Structural Design Optimization of the Top Beam of a Propeller Processing Machine Tool Using FEA Analysis

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Abstract. Aiming to reduce the weight under its rigidity and strength restrictions of a propeller processing machine tool, structural design optimization of the top beam of the machine tool is carried out using FEA analysis approach. Based on static and modal analysis, both the thickness and the width of the rib are optimized and light-weighted, followed FEA analysis results demonstrate that although the maximum stress and deformation of the optimized structure go up slightly, but are still allowable. After optimization, the weight of the top beam is reduced by approximately 8.03% and the dynamic characteristics of the top beam are also enhanced, which is beneficial to the noise reduction of the machine tool.

1. Introduction
Nowadays, the design of the key structure of machine tools in our country more adopts the method of combining traditional materials and design experience, which leads to the high cost of machine tool design[1]. The structural parts of the machine tool bear the cutting load and also affect the machining accuracy of the machine tool[2]. The structural parts of the machine tool can be divided into moving parts and supporting parts. The lightweight design of the supporting parts can save materials and evenly distribute the load of the machine tool[3-5]. Therefore, the lightweight design of machine tools has become a hot topic in the field of machine tool manufacturing today, while ensuring the rigidity, strength, dynamic and static performance and other design requirements.

The top beam of the propeller processing machine tool is the load-bearing part of the machine tool, and plays the role of supporting key components such as the spindle and the tool holders[6]. The dynamic and static characteristics of the top beam also have a significant impact on the overall performance of the whole machine. Improper design may cause the excitation frequency to coincide with the inherent frequency of the top beam, resulting in resonance, which directly affects the machining accuracy and operational stability of the propeller processing machine tool[7-9]. In this paper, finite element simulation is used as design technical support to carry out static analysis and modal analysis on the top beam of the propeller processing machine tool, select an optimized design scheme with better comprehensive performance, and then carry out harmonic response analysis on the optimized design scheme to obtain the deformation frequency response curve under the action of the excitation force, so as to identify the weakest link of the top beam of the propeller processing machine tool. It not only improves the static mechanical properties of the top beam structure, but also improves its own structural vibration resistance and saves materials.
2. FEA modeling of the machine tool

2.1. Three-dimensional Model
Using Solidworks software to carry out three-dimensional modeling of the gantry CNC machine tool. In order to increase the calculation speed, the model needs to be simplified. The specific steps are as follows: Since the fine structure will not have a large impact on the quality and stiffness of the machine tool, so rounded corners and bolts in various places can be ignored when modeling. In order to facilitate the subsequent work of the finite element analysis, the parts that have little impact on the performance of the whole machine are removed, and the bed, X-axis trailing plate, columns, crossbeam, top beam, counterweight fixed pulley, cross brace, Y-axis trailing plate and spindle system are retained; the simplified three-dimensional model of the propeller processing machine tool is shown in Figure 1 and the top beam model is shown in Figure 2.

2.2. FEA Model
Importing the top beam model into Ansys Workbench software to establish the finite element model. The structure of the top beam of the QLM2365/5X propeller processing machine tool is relatively regular. There is a rib structure inside. The material of the top beam is Q235, its elastic modulus is 2.1×1011pa, the Poisson's ratio is 0.33, the density is 7850kg/m3, and the weight of the top beam is 1560.44kg. The parameters of the finite element model were set as shown in Table 1. The bed is meshed, and the finite element model of the bed after meshing is shown in Figure 3.

1. Bed 2. X-axis trailing plate 3. Column 4. Crossbeam 5. Counterweight fixed pulley 6. Top beam 7. Cross brace 8. Y-axis trailing plate 9. Spindle system

Figure 1. The structure of the propeller processing machine tool

Figure 2. Simplified top beam model.

Table 1. Material parameters of top beam.

| Material | Weight  | Density     | Elastic modulus | Poisson's ratio |
|----------|---------|-------------|-----------------|-----------------|
| Q235     | 1560.44kg | 7850kg/m³   | 2.1×1011pa      | 0.33            |
3. FEA analysis under static load

The static analysis of the top beam is to calculate the deformation and stress of the top beam under a fixed constant load when it is constrained. It has been determined that the material of the top beam is Q235, which has good vibration resistance and resistance to deformation. When formulating the structural design scheme of the top beam of the propeller processing machine tool, it is necessary to comprehensively consider all the performance indicators of the top beam to obtain the optimal structure. The optimal criteria are: the maximum stress, the maximum deformation, the smaller the weight of the cross beam, and the larger the first four orders of the intrinsic frequency, the better.

The restraint form of the top beam of the propeller processing machine tool is that the two ends of the top beam and the connection part of the screw on the machine tool column are fixedly restrained to achieve the purpose of complete fixation. The main force of the top beam is the weight of the top beam in the Z direction of 15292.31N, the milling force of the X, Y, and Z axes is 40,000N, and the weight of the spindle system and the crossbeam is 118216N. Through the above loading constraints, reload calculations in Ansys Workbench, the stress and deformation of the top beam are shown in Figure 4 and Figure 5.

It can be seen from Figure 4 that the stress value of most areas of the top beam is between 0~3.1943Mpa, which is within the maximum stress range of the bed material, indicating that the density of the mesh division is reasonable and the accuracy of the calculation is high. The maximum value of local stress reaches 3.1943Mpa, which is due to the existence of a sharp corner structure at the connection between the top beam and the column, which leads to a local stress concentration and does not have a large impact on the overall analysis of the top beam.

It can be concluded from Figure 5 that after the rigidity analysis and calculation of the top beam, the deformation distribution cloud diagram of the top beam is obtained. The overall deformation position of the top beam appears at the middle position of the top beam, and the maximum value is 0.06378mm, which indicates that the structural deformation of the bed is very small and can meet the requirements of machining accuracy.
4. Model analysis
During the machining process, the top beam and other parts of the propeller processing machine tool will vibrate to a certain extent due to various forces. When the vibration frequency is the same as the inherent frequency of the top beam itself, resonance will occur[10-12]. As a result, the machining accuracy of the machine tool is greatly reduced due to the large deformation of the top beam. The vibration of the top beam structure can be represented by the linear combination of the intrinsic vibration modes of each order, where the lower order vibration modes have a greater degree of dynamic influence on the structure than the higher order ones, so only the first six orders of the top beam lower order modes are studied. The results of the first six orders of the top beam modal analysis are shown in Table 2, and the first six orders of the vibration diagram are shown in Figure 6.

| Orders | Inherent frequency /Hz | Vibration type | Maximum deformation /mm |
|--------|------------------------|----------------|-------------------------|
| 1      | 57.71                  | Downward bending vibration around X-axis | 0.5423 |
| 2      | 125.32                 | S-shaped bending vibration around X-axis | 0.5369 |
| 3      | 135.54                 | Bending vibration around Y-axis | 0.6118 |
| 4      | 144.87                 | Torsional vibration around X-axis | 0.6110 |
| 5      | 198.46                 | Vibration around X wave type | 0.6275 |
| 6      | 210.3                  | Torsional vibration around X-axis | 0.7326 |

Figure 6. The first six order vibration diagram of the top beam.
It is not difficult to see from Table 2 that the deformation displacement of the first six order vibration process is between 0.5397~0.7068mm, and the fifth order resonance displacement reaches the maximum value of 0.7068mm. For the precision machining process, this value is too large. In order to reduce the machining error, it is necessary to avoid the fifth order inherent frequency in the machining process. The first order inherent frequency is the smallest, at 57.71Hz, and according to the relationship between machine speed and frequency:

\[ n = 60f \]  

(1)

Where \( f \) is the inherent frequency of the machine tool, the critical speed of the first order is 3443.4r/min, so the possibility of resonance is very small, indicating that in the process of processing, the top beam of this type of propeller processing machine tool meets the requirements of the product when processing.

5. Structural design optimization

The rib structure of the top beam plays a role in strengthening the bed and reducing the weight of the top beam itself, and it is easy to see that the top beam has good rigidity and static strength in the static analysis[13]. However, the inherent frequency of the first order is slightly lower than the other stages, so this paper takes the width and thickness of the rib of the top beam as the object of study to optimize the design. The thickness of the rib is based on the depth of the upper and lower grooves of the rib, so the increase in the width of the rib brings about a much lower weight increase than the reduction in the overall weight of the top beam brought about by the increase in the thickness, which is used as a lightweight design scheme. As an important mechanical processing equipment, the role of the top beam of the propeller processing machine tool is also particularly important. First of all, the original shape and size of the top beam itself cannot be changed, and secondly, the strict assembly relationship between the top beam and the column and its structure must be followed[14-16]. The optimization design goal of this paper is set to make the top beam lightweight design under the premise of ensuring the inherent frequency, static strength and rigidity of the bed basically unchanged.

In the finite element analysis process, the thickness and width of the ribs are set as design variables. In the process of optimizing the structure of the top beam of the propeller processing machine tool, it is necessary to take into account the satisfaction of casting and material manufacturability, such as the thickness of the rib cannot exceed the thickness of the top beam itself, and the top beam itself should be as uniform as possible. To meet the above requirements, the steps of the optimized design scheme in this paper are as follows.

Step 1: The width of the top beam rib of the propeller processing machine tool is used as a design variable. The original size of the rib is 150mm. The optimization plan is to increase the width of the rib to 200mm in 5 steps on the original basis, each time increasing by 10mm. As the width of the rib increases, the width of the rectangular groove above the rib is also reduced to 450mm in 5 steps. It can be seen from Figure 7 that as the width of the rib increases, the first order frequency also shows a downward trend, but the decrease is only 0.038Hz, so changing the width of the rib has little effect on the frequency of the rib. It can be seen that when the width of the rib is 160mm-190mm, the frequency drops relatively slowly, so four options of the width of the rib are 160mm, 170mm, 180mm and 190mm.

| Width of Rib /mm | First Order Frequencies /Hz |
|-----------------|-----------------------------|
| 160             | 57.736                      |
| 170             | 57.728                      |
| 180             | 57.718                      |
| 190             | 57.711                      |
| 200             | 57.698                      |
Figure 7. The relationship between the width of the rib and the first order intrinsic frequency.

Step 2: Set the thickness of the top beam as a variable. As shown in the model shown in Figure 2, the thickness of the top beam is intended to be set relative to the depth of the upper and lower grooves of the top beam, that is, the groove depth increases as the thickness of the rib increases. The thickness of the rib is increased by 5 mm each time, and the total increase is 5 times. It can be seen from Figure 8 that the change in the thickness of the ribs has a small effect on the first order intrinsic frequency of the top beam. With the increase of the thickness of the rib, the decrease of the frequency is only 2.064 Hz, and the drops of the last four times are small. Therefore, in the optimization design process, four options for the thickness of the ribs are 30mm, 35mm, 40mm and 45mm.

Table 4. Thirst order frequencies of the rib thickness.

| Thickness of Rib /mm | First Order Frequencies /Hz |
|----------------------|-----------------------------|
| 25                   | 58.219                      |
| 30                   | 57.415                      |
| 35                   | 57.133                      |
| 40                   | 56.722                      |
| 45                   | 56.155                      |

Figure 8. The relationship between the thickness of the rib and the first order intrinsic.

Step 3: Combining the four options for increasing the width of the ribs and the four options for increasing the thickness to obtain a total of 16 first order intrinsic frequencies of each combined top beam, as shown in Table 5.

Table 5. The inherent frequencies of 16 combinations.
The nine sets of data in Table 5 were selected as the nine available options, as shown in Table 6.

Table 6. Nine optimization schemes.

| Width of Rib /mm | Thickness of Rib 30mm | Thickness of Rib 35mm | Thickness of Rib 40mm |
|------------------|-----------------------|-----------------------|-----------------------|
| 160              | 57.401                | 57.115                | 56.712                |
| 170              | 57.39                 | 57.13                 | 56.691                |
| 180              | 57.375                | 57.099                | 56.701                |
| 190              | 57.366                | 57.098                | 56.705                |

The nine sets of data in Table 5 were selected as the nine available options, as shown in Table 6.

Table 7. Comparison of parameters of 12 optimized design schemes.

| Optimized design schemes | Top beam weight /kg | Weight change rate % | First order intrinsic frequency /Hz | Rate of change of the first order intrinsic frequency % |
|--------------------------|---------------------|----------------------|-------------------------------------|-----------------------------------------------------|
| Scheme 1                 | 1498.43             | -3.97                | 57.401                              | -0.535                                               |
| Scheme 2                 | 1500.45             | -3.84                | 57.39                               | -0.554                                               |
| Scheme 3                 | 1502.47             | -3.71                | 57.375                              | -0.58                                                |
| Scheme 4                 | 1505.5              | -3.52                | 57.366                              | -0.596                                               |
| Scheme 5                 | 1466.75             | -6.00                | 57.115                              | -1.031                                               |
| Scheme 6                 | 1469.11             | -5.85                | 57.13                               | -1.005                                               |
| Scheme 7                 | 1471.47             | -5.70                | 57.14                               | -1.06                                                |
| Scheme 8                 | 1473.83             | -5.55                | 57.098                              | -1.71                                                |
| Scheme 9                 | 1435.08             | -8.03                | 56.712                              | -1.729                                               |
| Scheme 10                | 1437.77             | -7.86                | 56.691                              | -1.702                                               |
| Scheme 11                | 1440.47             | -7.69                | 56.701                              | -1.748                                               |
| Scheme 12                | 1443.16             | -7.52                | 56.705                              | -1.741                                               |

It can be seen from Table 7 that the weight reduction of the top beam in Scheme 9 is the largest, which is about 8.03%, while the first order intrinsic frequency drops by about 1.729%, both of which are within the allowed variation range, and the optimized design scheme is reasonable. Further analyze the optimization scheme of Scheme 9, import the three-dimensional model of the top beam after the optimization design into ANSYS Workbench software, and perform static analysis on it. The maximum stress of the optimized top beam itself is 7.287Mpa, as shown in Figure 9, which is close to the maximum stress of 3.194Mpa before optimization. The maximum deformation value of the top beam before optimization is 0.063782 mm and after optimization is 0.12997 mm. Although the stress and deformation before and after optimization are slightly increased, the maximum values are all of the same order of magnitude, which are within the range of the top beam's own material requirements, which indicates that the design of Scheme 9 is feasible and indirectly verifies that the optimized design scheme is feasible and effective.
6. Conclusion

Based on the concept of lightweight design, this paper proposes a new optimal design scheme for the top beam. Taking the width and thickness of the top beam rib as the design entry point, the top beam itself weight is reduced by changing the width and thickness of the top beam rib under the premise of meeting the top beam stiffness and strength limitation requirements. The optimized top beam model was subjected to modal analysis, and the optimal design scheme was selected among 12 optimized design schemes by using the rate of change of the intrinsic frequency of the first section as the reference basis and the rate of weight change. The optimized top beam model was subjected to FEA analysis, and the stress and total deformation clouds were compared with the results before the optimization. It is concluded that the top beam of the propeller processing machine can be designed to be lightweight by changing the width and thickness of the rib, provided that the requirements of the static foundation and the material casting process are met. The optimum size of the top beam rib is also given in this paper, which provides a basis for the subsequent research.

7. References

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