# Dynamic features and optimal lathe bed structure of horizontal machining center

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ABSTRACT. *The static and dynamic performance of the lathe bed structure is essential to the performance of the entire machine. Targeting the lathe bed of a horizontal machining center, this paper analyses the dynamic features and optimizes the lathe bed structure. Firstly, a simplified finite-element model of the lathe bed was established, and the finite-element analysis software ANSYS Workbench was adopted for futher analysis. Secondly, the natural frequencies of the first four orders and vibration modes of the lathe bed were investigated through the static analysis and vibration modal analysis of the slide structure, thereby measuring the dynamic performance of the lathe bed by the natural frequency. Next, the author identified the weak links of the structural stiffness of the lathe bed and proposed an effective solution to these weaknesses. The solution reduced the maximum deformation of the optimized lathe bed by 8.3% and the maximum stress by 0.13%, which achieves the optimization goal. Finally, machining experiments were conducted on a trial machine and a setting machine with the same contour and specifications. The results show that the proposed optimization solution is feasible for the improvement of the lathe bed. The research findings boast profound significance for similar studies.*

RÉSUMÉ. *Les performances statiques et dynamiques de la structure du banc du tour sont essentielles à la performance de la machine entière. Ciblant le banc de tour d'un centre d'usinage horizontal, cet article analyse les caractéristiques dynamiques et optimise la structure du banc de tour. Tout d'abord, un modèle simplifié par éléments finis du banc de tour a été mis en place et le logiciel d'analyse par éléments finis ANSYS Workbench a été adopté pour une analyse ultérieure. Deuxièmement, les fréquences propres des quatre premiers ordres et les modes de vibration du banc du tour ont été étudiés par analyse statique et analyse modale de vibration de la structure de la glissière, mesurant ainsi la performance dynamique du banc du tour par la fréquence naturelle. Ensuite, l'auteur a identifié les maillons faibles de la rigidité structurelle du banc du tour et proposé une solution efficace à ces faiblesses. La solution a réduit la déformation maximale du banc de tour optimisé de 8,3% et la contrainte maximale de 0,13%, ce qui permet d'atteindre l'objectif d'optimisation. Enfin, des expérimentations d'usinage ont été menées sur une machine d'essai et une machine de réglage ayant les mêmes contours et spécifications. Les résultats montrent que la solution d'optimisation proposée est réalisable pour l'amélioration du banc du tour. Les résultats de la recherche ont une signification profonde pour des études similaires.*

KEYWORDS: *natural frequency, dynamic performance, structural optimization.*

Journal européen des systèmes automatisés – n° 3/2017, 285-298

*MOTS-CLÉS: fréquence naturelle, performance dynamique, optimisation structurelle.*

DOI:10.3166/JESA.50.285-298 

## 1. Introduction

The horizontal machining center is composed of a lathe bed, an upright column, a workbench and a spindle box, of which the lathe bed is one of the key components of the machine tool and the basis of the whole machine tool, which mainly plays the role of bearing, supporting all the parts of the machine tool (Rong *et al.*, 2000; Mahdavinejad, 2005; Bao & Tansel, 2000; Bai *et al.*, 2009; Asad *et al.*, 2007). Therefore, the static and dynamic performance of the lathe bed structure is one of the important factors that determine the performance of the machine tool, directly affecting its accuracy and stability. Qiu Zheng, Zhang Song, *et al.*, set up the finite element model of the feed system, conduct analysis using ANSYS Workbench software, and optimize the dimension of the workbench with improvement of its natural frequency as the goal (Belblidia *et al.*, 1999; Rong *et al.*, 2000). Wu Xiuhai *et al.* obtain the characteristic parameter of guide rail joint by dynamic testing method and apply it to digital simulation model, thus improving the precision of the model (Yildirim, 2001). Chen Xin takes the natural frequency and relative vibration displacement as the evaluation criteria for the design scheme to dynamically optimize the grinder lathe bed and the whole machine (Paredes *et al.*, 2001).

This study takes the lathe bed of a large horizontal machining center as the research object. Due to lack of support for the dynamic characteristic analysis in the traditional experience design of the lathe bed, the lathe bed has such problems as vibration and unreasonable distribution of structures. Therefore, it is necessary to dynamically analyze the lathe bed of the horizontal machining center and then to optimize its structure based on the analysis. In this paper, the three-dimensional lathe bed model is established, the static and dynamic analysis of lathe bed structure is carried out using finite element software ANSYS Workbench, and the lathe bed structure is optimized according to the analysis results.

## 2. Theory of dynamic characteristic analysis

The oscillatory differential equation of an undamped system with n degrees of freedom is (Zaeh *et al.*, 2004):

$$[m][\ddot{x}]+[k][x]=[F] \tag{1}$$

Where, $[m]$ is $n\times n$-dimensional mass matrix, $[k]$ is $n\times n$-dimensional stiffness matrix, $[x]$ is n-dimensional column vector, and $[F]$ is external excitation matrix.

That is,

$$[m]\ddot{x}+[k]x=0 \tag{2}$$

Let the solution of the equation be

$$[x]=[A]\sin(\omega t+\theta) \tag{3}$$

Substitute Equation (3) into (2) and simplify it, it can be obtained that

$$[k]-\omega^2[m]=0 \tag{4}$$

Solve this equation to obtain n reciprocal positive roots of ω, $\omega_{oi}$ (i=1, 2, 3…, n), which are usually arranged in ascending order of $0<\omega_{o1}<\omega_{o2}\ldots<\omega_{om}$. Then the i-th order natural frequency is

$$f_{oi}=\frac{1}{2\pi}\sqrt{\frac{k_i}{m_i}} \tag{5}$$

It can be seen that the natural frequencies signify the advantages and disadvantages of their dynamic characteristics. The natural frequency of each order is proportional to K. When the natural frequency is increased, the static stiffness of the structure is also increased accordingly. Therefore, the overall performance of the bed relies on the improvement of the bed structure, which mainly starts from raising the stiffness of the weak links.

## 3. Static analysis of lathe bed structure

### *3.1. Establishment of the finite element model*

The horizontal machining center bed is made of QT-400-15 precision casting, with a dead weight of 8,300kg. The interior mainly adopts square reinforcement design, with the thickness of the gusset of 20mm, and the thickness of the main plate of 25mm. The span of the two guide rails of the X-axis is 800mm and span of the two guide rails of the Z-axis is 750mm. In the process of solid modeling, the chamfering, fillets and grooves are removed and the smaller holes are removed for reasonable simplification. Finally, the simplified three-dimensional solid model of the lathe bed of the horizontal machining center is as shown in Figure 1.

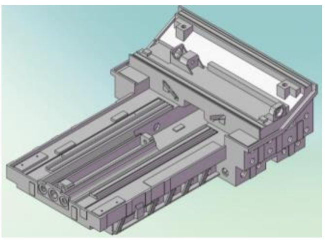


*Figure 1. Simplified 3D solid model of lathe bed*

### *3.2. Static analysis of lathe bed*

According to the research content of the subject and the actual working conditions of the machine tool (the upright column is at the middle position of X-axis travel, slide is near the limit position at the rear end of Z-axis travel, and the spindle box is near the limit of 150mm at the lower end of Y-axis), the horizontal machining center is mainly subject to its own gravity, maximum bearing load of the workbench, and cutting force, while neglecting the influence of other attachment forces. The undertaken force is shown in Table 1.

*Table 1. Dead weight and load of components of the horizontal machining center*

| Project | Unit | Force | Remark |
|---|---|---|---|
| Bed component gravity $G_1$ | N | 90 000 | Chip conveyor |
| Column assembly gravity $G_2$ | N | 25 000 | |
| Spindle assembly gravity $G_4$ | N | 4 500 | |
| Workbench slide and table assembly gravity $G_5$ | N | 9 500 | Workbench center |
| Maximum load-bearing of worktable $G_6$ | N | 12 000 | Maxload of worktable |
| Exchange desk component gravity $G_7$ | N | 26 000 | |
| cutting force $F_C$ | N | 10 000 | Max cutting resistance |

According to the corresponding load distribution of lathe bed in Table 1, the upright column assembly gravity $G_2$ (column center), the spindle assembly gravity $G_4$, the cutting force $F_C$, the gravity of workbench and workbench slide $G_5$, the maximum load of workbench $G_6$, and the exchange station gravity $G_7$ are respectively applied on the mounting surfaces of the lathe bed by Remote Force command. At the same time, gravity is applied to the lathe bed by the Standard Earth Gravity command or gravity acceleration. The final loading conditions of the lathe bed model are shown in Figure 2:

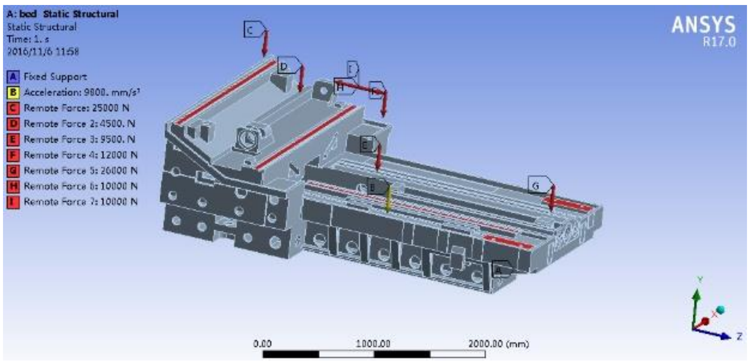



*Figure 2. Load of lathe bed*

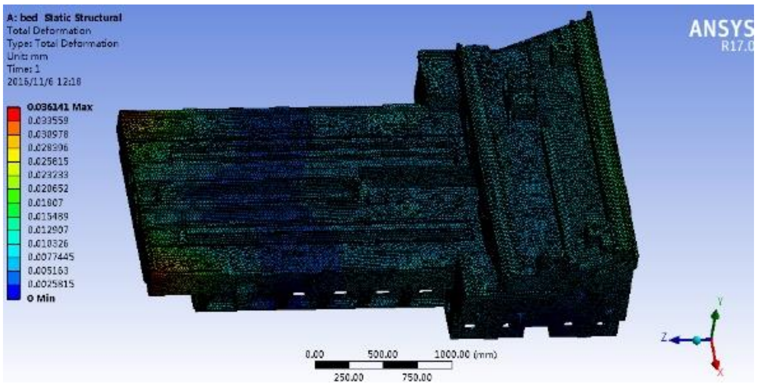


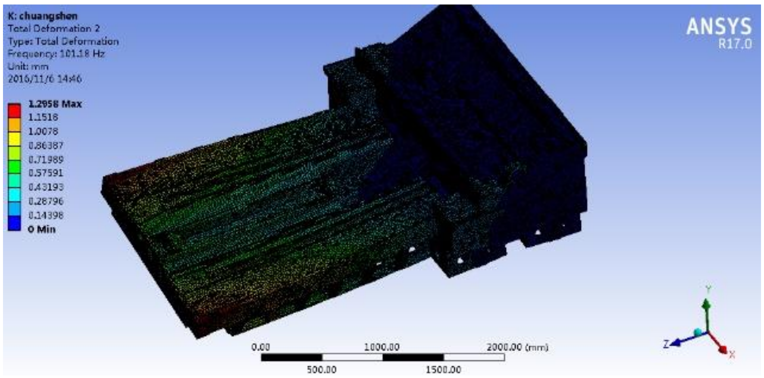

*Figure 3. Comprehensive deformation of lathe bed*

According to the Figure showing comprehensive deformation of lathe bed, it is found that the maximum deformation of lathe bed lies in the two edge angles on the front side of lathe bed (the maximum deformation is 0.0361mm), which is mainly influenced by the gravity of the exchange station and parts and the working load. The rest of the deformation mainly occurs on the rare side of the lathe bed, and the rear rail surface of the X lathe bed. In the subsequent structural optimization design, these problems shall be improved in order to obtain a better performance of the structure.

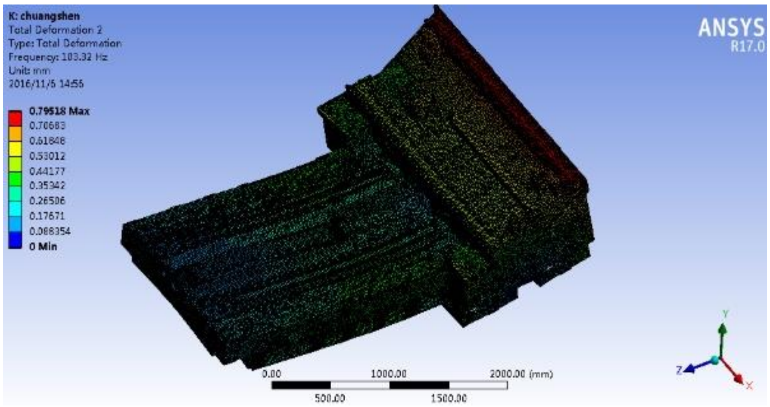
### 4. Lathe bed modal analysis

The dynamic performance of the lathe bed in the machining center reflects the vibration resistance of the structure when subject to dynamic loads, which has an important influence on the accuracy of the machine tool. Higher-order modality has a higher damping value, and their role in the vibration mode analysis is relatively small. Therefore, the general modal analysis mainly focuses on the lower-order modality with a relatively large impact on the vibration mode (Changenet *et al.*, 2006). This paper applies ANSYS Workbench to the modal analysis of lathe bed, gives the first four-order modal analysis results under full constraint, and obtains the natural frequencies of lathe bed and the corresponding vibration modes, as shown in Figure 4.

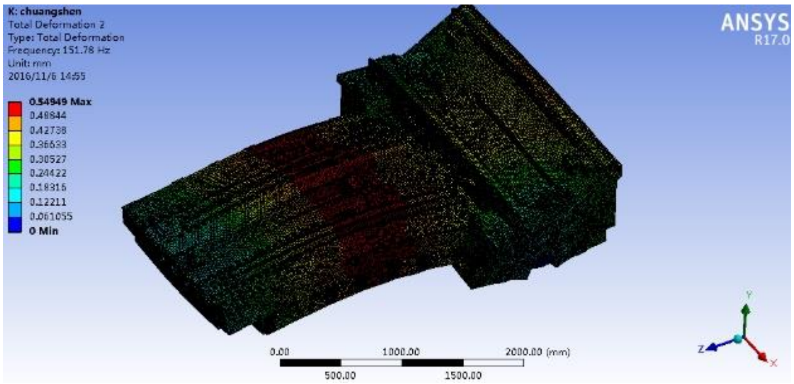
According to the Figure showing vibration modes of the first four orders of the lathe bed, it is found that outer edges in the front of the lathe bed, the middle position of the lathe bed, and the mounting surface of the outer X guide rails of the lathe bed are the weak links. These weak links may cause lack of stiffness of lathe bed, but only the bearing surface of the middle position and X-direction guide rails are the key influencing links, and the others have little impact on the accuracy change of the whole machine tool.



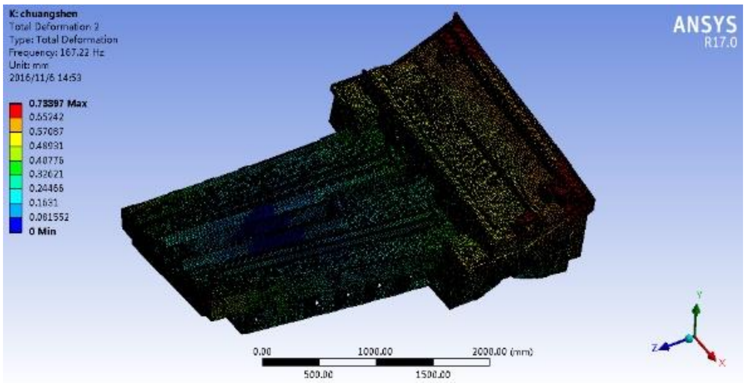

*(a) First order*




*(b) Second order*




*(c) Third order*




*(d) Fourth order*

*Figure 4. Vibration modes of lathe bed*

*Table 2. Natural frequencies and vibration modes of the first four order modality of the lathe bed*

| Modal order | Natural frequency (Hz) | Mode shape |
|---|---|---|
| First order | 101.18 | The bed is twisted left and right, the front side is the largest |
| Second order | 103.32 | The middle of the bed is concave and the rear side has the largest |
| Third order | 151.78 | The middle of the bed is raised, the middle amplitude is the largest |
| Fourth order | 167.22 | The bed is twisted left and right, with the largest amplitude on the back side |

## 5. Structural optimization of lathe bed

The original machining center lathe bed adopts three-point support, the advantage is that it is convenient to debug and install the machine tool. However, in that process of trial production and running, the stiffness of three-point support is found to be insufficient. In addition, according to the static and dynamic analysis results of the lathe bed, the deformation of the bearing surface of the rear guide rail in the x direction of lathe bed is relatively large, and the stress is also concentrated. After many times of repeated design, the optimization scheme has drawn up as follows: In both sides and the rare side of the bottom of the lathe bed, two additional foundation holes are added, and the bearing surface of rear guide rail the of X-axis is thickened by 30mm. The specific three-dimensional structure is as shown in Figure 5.

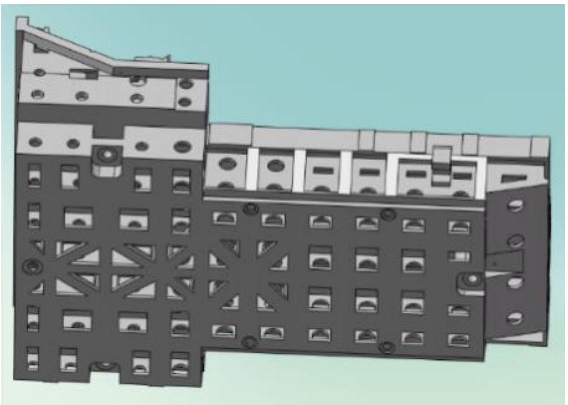


*Figure 5. Optimized structure of lathe bed*

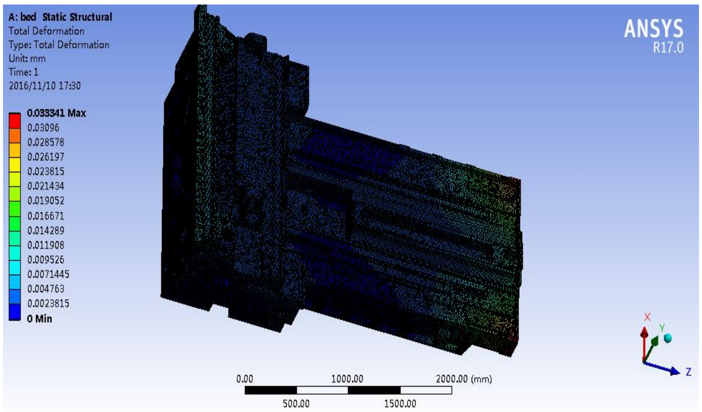
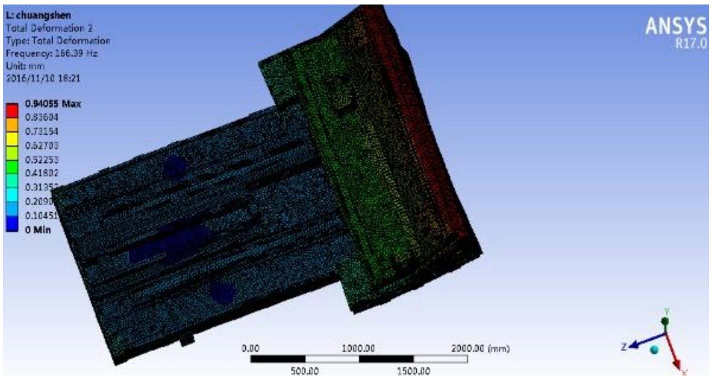
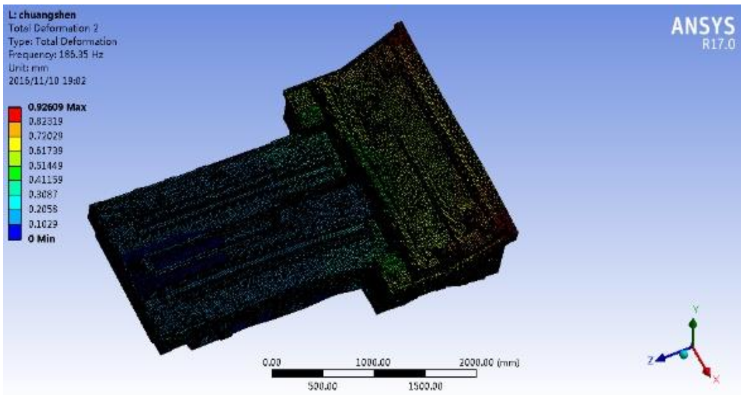



*Figure 6. Comprehensive deformation of lathe bed of setting machine*

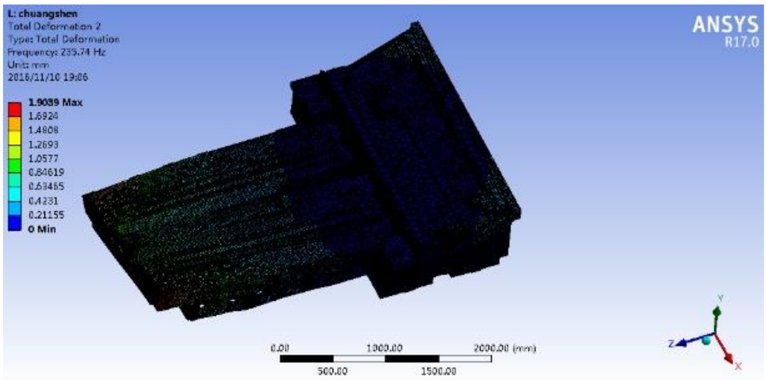
According to Figures 6 and 7, it can be seen that the maximum deformation position and stress concentration point of the lathe bed of the setting machine are consistent with those of the trial machine, and its maximum deformation amount and the maximum stress value are better than those of lathe bed in the trial machine scheme. In addition, the natural frequency of the first four orders is remarkably increased after the lathe bed is changed for the multi-point support. It is proved theoretically that the multi-point support is superior to the three-point support, so the optimization direction of this paper is accurate and the optimization scheme is feasible. Table 3 shows the comparison of performance of the lathe bed of the horizontal machining center before and after structural optimization.



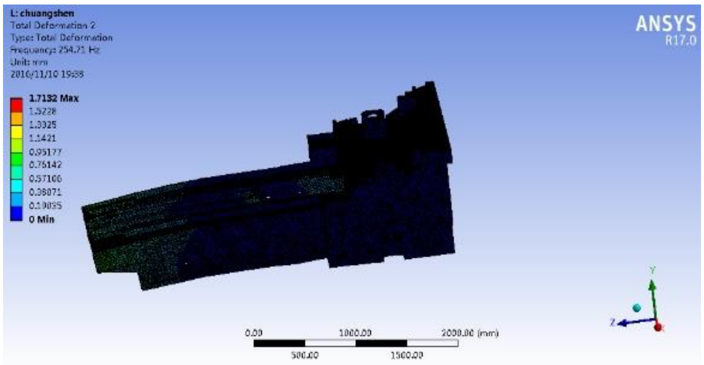

*(a) First order*




*(b) Second order*




*(c) Third order*




*(d) Fourth order*

*Figure 7. Modality vibration modes of the first four orders of the setting machine lathe bed*

*Table 3. Comparison of performances of lathe bed before and after structural optimization*

| Project | Trial machine bed | Shaped machine bed | change rate |
|---|---|---|---|
| Max deformation (mm) | 0.036 | 0.033 | -8.3% |
| Max stress (MPa) | 71.55 | 71.54 | -0.13% |
| First order (Hz) | 101.18 | 166.39 | +64.4% |
| Second order (Hz) | 103.32 | 186.35 | +80.3% |
| Third order (Hz) | 151.78 | 235.74 | +55.3% |
| Fourth order (Hz) | 167.22 | 254.71 | +52.3% |

## 6. Precision test of precision machining test pieces

In order to verify the results of the above-mentioned theoretical analysis, the precision test of the test pieces of the same specification is performed using the trial machine and the setting machine respectively. The test object is the contour precision machining. There are 3 contour test pieces for the trial machine and 3 for the setting machine, which meet the machining and testing modes. The precision test value takes the arithmetic mean of the test results.

The contour test pieces are as shown in Figure 8.

Tool: φ32mm carbide end mill

Cutting parameters are shown in Table 4.

*Table 4. Cutting parameters of contour machining*

| Project | Trial machine scheme | Setting machine scheme | Remark |
|---|---|---|---|
| cutting speed (mm/min) | 50 | 50 | |
| Feed rate (mm/t) | 0.1 | 0.1 | |
| Cutting depth (mm) | 0.2 | 0.2 | |

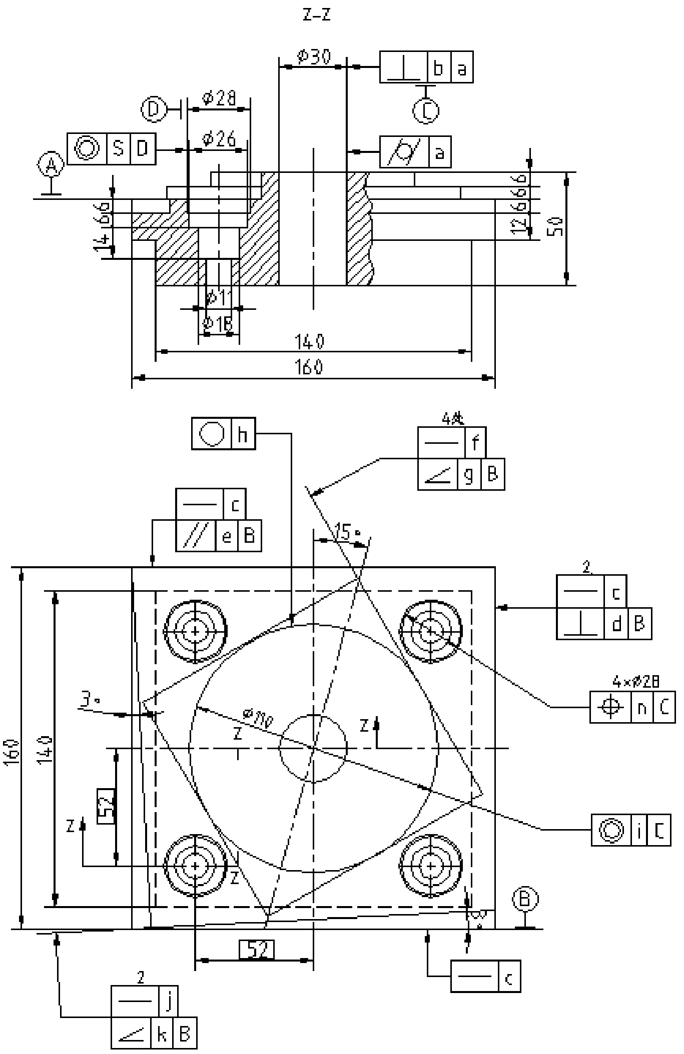



*Figure 8. Contour test pieces*

Test results are shown in Table 5.

*Table 5. Statistics of test results for contour machining*

| Test items (Precision level) | | Allowable error | Trial machine | Setting machine | conclusion |
|---|---|---|---|---|---|
| Center hole | Cylindricity | 0.007 | 0.006 | 0.006 | identical |
| | The center axis of the hole and the perpendicularity of the base surface A | ϕ0.007 | ϕ0.005 | ϕ0.006 | excelled |
| square | Straightness of the side | 0.007 | 0.006 | 0.006 | identical |
| | The perpendicularity between adjacent faces and face B | 0.007 | 0.007 | 0.005 | excelled |
| | Parallelism relative to face B | 0.007 | 0.006 | 0.005 | excelled |
| diamond | Straightness of the side | 0.007 | 0.007 | 0.005 | excelled |
| | The inclination of the side to the base B | 0.007 | 0.007 | 0.005 | excelled |
| circular | Roundness | 0.012 | 0.010 | 0.008 | excelled |
| | Concentricity of outer circle and round hole C | ϕ0.016 | ϕ0.014 | ϕ0.010 | excelled |
| Inclined plane | Straightness of the face | 0.007 | 0.005 | 0.005 | identical |
| | The inclination of the 3° angle to the B surface | 0.007 | 0.007 | 0.004 | excelled |
| Boring | Position relative to the inner hole C | ϕ0.03 | ϕ0.025 | ϕ0.018 | excelled |
| | Concentricity of inner hole and outer hole D | ϕ0.013 | ϕ0.010 | ϕ0.008 | excelled |

It can be seen from Table 5 that after the lathe bed structure is optimized, the precision of the contour test piece of the machining center of the setting machine is almost better than that of the trial machine. It shows that the optimization scheme of lathe bed is correct and the method is feasible, which achieves the unity of the theory and practice.

## 7. Conclusions

(1) This study analyzes the static and dynamic characteristics of the lathe bed of the trial machining center. Through the research on the vibration mode of each

frequency, the deformation of the trial lathe bed in different frequency vibration mode is analyzed, and the weak links of structural stiffness are pointed out.

(2) This study optimizes the lathe bed structure. The comparison of mechanical properties before and after the structural optimization of the lathe bed shows that the maximum deformation of lathe bed after optimization is reduced by 8.3%, and the maximum stress is reduced by 0.13%, thus the optimization goal has been achieved and a more reasonable lathe bed structure has been determined, which has laid the foundation and provided a certain direction for the optimization of the whole machine of the horizontal machining center.

(3) Through machining experiments on contour test pieces of the same specification using trial machine and the setting machine respectively, it is shown that the optimization scheme for the lathe bed mechanism is feasible, thus the research has certain practical significance.

*Acknowledgements*

*The project of Shaanxi Provincial Education Department (16JK1051). The key project of Baoji University of Atrs & Science (ZK2017015), (ZK15030).*

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