been varied from 10000 to 60000 rpm with constant intervals. The feed rates have been varied accordingly to maintain constant chip load. Depth of cut was constant for each run. The variations of all machining parameters and the workpiece material were same as the previous experiment in mild steel frame. The vibration data have been collected by accelerometers using Bruel and Kjaer 4 channel dynamic signal analyzer. The recorded signals were processed in MeScope Ves and the amplitudes of vibration were determined in this software. The experiment has been performed with a two flute milling cutter. Furthermore, the machining experiment has been repeated with a four flute milling cutter with same process parameters. These results were compared with the results of harmonic response analysis performed in ANSYS in order to determine the similarity.

4.2 Experimental frequency response analysis

The amplitude of vibration of the machine structure was approaching towards nanometer level during machining with two flute milling cutter. The FRF plots along Y and Z axes have been shown in Fig. 14. It shows that the maximum deformation of the structure was 0.47 micron in Y direction while cutting the

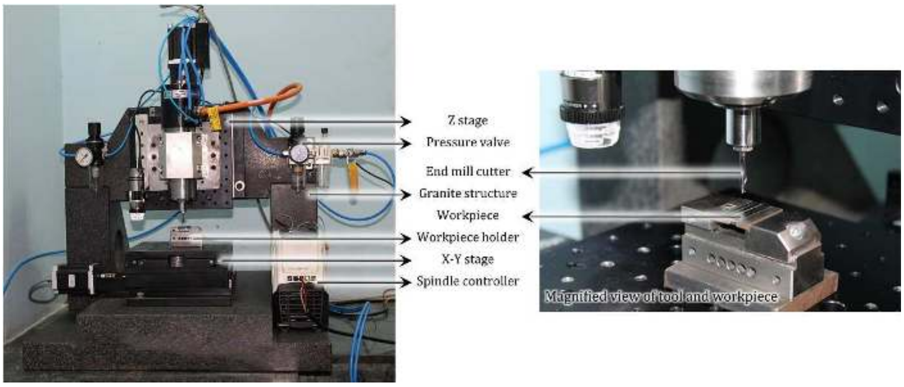


Fig. 13 Assembled high-speed micro-milling machine tool with the proposed machine structure made of natural granite

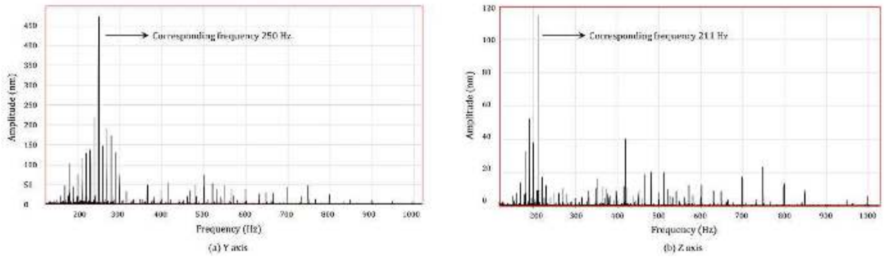


Fig. 14 The results of experimental frequency response analysis of the developed high-speed micro-milling center while machining with two flute milling cutter

same material as previous in mild steel structure. The corresponding frequency at which the maximum deformation occurred was 250 Hz. The maximum deformation of the structure in the Z direction was 0.112 micron while the machining operation and the corresponding frequency at resonance was 211 Hz. The plot was almost similar to that one found in the harmonic response analysis of that model in ANSYS. Slight variation was due to discontinuous plot in experimental frequency response analysis and also due to the presence of higher order frequency. It was found that the resonance zone occurred in between 190 Hz and 270 Hz. However amplitude is more dominant at 250 Hz along Y axis and 211 Hz along Z axis. The amplitude of vibration has been reduced with increase in rotational speed because lower force transmission at higher rotational speed. Hence, it was observed that the deformation has been reduced 12 times as compared to the previous mild steel structure.

The machining operation performed by four flute milling cutter precipitated that maximum deformation along Y direction was 0.38 micron and corresponding resonant frequency was 250 Hz. However, the maximum deformation along Z axis was 0.091 micron and corresponding resonant frequency was 211 Hz. The frequency response plots along Y and Z axes for this experiment have been depicted in Fig. 15. This experiment has shown good similarity

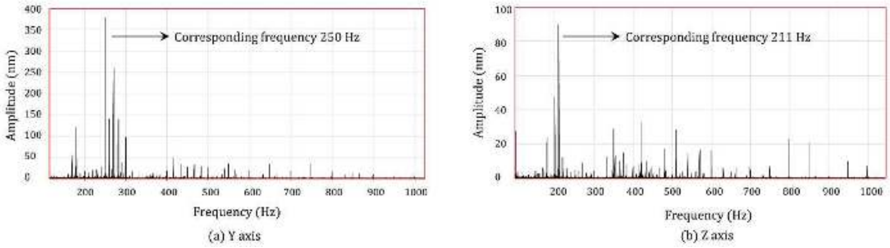


Fig. 15 The results of experimental frequency response analysis of the developed high-speed micro-milling center while machining with four flute milling cutter

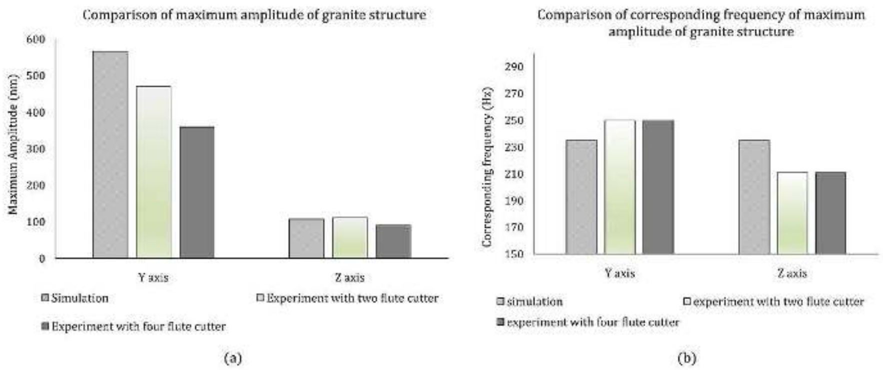


Fig. 16 (a) Comparison of maximum amplitude; and (b) comparison corresponding frequencies of the developed micro-milling center, determined in ANSYS harmonic response analysis, experimental frequency response analysis with two flute cutter and experimental frequency response analysis with four flute cutter

with the resonant frequencies achieved in previous experimental result as well as in the result of ANSYS. The plot was almost similar with the plot achieved in simulation. Probable occurrence of second order frequency (higher order) has led to a peak at 422 Hz along Z axis. The variation of amplitude was due to the variation of cutting force in both experiments. However, a constant average cutting force was considered in harmonic response analysis in ANSYS. The comparisons of maximum amplitudes and corresponding frequencies determined in both the experiments and the FEM analysis have shown in Fig. 16.

The deformation found was almost negligible, therefore, it can be concluded that the structure is vibration free upto the maximum rotational speed of the working spindle. The amplitude has a tendency to be reduced with operating frequency. Therefore, the machine structure can be operated for ultra-high speed machining as well. The machining vibration has been reduced to great extent using the developed machine structure. During micro-milling operation, an average surface finish of 100 nm has been achieved on difficult-to-cut materials. The machining was performed upto the maximum rotational speed

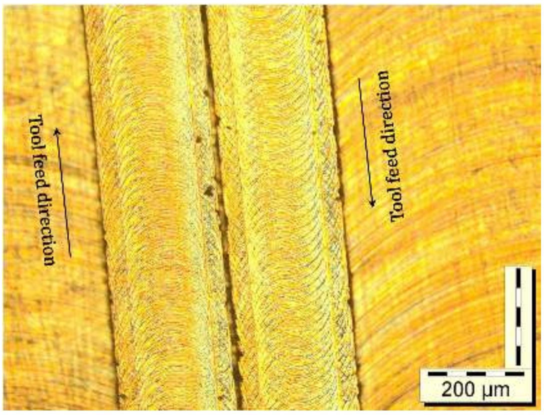


Fig. 17 Tool feed marks on micro grooves of 200 µm width and 60 µm depth, generated on high strength super alloy Ti6Al4V by high-speed micro-milling

of the spindle without breakage of the cutting tool. Hence the accuracy, precision, surface finish of micro-machining as well as the tool life of the micro cutting tool was maintained for that micro-milling center. Figure 17 depicted the movement of the milling cutter on the workpiece surface during micro-milling operation. It can be seen that the tool movement was uniform along the infeed direction and any transverse movement due to vibration was not found.

5 Conclusions

In this study, a design approach of vibration free machine structure has been presented for a high-speed micro-milling machine tool. This design methodology has incorporated the modeling and assembly of the machine structure including material selection. Subsequently, FEM analyses i.e. static structural analysis, modal analysis and harmonic response analysis have been performed to evaluate the static and dynamic performances of the assembled machine structure. Based on the results of FEM analyses, the best model have been selected as vibration free and proposed for manufacturing. Eventually, experimental frequency response analyses have been performed during machining to check the similarity with FEM results. The experimental results have shown well similarities with the FEM results in terms of resonant frequencies, amplitude of vibration and natural frequencies. During high-speed micro-milling operation, an average surface roughness of 100 nm with uniform tool feed marks have been observed on the machined surface of difficult-to-cut materials. Based on this study, the following conclusions can be made:

1. The design approach has been focused on developing a rigid vibration free machine structure. The developed machine structure has provided good static, dynamic stiffness and damping performance to the high-speed micro-milling machine tool. The amplitude of vibration has been reduced by 12

times with this structure as compared to previous mild steel structure. Therefore, the machine tool vibration has been reduced significantly utilizing this machine structure. The design approach is appropriate for vibration isolation without utilizing any vibration absorber.

2. Natural granite has shown better stiffness, rigidity and damping performance as compared to cast iron. Therefore, the machine structure made of natural granite has shown better performance than structure made of cast iron to reduce structural vibration.
3. The amplitude of vibration has been reduced with increasing the cutting speed (rotational speed). This is due to lower force transmission at higher cutting speed. Hence, the developed machine structure can be utilized for ultra-high speed machining as well.
4. The amplitude of vibration in the machine tool has been reduced during machining with a four flute milling cutter as compared to two flute milling cutter under similar machining condition. Machining with four flute milling cutter precipitated lower chip load resulted in lower cutting force. Therefore, force transmission to the machine tool has been reduced utilizing four flute milling cutter. Thus machine tool vibration has been reduced.
5. The design of the machine structure has been optimized to minimize the vibration. The thickness, height and width of all components of the structure have been determined considering vibration point of view; and to reduce cantilever effect, and to improve structural stiffness. The next step is to develop an ultra-high speed micro-milling center incorporating this developed machine structure which can be a part of the future research.

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Declarations

Funding

Not applicable

Conflict of interest

The authors declare no conflict of interest.

Availability of data and material

All data are accurate and generated from official sources.

Code availability

Not applicable

Authors' contributions

Vivek Bajpai has planned and defined the methodology of the research work and arranged necessary funding. Arnab Das has performed the simulations. Shashank Shukla has participated in the purchasing work and performed the assembly of the machine tool. Arnab Das, Mohan Kumar and Chitransh Singh have performed the experiments and analysis. Madan Lal Chandravanshi have provided the necessary instruments for experiments and prepared experimental planning. Arnab Das and Vivek bajpai have participated in paper writing and necessary revisions of it. All authors read and approved the final manuscript.

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Figures

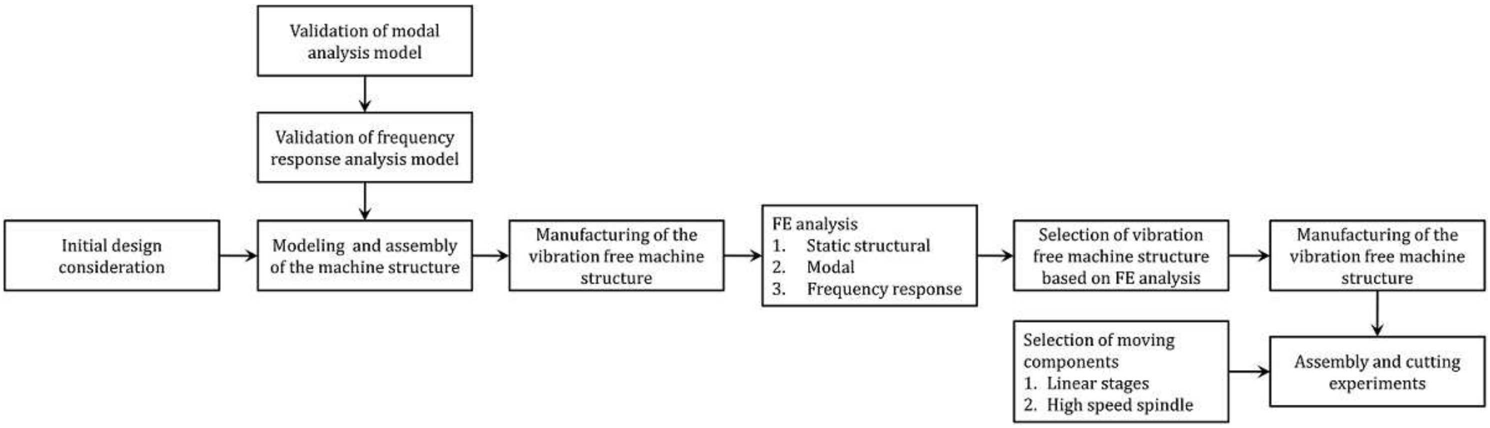
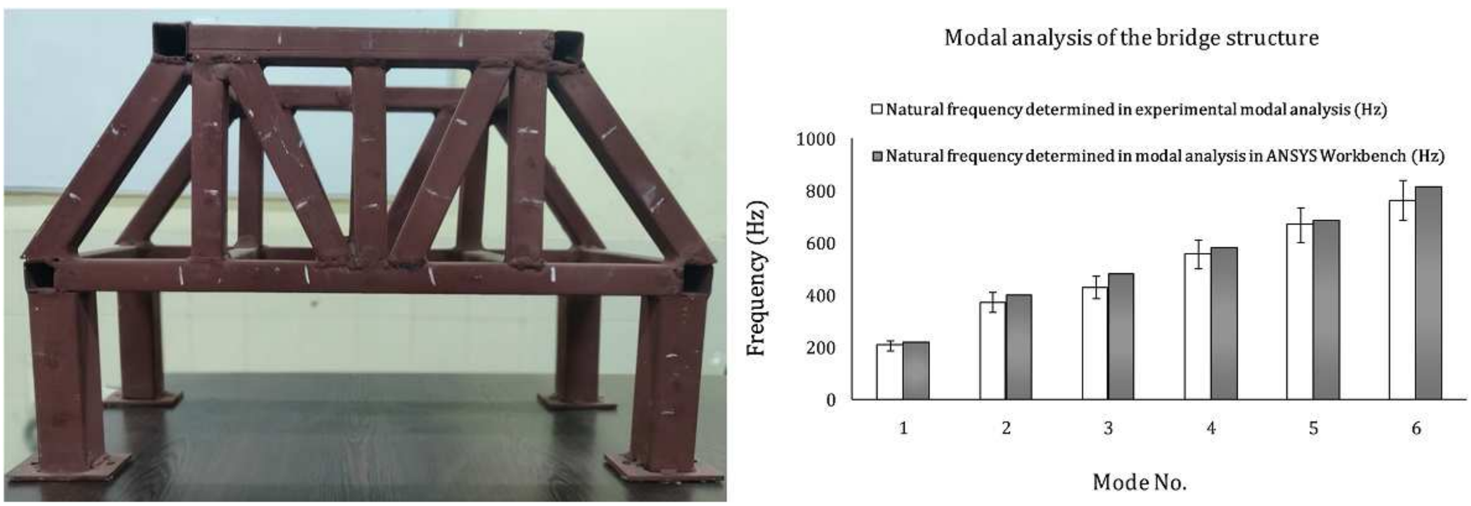


Figure 1

Flow chart of the design methodology



(a) (b)

Figure 2

(a) The bridge structure used for validation of modal analysis model in ANSYS workbench, (b) the comparison of natural frequencies of the bridge structure found in ANSYS modal analysis and experimental modal analysis

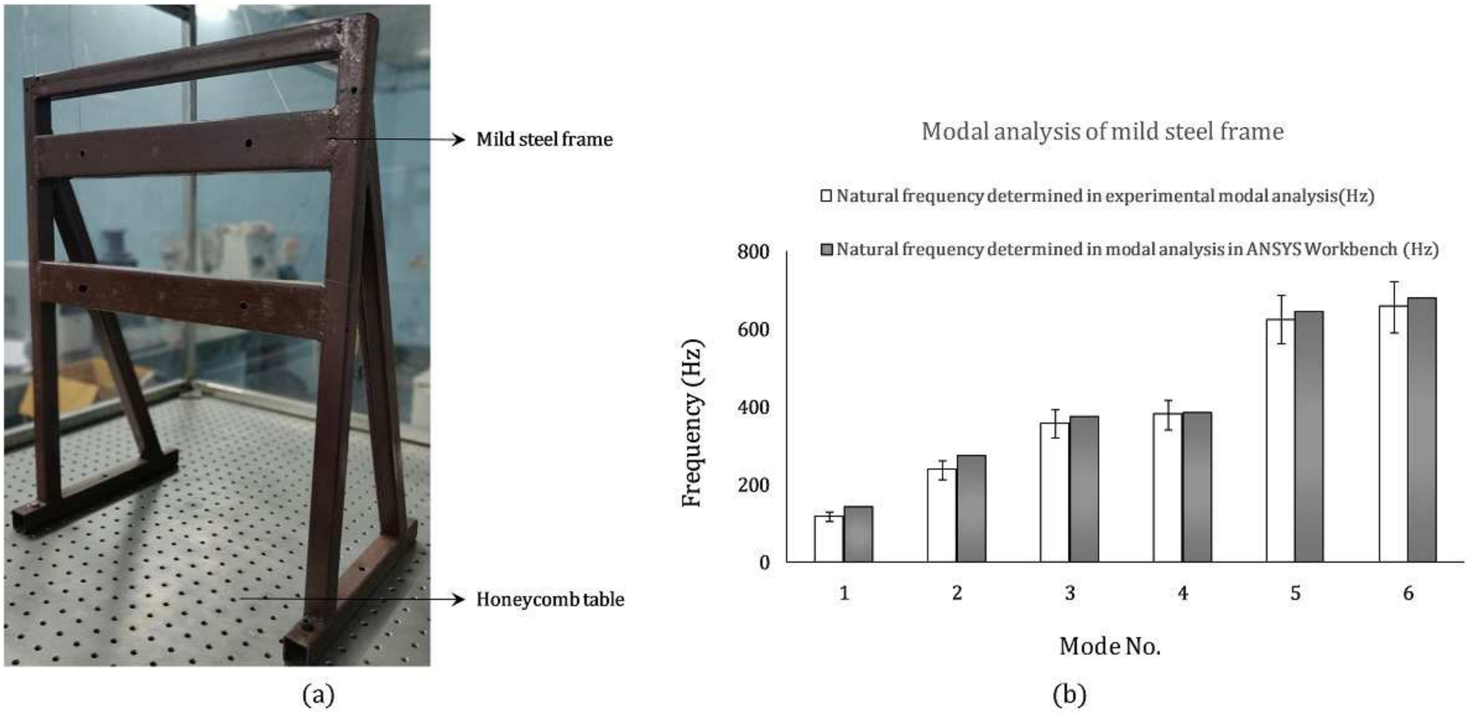


Figure 3

(a) The mild steel frame for the micro-milling set up, (b) the comparison of natural frequencies of the mild steel frame determined in ANSYS modal analysis and experiments

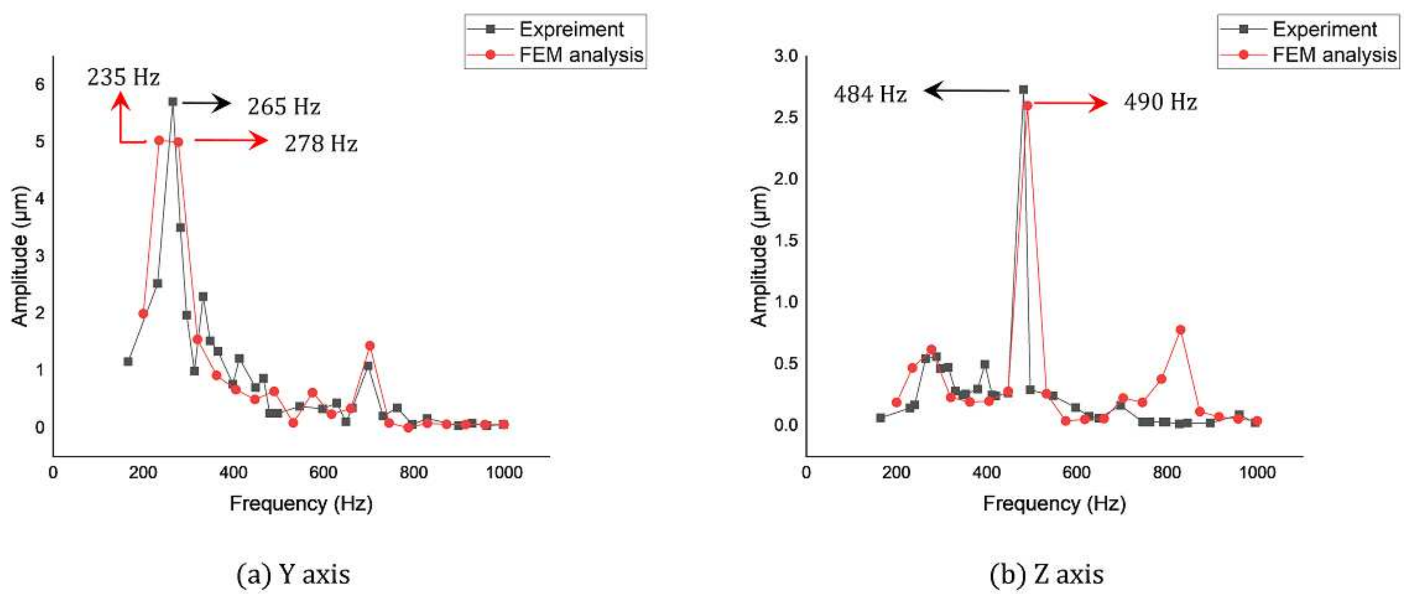


Figure 4

Comparison of the results of frequency response analysis on the mild steel frame determined in experiments and FEM analysis

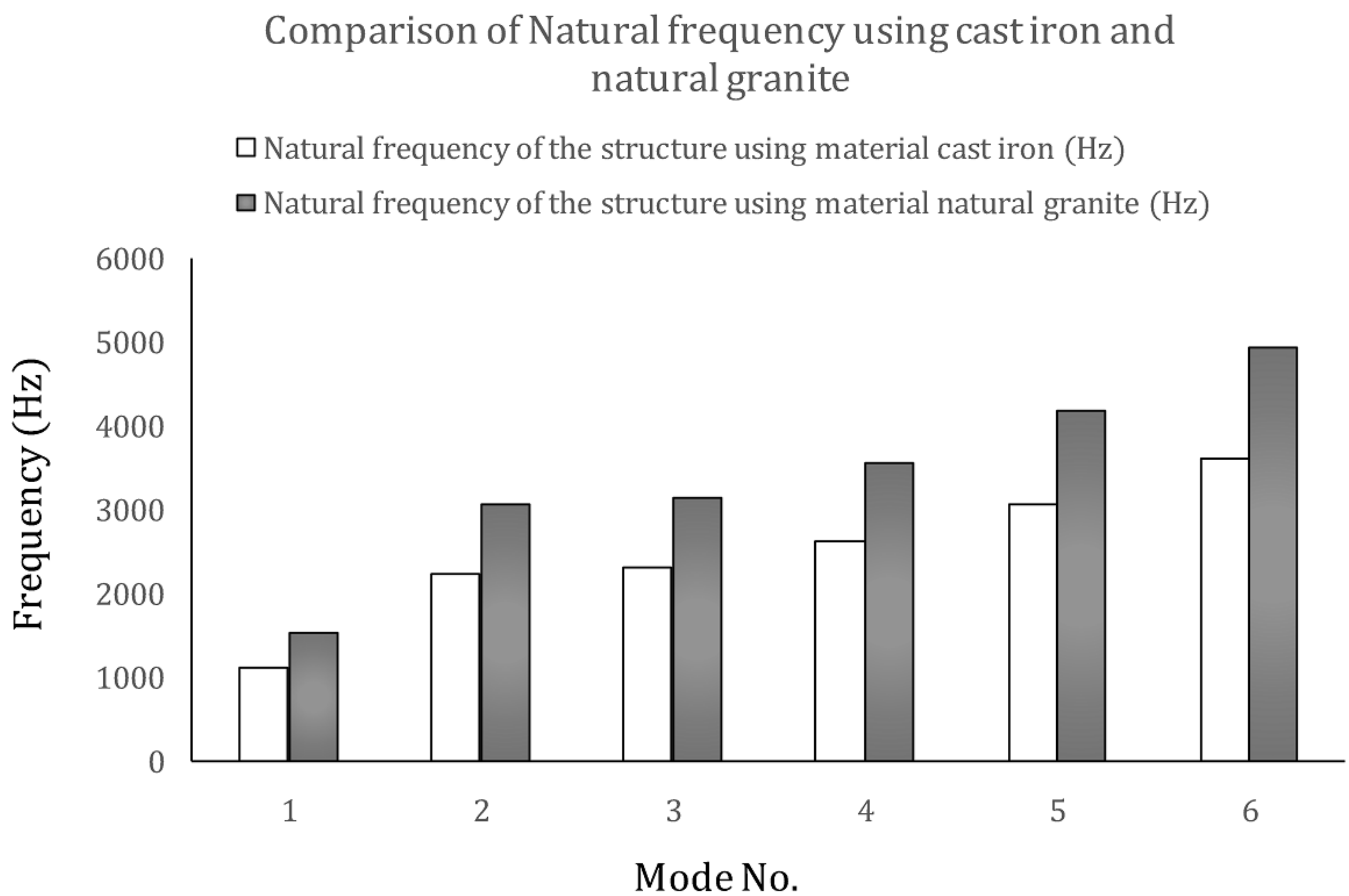


Figure 5

The comparison of natural frequencies of the proposed machine structure using cast iron and natural granite

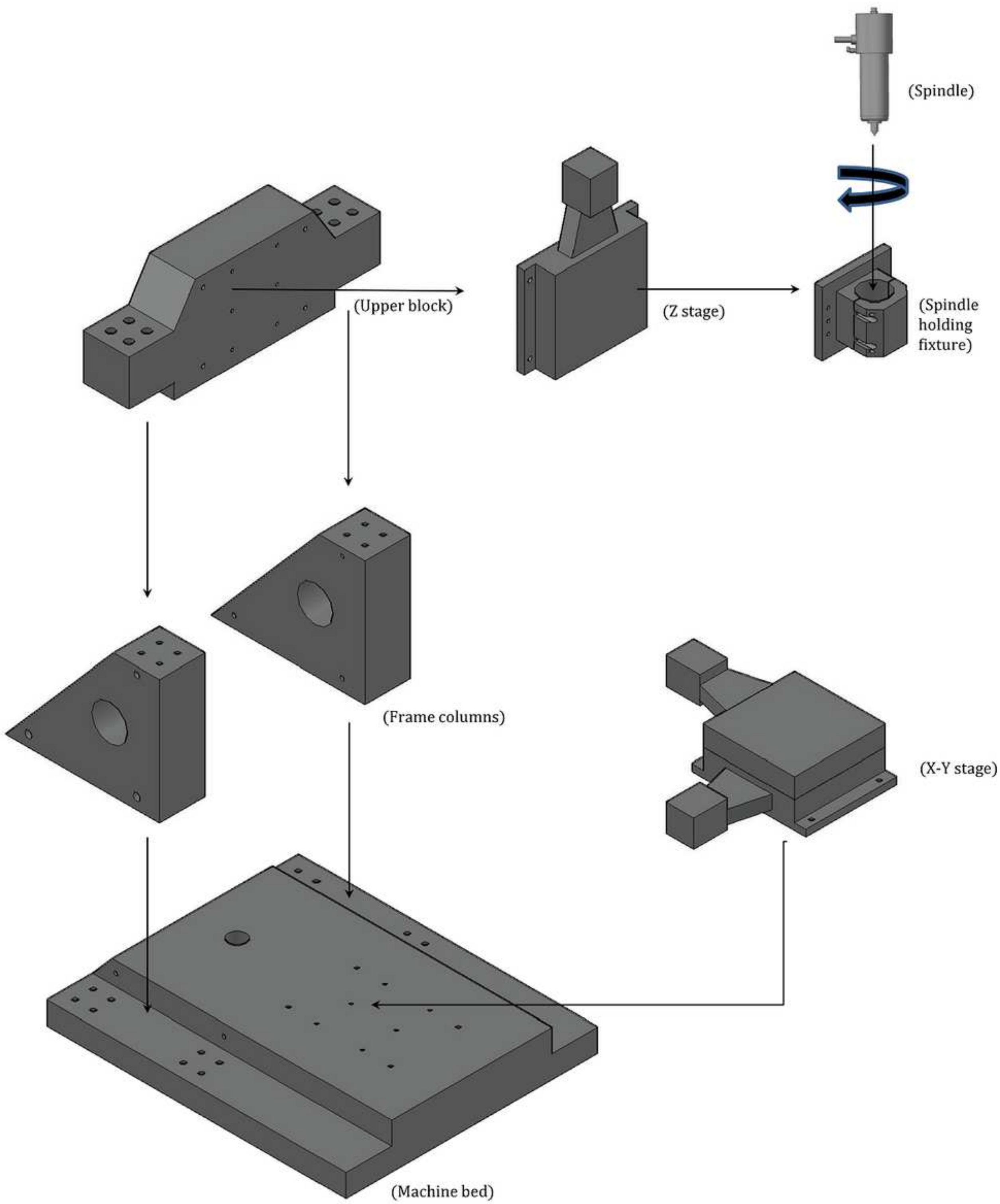


Figure 6

The components of the vibration free high-speed micro-milling center prepared in AutoCAD

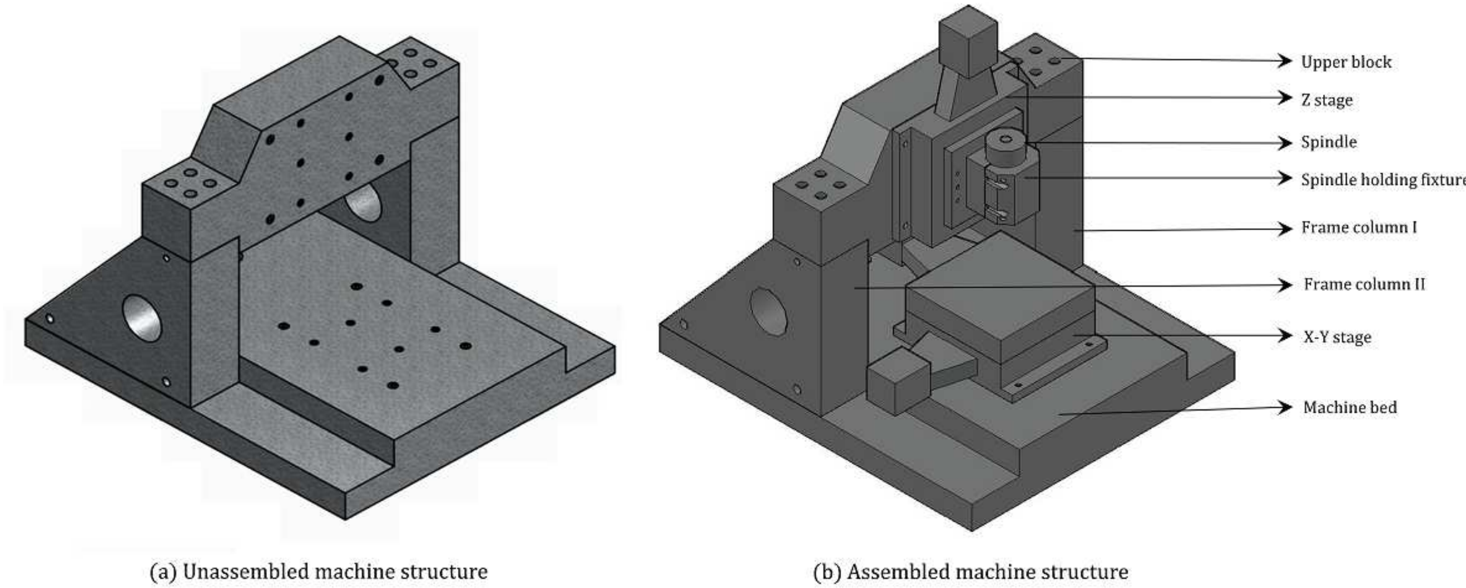


Figure 7

CAD model of the vibration free machine structure

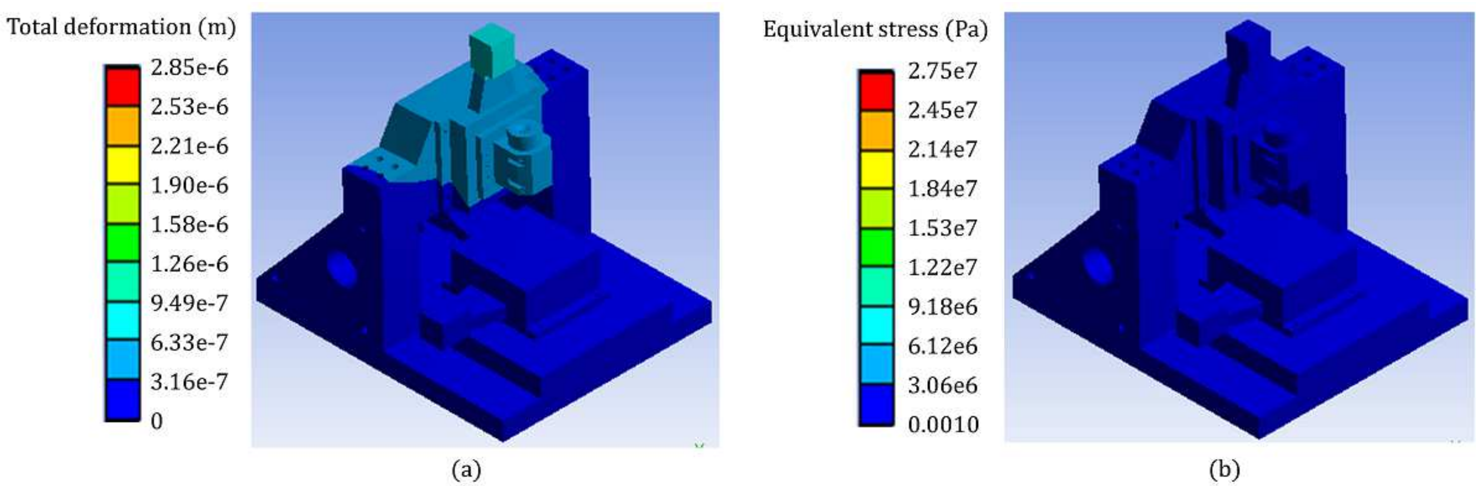


Figure 8

The results of static structural analysis of the proposed model determined in ANSYS workbench. (a) Total deformation, and (b) Equivalent stress

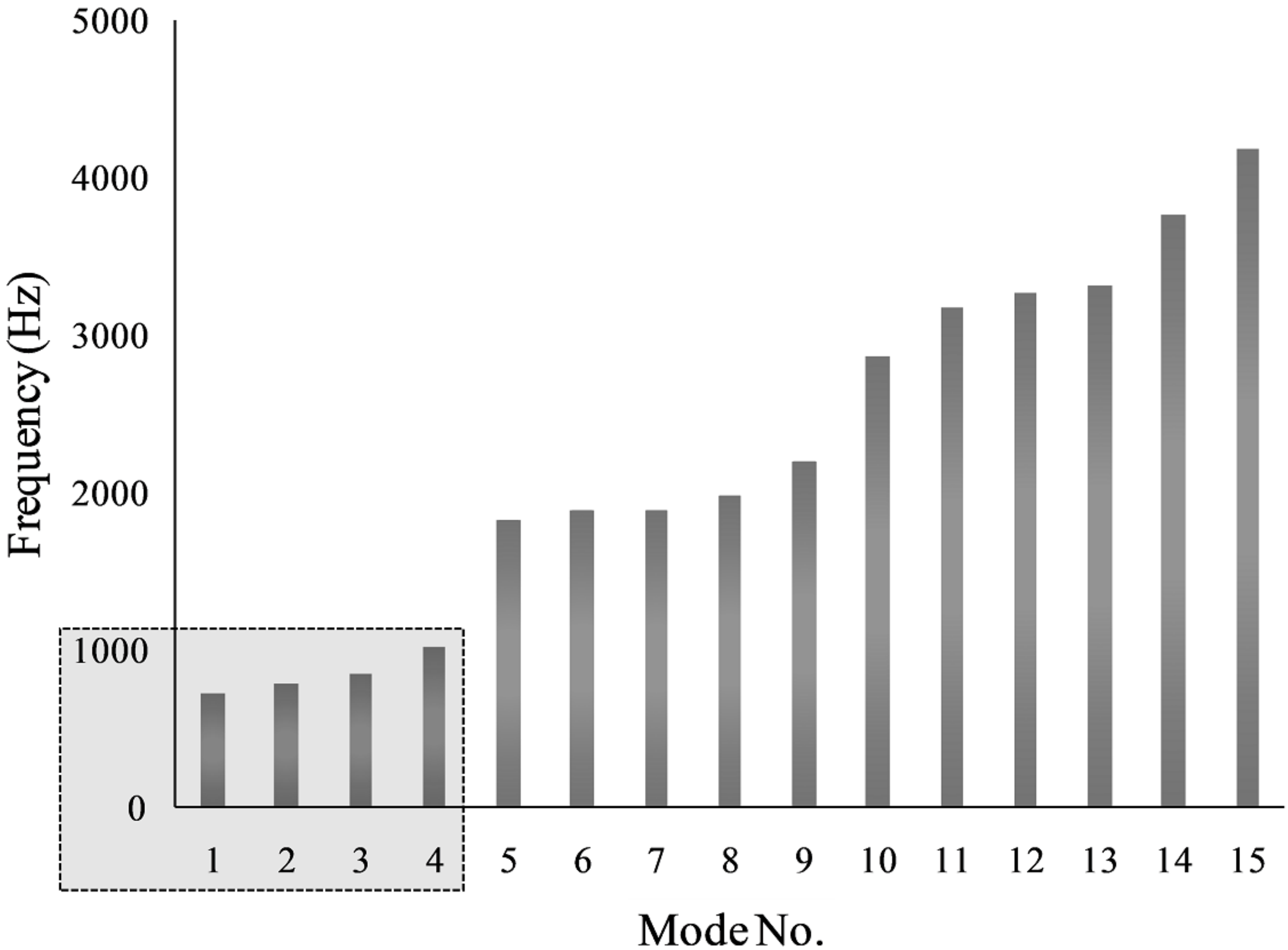


Figure 9

Plots of rst 15 natural frequencies of the proposed assembled machine structure determined in ANSYS modal analysis

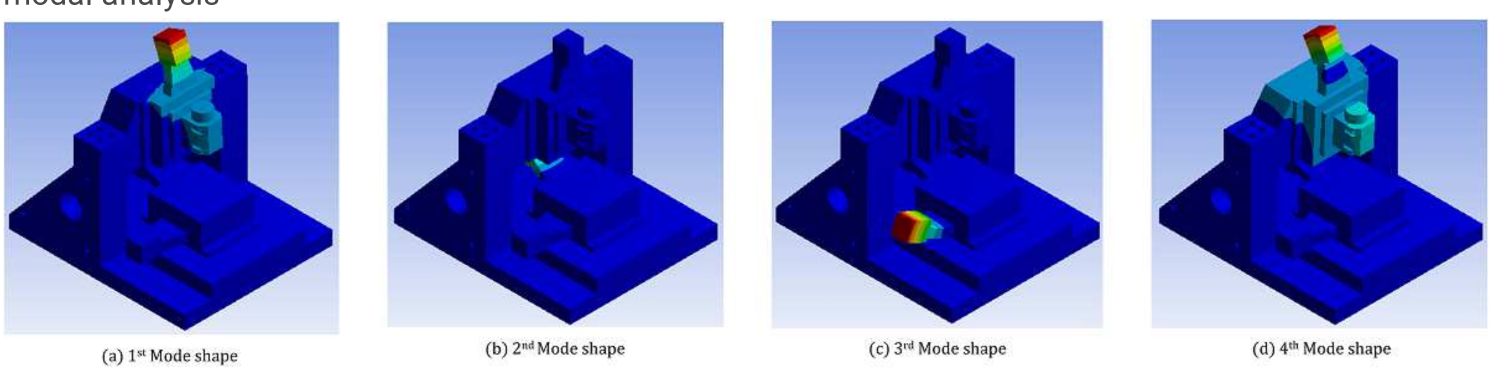


Figure 10

First four mode shapes of the assembled machine structure, determined in ANSYS modal analysis

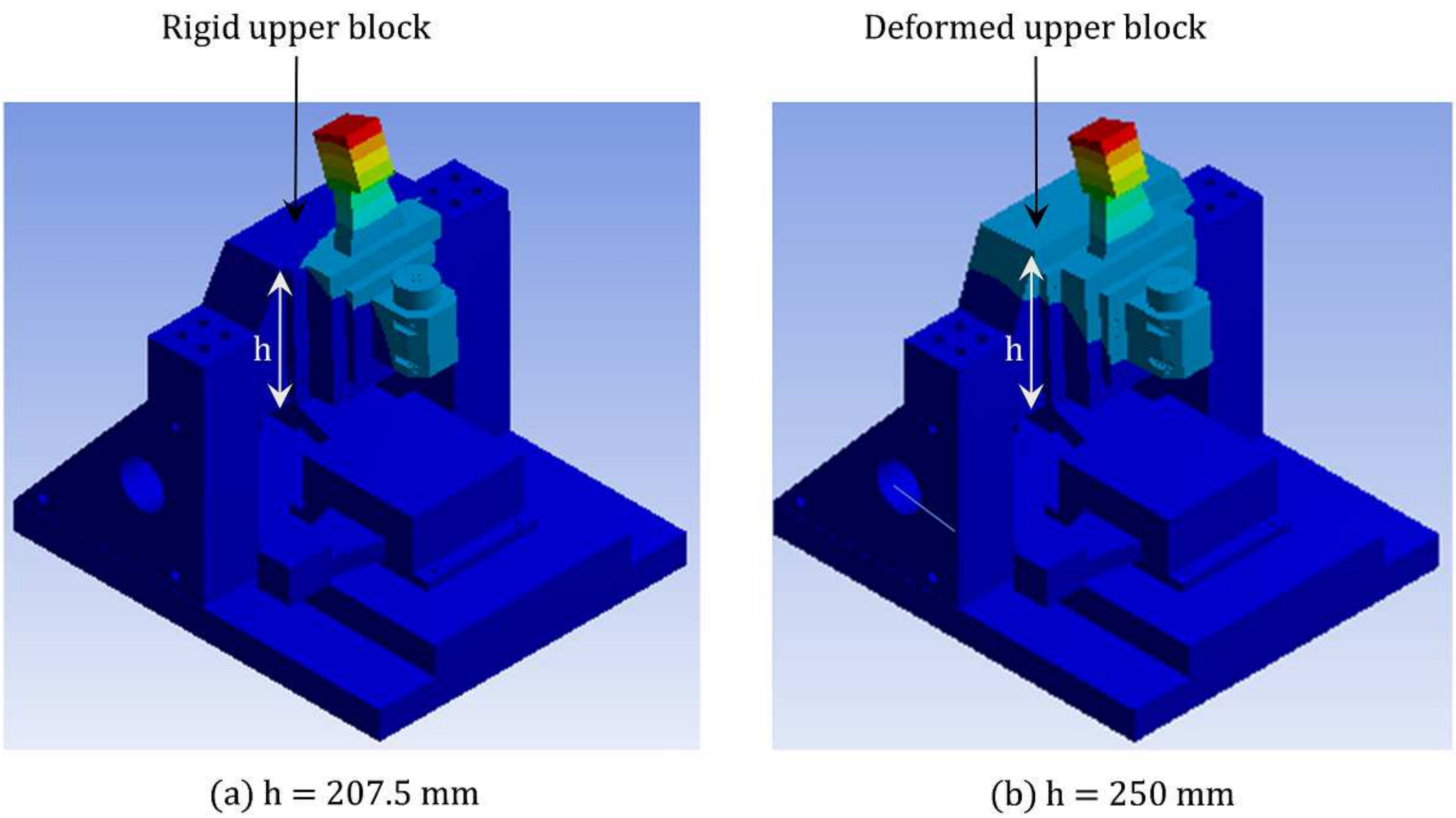


Figure 11

First mode shapes of the assembled machine structure with different heights of the upper block

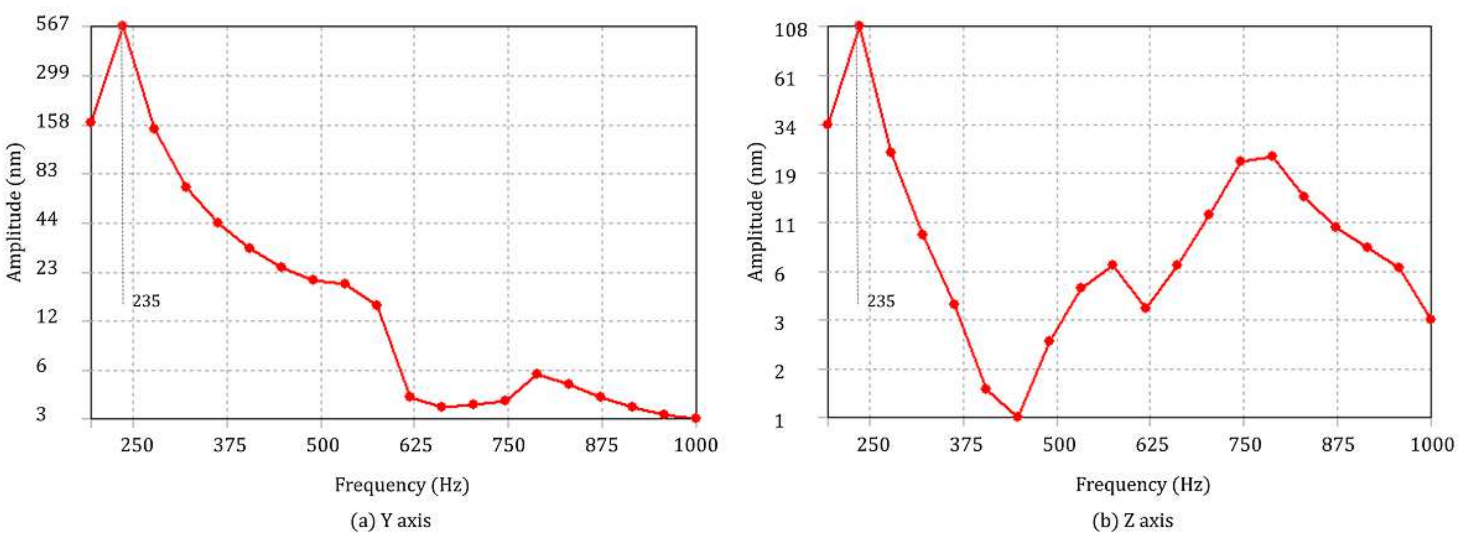


Figure 12

Plots of harmonic response analysis of the assembled machine tool model determined in ANSYS

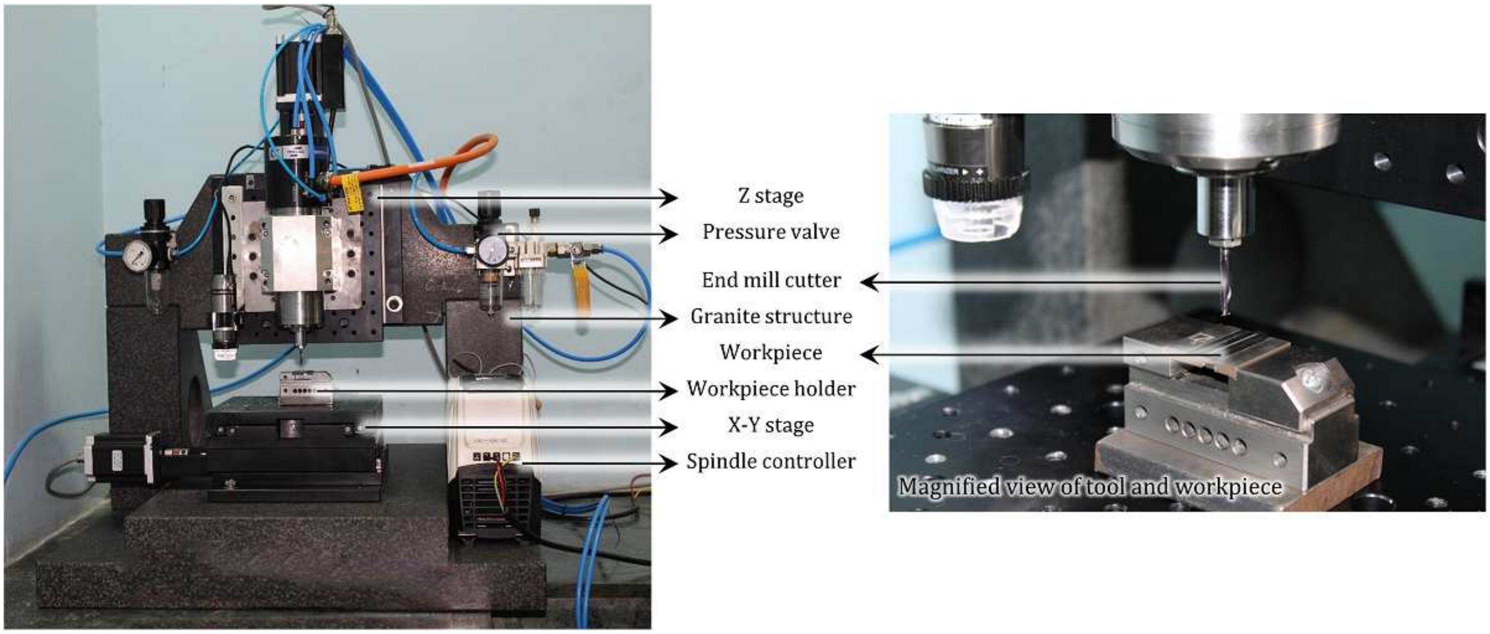


Figure 13

Assembled high-speed micro-milling machine tool with the proposed machine structure made of natural granite

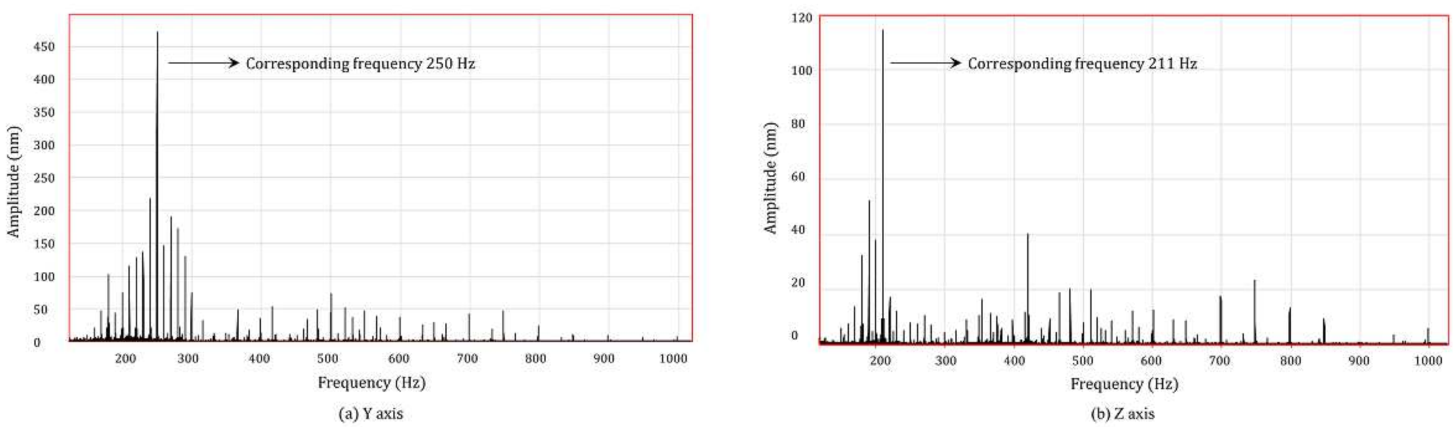


Figure 14

The results of experimental frequency response analysis of the developed high-speed micro-milling center while machining with two ute milling cutter

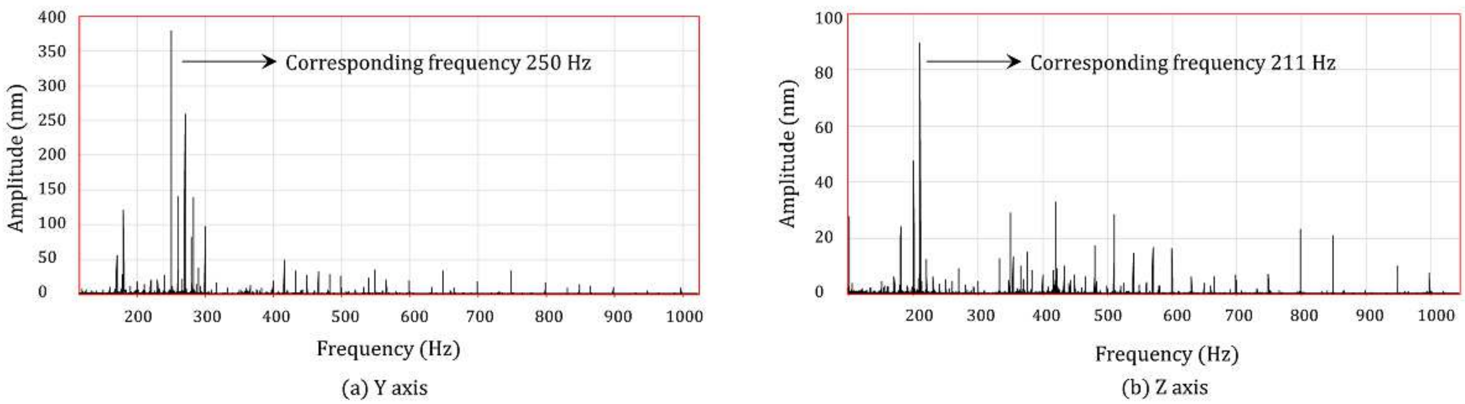


Figure 15

The results of experimental frequency response analysis of the developed high-speed micro-milling center while machining with four ute milling cutter

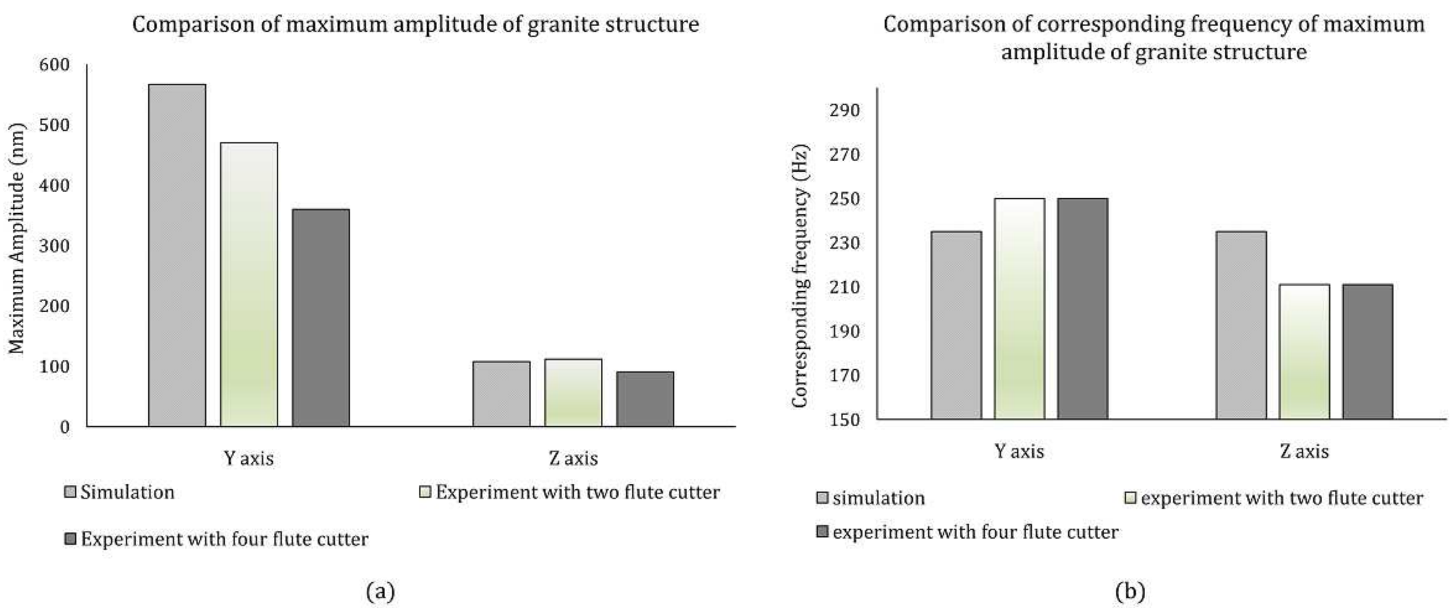


Figure 16

(a) Comparison of maximum amplitude; and (b) comparison corresponding frequencies of the developed micro-milling center, determined in ANSYS harmonic response analysis, experimental frequency response analysis with two ute cutter and experimental frequency response analysis with four ute cutter

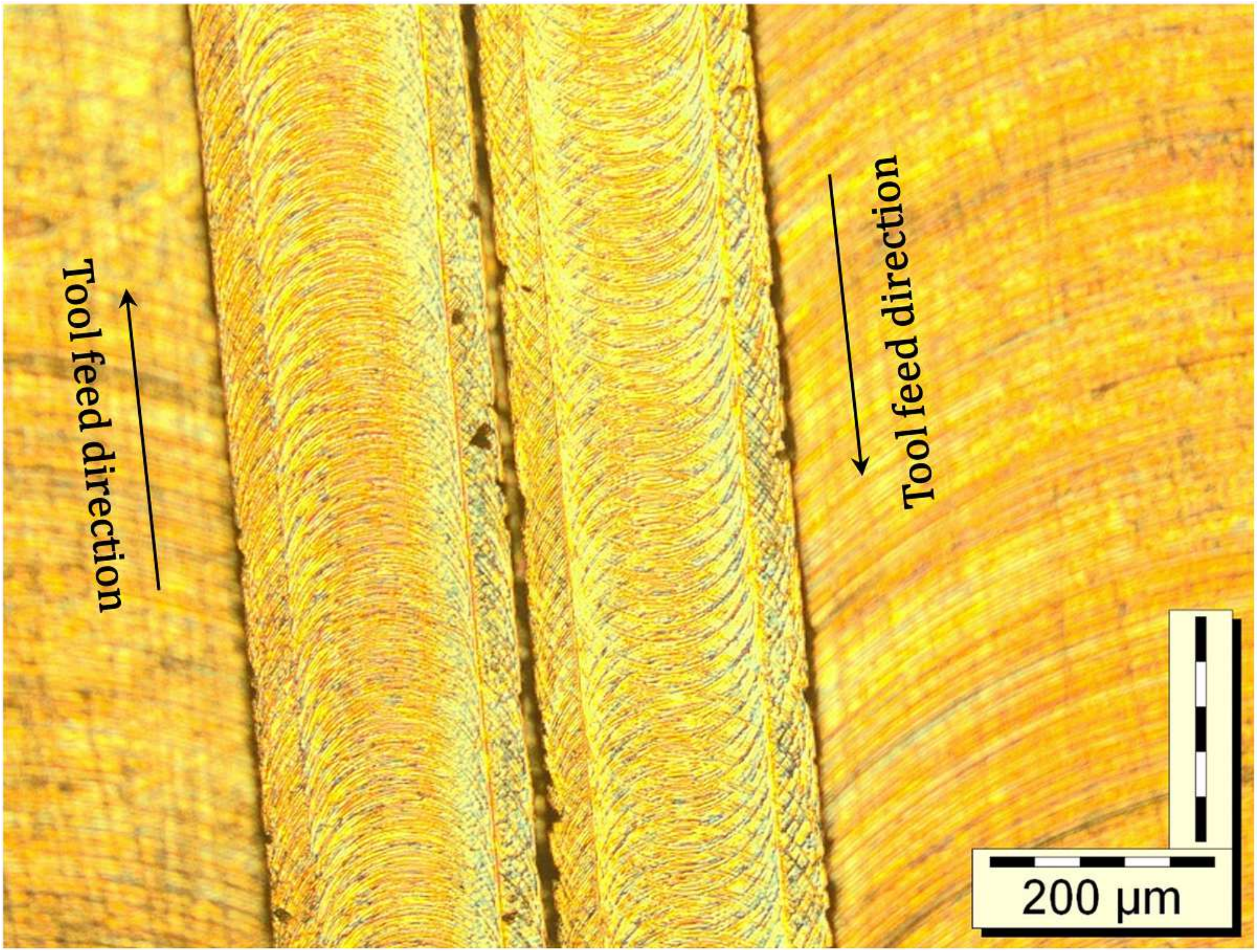


Figure 17

Tool feed marks on micro grooves of 200 µm width and 60 µm depth, generated on high strength super alloy Ti6Al4V by high-speed micro-milling