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STRUCTURAL REDESIGNING OF A CNC LATHE BED TO IMPROVE ITS STATIC AND DYNAMIC CHARACTERISTICS

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ABSTRACT: In the past, the design of CNC machine tools focused on their functional aspects and was hard to acquire any resonance with customers. Nowadays, despite the needs of low price, capabilities withstand at higher cutting speeds and operate at high acceleration and deceleration with high quality machine, many customers request good-looking machine. Regarding this, our study aims to provide various form designs of machine tool structure with the help of structural modifications made in CNC machine tool bed. After the lightening effect was verified by finite element simulation, scale-down models of an original bed and vertical ribs with hollow bed models were fabricated using rapid prototyping method and tested. The dynamic characteristics of those different form designs of the bed were analyzed experimentally. Numerical analysis was done and results were validated with experimental results. Results indicated that the cross and horizontal rib with hollow bed can increase the specific stiffness by 8% with 4% weight reduction and its dynamic performances is also better with increases in the first natural frequencies. The modified design is effective in improving the static and dynamic structural performances of high speed machine tools.
KEYWORDS: Machine tool bed, Form designs, Stiffness, Natural frequency

❖ INTRODUCTION

The proper design of machine-tool structures requires a thorough knowledge of their forms designs and properties of the material, the dynamics of the particular machining process and nature of the forces involved [1]. Recently, researches have been conducted in the field of improvement of structural design of machine tools by improving stiffness and lightening weight which is especially urgent for the structural parts, such as bed, column, worktable, beam and so on [2]. The arrangement of stiffening - ribs in machine tool structures is a key factor for structural stiffness and material consumption. So the lightening design of stiffening-ribs is significant for machining performance and energy saving.

It is generally accepted that the precision of machine tools is determined by their static, dynamic and thermal characteristics. Especially, the dynamic characteristics play an important role in high speed, precision machine tool structures, because vibration during the machining process results in chatter marks on the machined surface and thus creates a noisy environment [3].

High static stiffness against bending and torsion, good dynamic characteristics as reflected by high natural frequency and high damping ratio, ease in production, good long term dimensional stability, reasonably low coefficient of expansion, low cost and low material requirements are the basic properties of machine tool structures that engineers look for designing and fabricating. However, from user's point of view, machine tool vibration is an important factor because it adversely affects the quality of the machined surface. To improve both the static and dynamic performances, the machine tool structures should have high static stiffness and damping. Using either higher modulus material or more material in the structure, the static stiffness of a machine tool may be increased. But, it is difficult to increases the dynamic stiffness of a machine tool with these methods and increase in the static stiffness cannot increase it damping property. Material distribution plays on the important role in the structural strength and using material in required place can increase static stiffness with less mass.

Higher cutting speeds can be facilitated only by structures which have high stiffness and good damping characteristics. The deformation of machine tool structures under cutting forces and structural loads are responsible for the poor quality of products and which in turn is also aggravated by the noise and vibration produced. In many a situation, it is the level of deformation and vibration that determines the upper limit on the ability of the machine to produce components with high precision. All these above said deleterious effects greatly necessitate constant innovations and continuous research to keep them under check. Increasing structural stiffness could help in avoiding such problems. To increase the static stiffness and damping, different form designs can be used.

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The high speed machining process requests completely new demands for the mechanism of such processing equipment, as due to the process, path speeds exceeding 50m/min can be achieved. In this field, potential capacities of manufacturing processes require a dynamic behavior ten times higher than the conventional machine tools and increased accuracy. This can be solved by the systematic evaluation of suitable machine kinematics, by the application of linear direct drives as well as by mass reduction of the axis through light weight components of machine tool structure. The requirements of high speed machining and ways to improve the performance of machine tool have been studied [4].

Hollow boxes possess an efficient shape for engineering components due to their high inherent bending and torsional rigidities. For example, box-section steel girders are a familiar design of beams in bridges and other civil engineering structures. Currently, industrial interest exists in the use of tubes for the moving head of a milling machine. The milling machine heads have the topology of rectangular tubes with monolithic walls. The overall compliance of the milling head is partly due to macroscopic bending of the tube and partly due to the local compliance at the supports on the guide-rails.

The structural response was analyzed for beams of square sections with various internal topologies: a solid section, a foam-filled tube with monolithic walls, a hollow tube with walls made from sandwich plates, and a hollow tube with walls reinforced by internal stiffeners. Finite element analysis was used to validate analytical models for the overall stiffness of the tubes in three-point bending. Minimum mass designs were obtained as a function of the overall stiffness, and the relative merits of the competing topologies were discussed [5].

The weights of optimal compression structures of several types were studied and estimated. Minimum weights of columns having solid square or circular cross sections were compared with those of similar metal foam filled tubes in hollow tubes and tubes whose walls are foam core sandwiches. Similarly the minimum weights of wide sandwich compression panels were studied along with those of hat-stiffened, solid skin panels and panels in which the skins and stiffeners are themselves metal foam core sandwiches [6].

The minimum deflection and weight designs of laminated composite plates were studied. The finite element method using plate theory was used in conjunction with optimization routines in order to obtain the optimal designs. Various boundary conditions were considered and results were given for varying aspect ratios and for different loading types [5]. The analysis for slender beams with a varying cross-section under large non-linear elastic deformation was conducted. A thickness variation function was derived to achieve optimal - constant maximum bending stress distribution along the beam for inclined end load of arbitrary direction [7].

Internal stiffeners support the monolithic walls of the tube and increase the local bending stiffness adjacent to the supports. The shape, size, and orientation of stiffeners decide the improvement in stiffness. The compliance of the machine tool is one of the prime factors for deciding the static and dynamic characteristics and thus results the quality and performance.

Theoretical analysis and experimental tests showed that the size and distribution of the stiffening ribs could affect the stiffness and performance of a structure [8]. Kim et al designed composite-foam-resin concrete sandwich structures for the design of machine tool [9]. Chang and Lee et al have fabricated a hybrid column by adhesively bonding glass fabric epoxy composites laminates to the cast iron column. Results show that the dynamic characteristic of the column was improved [10,11]. Although the stiffness of machine tool structures can be increased either by employing higher stiffness materials or by increasing the sectional modulus of structures [12]. In machine tool, bed forms the vital part of the machine tool on which table and other relevant parts of the machine tool move. Many researches were carried out to enhance the performance of the machine tool bed by improving material property and optimizing structural attributes. It is therefore evident that the bed should be sufficiently strong and rigid. Further it should be easier to remove the chips produced during machining operation [13].

The main objective of this study is to increase the structural stiffness, reduce the weight and deformation of machine tool bed by designing and fabricating suitable forms, and experimentally validating them. The conventional distribution was re-arranged by mimicking above structures. Finite element simulation revealed that the modified type bed of cross and vertical ribs with hollow structure possesses better static and dynamic properties when compared with the conventional type. Scale-down models of original and modified beds were fabricated via Rapid Proto-typing Manufacturing (RPM). The results of the experiment were found to match with that of the static and dynamic analysis conducted through analytical results. The results were encouraging.

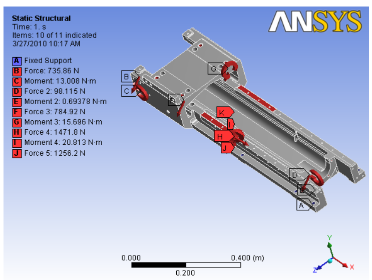


Fig. 1. Loading conditions on CNC lathe bed

❖ NUMERICAL STATIC ANALYSIS OF FORM DESIGNS OF BED

Before redesigning, the existing bed was subjected to static analysis considering the worst case cutting forces. The cutting forces were based on the data obtained from the company and the parameters at which these cutting forces result are Maximum depth of cut: 1mm; Component hardness: 30HRC; Maximum feed of X: 0.10mm/rev; Maximum feed of Z: 0.2 mm/rev; Spindle working speed:

700 rpm; Spindle power: 3.7 Kw. Apart from the cutting forces, four other static loads along with their respective moments were also considered. The loads along with their respective points of application and the supports for the bed are shown below in the figure 1.

The results of the FEM analysis conducted through ANSYS software indicated that the portion of the bed bearing the headstock region suffered the maximum deformation.

STRUCTURAL MODIFICATION OF MACHINE TOOL BED

The bed was then considered for alternate structures and emphases were laid on retaining the existing manufacturing method and hence restrict modifications to only structural redesigning. The static analysis results were once again scrutinized to identify locations where excessive, unnecessary material was present and areas where material mass could be supplanted by rib structures.

The presence of such rib structures proved beneficial as they reduced weight. In the above process, twelve different structural combinations were identified and each of them was analyzed under the prevalent conditions. The models along with the results of analyses are presented. From the table 1a and table 1b, it can be observed that two models i.e. Cross and Horizontal Ribs with bionic, vertical ribs with hollow display optimal characteristics. One of these two models, vertical ribs with hollow was taken up for further material redistribution so that the deformation could be reduced to existing levels while reducing only weight as shown in figure 2.

The existing and modified model of machine tool bed was then fabricated using the Rapid Prototyping Process.

Table 1a. Numerical Static Analysis comparison of various structurally modified bed

CATEGORY	EXISTING BED	CONVENTIONAL RIBS WITH BIONIC	CONVENTIONAL RIBS WITH HOLLOW	CROSS RIBS WITH BIONIC	CROSS RIBS WITH HOLLOW	PLUS RIBS WITH BIONIC	PLUS RIBS WITH HOLLOW
3D Model of Bed							
2D Drawing of Bed							
Weight 'W' (kg)	100.18	92.865	91.332	90.326	89.54	89.713	89.713
Headstock Critical Point	2.8429e-006	3.1371e-006	3.2441e-006	3.1656e-006	3.101e-006	3.1899e-006	3.0904e-006
Headstock Critical Point Deformation 'd' (μm)	2.8433	3.1391	3.2460	3.1662	3.1026	3.1901	3.1901
Specific Stiffness at Headstock Critical Point E/(W*d)	386.18	376.1	371.0	384.6	395.96	384.355	384.35
Bed Critical Point	1.6186e-006	1.8221e-006	1.7305e-006	1.844e-006	1.8547e-006	1.8376e-006	1.9102e-006
Bed Critical Point Deformation 'd' (μm)	1.619	1.8231	1.7306	1.8450	1.8549	1.8376	1.8375
Specific Stiffness at BedCritical Point E/(W*d)	678.21	649.7	695.9	660.6	662.301	667.247	667.28
% Reduction in Weight	-	7.30	8.83	9.84	10.62	10.45	10.25
I_{xx} (kg m^2)	1.0136	0.9564	0.9149	0.9439	0.9243	0.93816	0.9382
I_{yy} (kg m^2)	8.3504	7.9224	7.7835	7.7866	7.725	7.7498	7.7498
I_{zz} (kg m^2)	7.6914	7.2968	7.1965	7.1657	7.1217	7.1313S	7.1313

Table 1b. Numerical Static Analysis of Form design of Machine tool bed

CATEGORY	EXISTING BED	COMBINED RIBS WITH BIONIC	COMBINED RIBS WITH HOLLOW	CROSS & HORIZONTAL RIBS WITH BIONIC	CROSS & HORIZONTAL RIBS WITH HOLLOW	CROSS & VERTICAL RIBS WITH BIONIC	CROSS & VERTICAL RIBS WITH HOLLOW
3D Model of Bed							
2D Drawing of Bed							
Weight 'W' (kg)	100.18	91.272	90.649	90.912	90.144	90.828	90.044
Headstock Critical Point	2.8429e-006	3.161e-006	3.0954e-006	3.155e-006	3.1035e-006	3.1545e-006	3.0967e-006
Headstock Critical Point Deformation 'd' (μm)	2.8433	3.1637	3.0958	3.1603	3.1053	3.1547	3.0968
Specific Stiffness at Headstock Critical Point E/(W*d)	386.18	380.9	391.97	382.9	392.96	383.9	394.5
Bed Critical Point	1.6186e-006	1.7909e-006	1.8588e-006	1.7971e-006	1.8238e-006	1.8418e-006	1.8295e-006
Bed Critical Point Deformation 'd' (μm)	1.619	1.7913	1.8588	1.7985	1.8238	1.8420	1.8299
Specific Stiffness at BedCritical Point E/(W*d)	678.21	672.8	652.825	672.8	669.081	657.481	667.591
% Reduction in Weight	-	8.89	9.51	9.25	10.02	9.34	10.12
I_{xx} (kg m^2)	1.0136	0.9477	0.9295	0.9457	0.9264	0.9470	0.9275
I_{yy} (kg m^2)	8.3504	7.8352	7.7827	7.8168	7.7564	7.8127	7.7513
I_{zz} (kg m^2)	7.6914	7.2140	7.1789	7.1965	7.1537	7.1907	7.1469

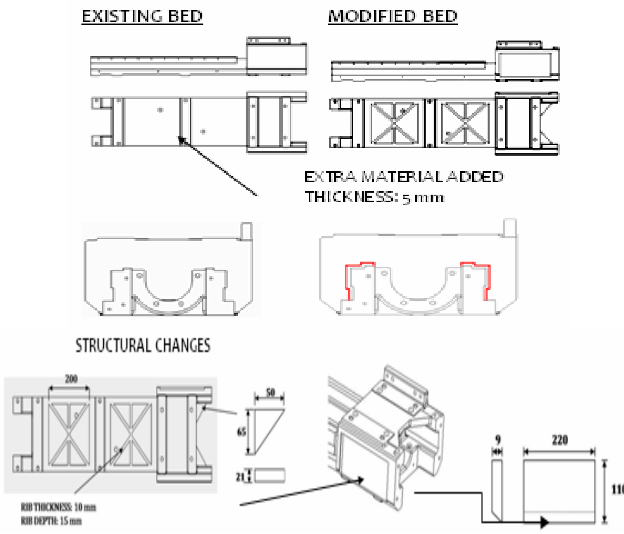


Fig. 2. Modified bed after final modifications

FEM SIMULATION AND EXPERIMENTS

FEM simulation was carried out using ANSYS software to analyze both the conventional and the cross and vertical ribs with hollow bed, in terms of static and dynamic characteristics studies. The 3-D models were established in a CAD system, Pro/E Wildfire 5.0, and were imported into ANSYS. The models were then modified or simplified to meet the FEM requirements. The material used was gray cast iron and the material specifications are listed in table 2. All DOFs of bottom surface were restricted and external loads were applied to corresponding positions of bed.

The deformation distributions and the static performances of the two beds are shown in Fig. 3a, 3b and results are tabulated in table 3.

Table 2. Material properties

Material	Elastic Modulus E (MPa)	Poisson's Ratio	Density (kg·m $^{-3}$)
Gray cast iron	1.1e5	0.28	7200
ABS	3.2e3	0.35	1200

Table 3. The comparison of simulation results

Bed	Weight W(kg)	Deformation (μm)	Specific stiffness E/Wd
Original type	100.18	1.619	678.21
Vertical ribs with hollow bed	96.887	1.5534	730.88
	-3.29	-4.05	+7.77

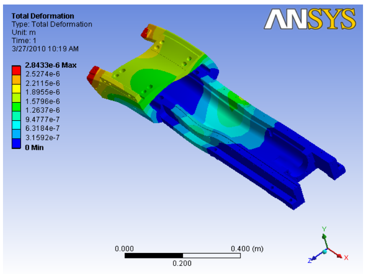


Fig.3a. Existing bed

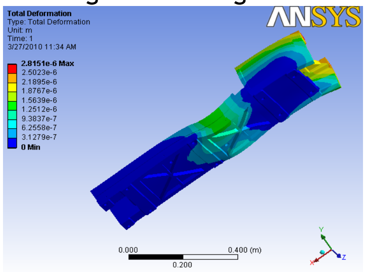


Fig.3.b. Modified bed

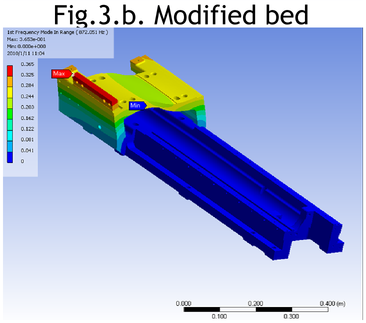


Fig.4. Existing bed

The modified model has a reduced deformation along with reduction in weight which ultimately increases specific stiffness. Thus the new model proves to be better than the existing model by all means. The maximum displacement of vertical ribs with hollow bed is reduced by about 4.05% with 3.29 % mass reduction, leading to the improvement of specific stiffness by 7.77%.

NUMERICAL MODAL ANALYSIS OF EXISTING BED

The dynamic performances reflect structure's resistance to vibration. So modal analysis was performed to determine the first four order natural frequency and the results are shown in table.4. First mode frequency of existing bed is shown in figure 4.

Table 4: Numerical modal analysis of existing bed- First 5 Natural frequencies in Hz

Mode	1	2	3	4
Existing bed - Natural frequency(Hz)	872.05	1175.88	1280.52	1340.48

EXPERIMENTAL MODAL ANALYSIS OF EXISTING BED

Among the many different methods to validate the CAD model, the natural frequency based evaluation method was adopted. The natural frequency of the bed was found experimentally through the impact hammer test (ASTM C215-91) using a piezoelectric accelerometer, NI PXI 1042Q, LabVIEW software, etc. The impact pulse indicating the magnitude of the input force was generated by the impact hammer. The frequency domain response was obtained by using signal analyzer available in sound and vibration toolkit of Lab VIEW. The experimental setup and response of the bed captured in time and frequency domains as shown in figure 5.The values of natural frequency thus obtained experimentally were then compared with the results of modal analysis conducted using ANSYS. The results were found to be almost similar to each other. The comparison is shown in table5.

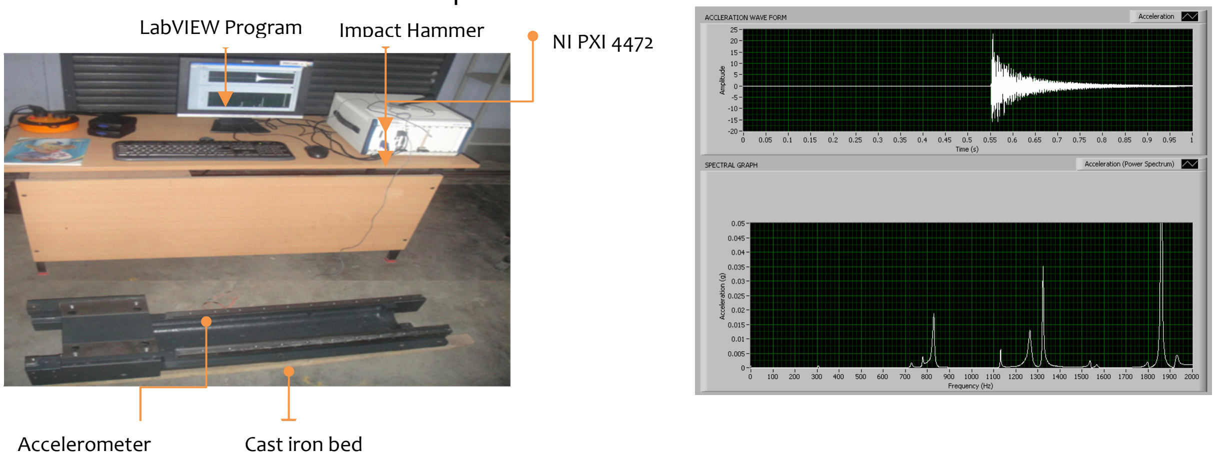


Fig.5. Experimental setup for modal analysis

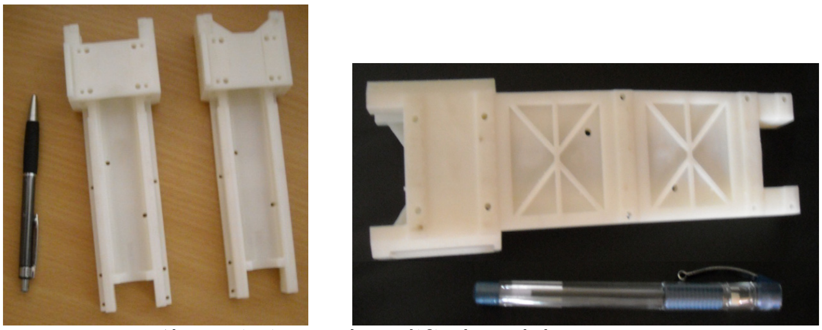
FABRICATION OF EXISTING AND MODIFIED BED

Table 5. Comparison of modal frequencies

Mode		1	2	3	4
Existing bed - Natural frequency(Hz)	Analytical	872.05	1175.88	1280.52	1340.48
	Experimental	840	1140	1260	1320

Stereolithography was adopted to fabricate the existing and modified bed to be used in the experiments. Taking cost and operation convenience into consideration, scale-down bed with scale ratio = 1:5 were produced. The structural dimensions were reduced based on similarity theory to ensure that the mechanical experiments on the scaled models could result in the same authenticity and credibility. All loads were calculated conformable to similarity theory. The material of the models was ABS (Acrylic Butadiene Styrene) plastic with properties as shown in table3. The SLA (Stereolithography Apparatus) machine along with the fabricated model of original and modified bed is as shown in figure 6a and 6b respectively.

Fig. 6a. Stereolithography



6b. Existing and modified model

FEM simulation of scale-down models showed that static specific stiffness was increased by 7.77% and first four order natural frequencies was improved by 75% in average. So static and dynamic experiments were conducted for both models to evaluate and compare their performances.

EXPERIMENTAL STATIC ANALYSIS EXISTING AND MODIFIED BED

The setup of the static test is shown in figure 7. The bottom was fixed with a support. According to FEM analysis all static loads were applied to corresponding areas. Digital dial-indicators were used to measure the displacement. The measurement of each point was repeated five times.

Figure 7. Experimental setup

EXPERIMENTAL MODAL ANALYSIS OF EXISTING AND MODIFIED BED

Any physical system can vibrate. The frequencies at which vibration naturally occurs, the modal shapes which the vibrating system assumes are properties of the system, and can be determined using modal analysis. Modal analysis is frequently utilized to abstract the modal parameters of a system, including natural frequencies, mode shapes and modal damping ratio. Since these parameters depend only on the system itself but dominate the response of the system to excitations. Modal analysis is the fundamental response analysis and has therefore gained increasing attentions.

The setup of the dynamic test is as shown in figure 5. The vibration responses of the bed of different forms were obtained using impact testing, where an instrumented hammer was used to excite the bed. The resulting vibration was measured using a low mass accelerometer. The experimental setup consisted of an impact hammer, a charge accelerometer, signal conditioner and data acquisition using LabVIEW software to record the response of the bed in time and frequency domains. The frequency domain response was obtained by using signal analyzer available in sound and vibration toolkit of Lab VIEW. This experimental result of the modal analysis characteristics of the existing and modified bed is shown in figure 8a and 8b.

Table 6. Experimental results comparison- Modal analysis

Mode		1	2	3	4
Existing bed	Natural frequency (Hz)	1175	2210	3400	4450
Vertical ribs with hollow bed		1542	2100	2150	2189

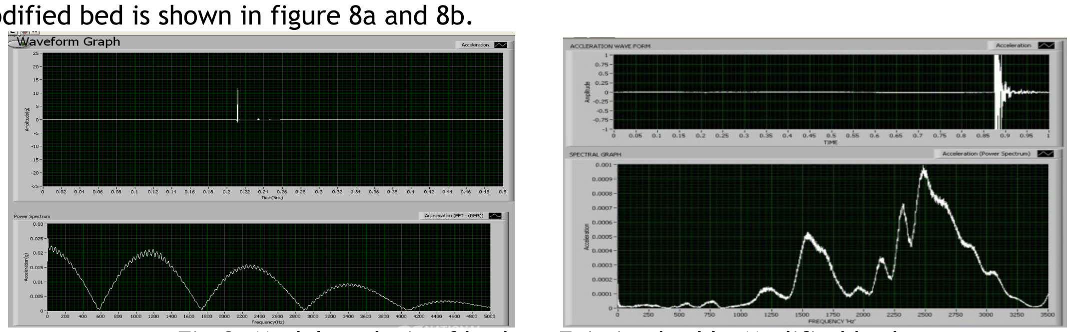


Fig.8. Modal analysis of bed. a. Existing bed b. Modified bed

RESULTS AND DISCUSSIONS

After three times of repeated experiments, the measured displacements were averaged. Final results are listed in table 7. It shows that the maximum displacement of vertical ribs with hollow bed model was reduced by about 8.08% with 3.66% mass reduction. The reductions of structural weight and deformation were more significant than the results of simulation. The possible reasons might be that the bed material in the experiments was ABS plastics rather than cast iron. In any case, however, the vertical ribs with hollow bed model achieved higher specific stiffness, indicating more efficient material distribution than the conventional ones.

Table 7. The comparison of simulation results

Bed	Weight W (kg)	Deformation (μm)	Specific stiffness E/Wd
Original type	0.138	1.619	14.331
Vertical ribs with hollow bed	0.133	1.5534	15.489
	-3.66%	-4.05%	+8.08%

The modal experiments were repeated five times on each bed model. The mode shapes were obtained.

In table 6, the first four natural frequencies are listed. The first natural frequencies of vertical ribs with hollow bed one were increased by about 31.23 %. But the remaining frequencies were lower than those of original type. However, the lowest modal shapes are usually most important for structural vibration. In addition, the first order mode shapes were the bending of bed along the Y direction, which would be critical for machining precision. So it could be concluded that the vertical ribs with hollow bed type reached better dynamic performance. There are several limitations associated with the scaled-model tests due to the different material and technological level. However, the focus is more on the relative effectiveness between original and vertical ribs with hollow bed models. So with the results comparison, the vertical ribs with hollow bed design have improved the static and dynamic performance of the bed, which is encouraging for further study. Casting can be used for manufacture.

CONCLUSIONS

(1) Based on the configuration principles, the existing bed was redesigned to improve the static and dynamic performances. Simulation results show that the static and dynamic performances of vertical ribs with hollow bed have been improved.
(2) Scale-down models were used to verify the improvements of vertical ribs with hollow bed design. Static and dynamic experiments show that the mass and deflection are reduced by 3.66% and 8.08%% respectively and the lower order natural frequencies are increased. Experimental results agree qualitatively with the FEM simulation.
(3) Structural vertical ribs with hollow offers a method to improve the conventional design of machine structure. Based on structural modifications, ribs parameters and distributions can be further optimized.

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