Rational design of the column of a heavy multipurpose machining center

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Abstract. The main purpose in the design of supporting constructions of heavy multipurpose machining center is the reduction of mass at the given precision and productivity of machining. Accomplish these ends the technology of rational design of supporting constructions is offered. This technology is based on the decomposition method and the finite elements method in the combination with optimization methods. The technology has four stages: 1) calculation of external forces and loads, 2) as a result the boundary conditions (force, kinematics) for individual supporting constructions are formed, 3) a problem about final optimal distribution of a material by the individual supporting constructions with the real cross-section is solved; 4) dynamic analysis. By the example of design of the column of a heavy multipurpose machining center the main stages of rational design of the individual supporting constructions are shown. At a design stage of the carrying system consisting of load-bearing structures with simplified geometry, optimum overall dimensions of the column are identified. For the admitted system of preferences, it is necessary to accept the fact that the carrying system with the column with the sizes of cross section of 1.8 m (along \(x\) axis) and 2.6 m (along \(y\) axis) is the best. The analysis of the work of the column under the torsion condition with the use of method of mechanics shows that the column with square cross sections = 2.46·2.46 m which rigidity on torsion is 26 % higher in comparison with a production version is the best. The results of calculation show that a production-release design of the column with longitudinal and transverse edges of rigidity is 24 % heavier than the column with the edges located on a diagonally at equal rigidity.

1. Introduction

For the past decades the finite elements method has been widely used in the machine-tool industry for the static and dynamic analysis of supporting constructions of task geometry \[1, 2\]. Good results in the technology of rational design of supporting constructions of machine tools can be achieved when using the finite elements method together with the optimization methods \[3–5\]. The description and practical application of the widely used optimization methods are presented in \[6, 7\].

The overall dimensions of a metal-cutting machine tool are specified at the stage of project requirements. Then on the basis of calculations of strength and rigidity a designer determines sizes of supporting constructions that should meet the requirements for accuracy and productivity of machining. While analyzing configuration arrangements for individual supporting constructions general methods of solids mechanics, for example the method of strength of materials \[8\], are also effectively used.
In research paper [9] the technology of the design of supporting constructions of heavy multipurpose machining centers (Figure 1) is presented, it allows to design machines without excess capabilities, to predict their performance under the conditions of intense mechanical forces and to give scientific grounding to the choice of design options. The technology has four main stages:

1) determining external forces using deterministic or stochastic models;
2) calculating the set configuration of the carrying system of a machine taking into account the required productivity and accuracy of machining; supporting constructions here have simplified cross section geometry. As a result the boundary conditions (force, kinematics) for an individual supporting construction are formed, which makes it possible to analyse supporting constructions independently of one another;
3) the result of the stage of modeling an individual supporting construction is an optimal construction with real cross section geometry and with the minimum possible mass, provided the boundary conditions formed at the previous stage are satisfied;
4) dynamic analyzing or simulation modelling average operating conditions of the carrying system with optimal supporting constructions.

Figure 1. The Design Technology of Machining Centers

The purpose of this paper is to calculate and analyse configuration arrangements for the most loaded supporting construction – the column of a heavy multipurpose machining center of a milling, drilling and boring group (Figure 2). To achieve the purpose the finite elements method, the optimization methods and the methods of strength of materials are used. A multipurpose machining center is a complex consisting of two items that are not put together with assembly operations at a manufacturing plant but designed to perform interrelated operating functions – a special milling-and-
boring machine (Figure 2, Items 1-3) and a slewing-sliding table (Figure 2, Items 5-7). This fact makes it possible to do calculations for a table and a milling-and-boring machine separately.

2. Results and Discussion

a. Analysis of the response of supporting constructions as part of a carrying system of a milling-and-boring machine. The supporting constructions of this machine have a complicated configuration. These are space thin-walled constructions, which have differently directed stiffening ribs, a multiply connected closed outline, differences in wall thickness, and so on, that makes it hard to analyse directly their response to external actions as part of a carrying system.

To make a comparative evaluation of the choice of external overall dimensions of the constructions their simplified cross-section geometry is presented. So, in particular, unlike the regular (production-release) construction (Figure 3), the column will have rectangular box-type cross-section without stiffening ribs and simply connected closed profile.

Design conditions for carrying systems of machines are chosen based on the analysis of operating experience of machines that are similar in terms of configuration. Yet, the operations should be identified where accuracy and productivity are close to the limiting values. Relating to drilling, milling and boring machines the operation of this kind is face milling. A number of values for calculation are chosen as reference ones according to the project requirements for machine designing. In particular, the machine should provide the maximum forward force of 40 kN, for fine milling – 3 kN. The following design conditions are accepted for the carrying system of a machine:

- the spindle head is in the upper limit position provided the column is in the centre of the bed, extensions of the spindle ($\leq 0.4$ m) and the slide ($\leq 0.6$ m) correspond to the limiting values;
- external loading for a machine is cutting forces applied at point $O$ on the spindle axis (Figure 2). The ratio for the components of cutting forces is $P_x : P_y : P_z = 0.5 : 1.0 : 0.7$;
- the bed rests on the resilient bearings (64 bearings), the rigidity of which is calculated as the load of 40 kN, and the vertical displacement is $2 \times 10^{-5}$ m;
- contact deformations of moving joints is taken into account: column – spindle head, spindle head – slide, column – bed. The calculation of point $O$ displacements (Figure 2) in the direction of $x$, $y$, and $z$ axes are done for finish machining based on the rough surface rod model. As a result only due to contact deformations at the joints point $O$ displacements of the spindle along $x$, $y$, and $z$ axes are $\delta_x = 13.89 \times 10^{-6}$ m; $\delta_y = 8.11 \times 10^{-6}$ m; $\delta_z = 5.45 \times 10^{-6}$ m, respectively.

The mathematical model of the carrying system of a milling-and-boring machine is formulated as a mathematical programming problem:

\[
 f(X) = \sum_{i=1}^{n} \rho_i V_i
\]

under constraints:

upon stress

\[
 g_i = 1 - \frac{\sigma_{eqv}}{\sigma_{allow}} \geq 0,
\]

upon point $O$ displacements along axes

Figure 2. Machining center: 1 – column; 2 – spindle head; 3 – bed; 4 – work piece; 5 – pallet; 6 – slewing table.

Figure 3. The production version
\[ g_2 = 1 - \Delta_x \left[ \frac{\Delta_y}{\Delta_z} \right] \geq 0, \quad (3) \]
\[ g_3 = 1 - \Delta_y \left[ \frac{\Delta_z}{\Delta_z} \right] \geq 0, \quad (4) \]
\[ g_4 = 1 - \Delta_z \left[ \frac{\Delta_x}{\Delta_x} \right] \geq 0, \quad (5) \]
\[ g_5 = V_i \geq 0, \quad i = 1, 2, \ldots, n, \quad (6) \]

where \( n \) is a number of plate-type finite elements; \( \rho \) is material density; \( V \) is the volume of the finite element; \( \sigma_{eqv}, \sigma_{allow} \) – equivalent stress and allowable stress; \( \Delta_x, \Delta_y, \Delta_z, [\Delta_x], [\Delta_y], [\Delta_z] \) – design and allowable (in brackets) point displacements along \( x, y, z \) axes, respectively. The target function (1) is the construction mass. The design variable is thickness of the profile wall of supporting constructions.

When identifying allowable displacements \([\Delta_x], [\Delta_y], [\Delta_z]\) of the carrying system in the machining medium the departures from flatness and straightness on the machined surface are considered. In accordance with the accepted standards of accuracy when milling the tolerances for flatness and straightness are:

- on vertical displacement of the spindle head (5 m) – \( 60 \cdot 10^{-6} \) m;
- for the transverse displacement of the column (it is 4.5 m to the centre of the bed) – \( 60 \cdot 10^{-6} \) m.

According to the data obtained from the manufacturing plant the machining error at a cutting zone is distributed among a milling-and-boring machine and a table as 70 % and 30 %, respectively. When doing calculations it is impossible to consider all the factors that affect the machining accuracy, so, the safety factor for rigidity is taken as 1.5. Finally the tolerances for flatness and straightness for a milling-and-boring machine are \( 28 \cdot 10^{-6} \) m.

Point \( O \) displacements at a cutting zone are determined by the sum of displacements as a result of deformations of the carrying system itself and contact deformations at the joints. To calculate the carrying system point \( O \) displacements obtained as a result of contact deformations are subtracted from the obtained tolerance for point \( O \) displacements (28 \( \cdot 10^{-6} \) m). The total is:

\[ [\Delta_x] = 14.11 \cdot 10^{-6} \text{ m}, [\Delta_y] = 19.89 \cdot 10^{-6} \text{ m}, [\Delta_z] = 22.55 \cdot 10^{-6} \text{ m}. \]

While optimizing the carrying system of a machine it is believed that the displacements in the cutting zone along \( x, y, z \) axes must be less than or equal the corresponding allowable displacements. For the admitted system of preferences the option when the mass is the least and the displacements equal or near the allowable values is considered to be effective.

The problem of constrained optimization (1) – (6) is solved with the penalty function method as (\( r \) – penalty parameter)

\[ \phi(X, r) = f(X) + r \sum_{j=1}^{k} \left[ 1 / g_j(X) \right], \quad (7) \]

but using Davidon-Fletcher-Powell method to solve the problem of unconstrained optimization (7) [6, 7].

Figure 4 shows the deformed state of the carrying system of a milling-and-boring machine. To solve the problem with the finite elements method a plate-type four-node finite element is used. It can be seen that the column suffers the combined effect of bending and torsion and is the most loaded element of the carrying system of a machine. The data obtained while studying the carrying system utilizing supporting constructions of different type and for various operating conditions compared to the production versions are presented in Table 1. The average operating conditions: finish machining, face milling cutter 250 mm in diameter, the number of teeth is 20, the depth of cut is 0.5 mm, feed per tooth is 0.05 mm, spindle rotation frequency is \( 170 \text{ min}^{-1} \).
The calculation results show that it is possible to improve the configuration of the supporting constructions compared to their production versions. For the admitted system of preferences it should be recognized that the carrying system with the column with the sizes of cross section of 1.8 m (along \( x \) axis) and 2.6 m (along \( y \) axis) is the best. The carrying system with the column of \( 1.6 \times 2.8 \) (m) is rejected due to the great extent of the column section along \( y \) axis, as it brings to further decrease in the working stroke of the column along this axis. It should be stated that in the course of optimization point \( O \) displacement along \( y \) axis is an active constraint. If to take the average operating conditions as a basis while designing a machine, then when calculating the production version it is clear that the advantage by mass is about 30 %, with the standards of machining accuracy to be retained.

### Table 1. Calculation Results for the Carrying System

| Parameter                        | Operating Conditions | Limiting Production | Calculation Production |
|----------------------------------|----------------------|---------------------|------------------------|
| Cross-section sizes (along axes), m: |                      |                     |                        |
| – column \((x–y)\)               | 2.0\( \times \)2.46  | 2.3\( \times \)2.3  | 2.46\( \times \)2.46   | 1.8\( \times \)2.6  | 1.6\( \times \)2.8  | 2.0\( \times \)2.46  |
| – spindle head \((z–x)\)         | 1.3\( \times \)2.20  | 1.8\( \times \)1.9  | 1.30\( \times \)2.20   | 1.7\( \times \)2.0  | 1.7\( \times \)2.0  | 1.3\( \times \)2.20  |
| – slide \((z–y)\)                 | 0.6\( \times \)0.80  | 0.7\( \times \)0.7  | 0.80\( \times \)0.80   | 0.8\( \times \)0.8  | 0.8\( \times \)0.8  | 0.6\( \times \)0.80  |
| Point \( O \) displacements, micron: |                      |                     |                        |
| – \( x \) axis (norm 14.11)      | 7.34                 | 6.75                | 6.43                   | 9.51                 | 9.58                 | 6.28                   |
| – \( y \) axis (norm 19.89)      | 19.89                | 19.89               | 19.89                  | 19.89                | 19.89                | 19.89                  |
| – \( z \) axis (norm 22.55)      | 11.58                | 11.74               | 11.56                  | 12.10                | 12.69                | 11.20                  |
| Wall thickness, mm:               |                      |                     |                        |
| – \( yz, xz \) plane             | 98                   | 94                  | 96                     | 99                   | 98                   | 71                      |
| – \( xy \) plane                 | 146                  | 162                 | 147                    | 107                  | 106.7                | 99                      |
| Mass, tons                       | 169.9                | 179.7               | 181.0                  | 158.2                | 157.2                | 119.3                   |
b. Analysis of the configuration arrangements for the column

The production column (Figure 3) is a spatial thin-walled construction that consists of two parts connected with a bolted joint. The front of the column, which has guides for a spindle head, consists of two closed boundaries. The inner chamber of the back of the column has longitudinal and transverse edges crossing throughout the height and it is used to position a counterweight (it unloads vertical travel actuator of a spindle head from its weight).

1. To test the multiply connected contour of the production column for torsional rigidity simplified representation of the column is presented (Figure 5) and Prandtl analogy is applied [8]. In the case of \( n \)-connected cross-section the torsional moment is calculated from formula

\[
M = 2 \left( \int \int \phi \, dx \, dy + \sum_{i=1}^{n-1} \phi_i F_i - \phi_0 F_0 \right),
\]

where \( \phi \) – Prandtl function; \( F_i, F_0 \) – areas bounded by inner and outer contours. In the case being considered formula (8) becomes as:

\[
M = 2 \sum_{i=1}^{n-1} f_i w_i = 2 \sum_{i=1}^{n-1} f_i \phi_i,
\]

where \( f_i \) – area bounded by the contour, which everywhere halves the section thickness in cross-section; \( w_i \) – membrane slack surface function (\( w_i \) is identical with \( \phi_i \)). The reference angle of torsion is calculated from formula

\[
\theta = \frac{M}{(Gk)_{\ell}},
\]

where \( Gk \) – cross-section rigidity for torsion. Figure 6 shows the calculation results (9) – (10) compared to the production version (Figure 6, a), rigidity for torsion of which is taken as 100 %.

The data analysis shows that under these conditions the column with square cross section (Figure 6, d) which rigidity for torsion is 26 % higher in comparison with a production version is the best. In Fig.6, f there is the case where some improvement in rigidity also occurs. In reality the column with square cross section of \( 2 \times 2 \) m (Figure 6, e) is also produced. However, section rigidity in this case is 35 % lower than rigidity for torsion of a production version.

![Figure 5. Cross section of the column](image)

![Figure 6. Cross sections of the column](image)
2. The column can have various configurations of edges which to great extent determine its rigidity and, therefore, rigidity of a machine on the whole. A simplified construction of the column is considered to study the way a configuration of edges influences the column rigidity. Figure 7 shows the configurations of edges (there are no edges on the front wall). The configuration of edges in a production version is presented in Figure 7, a. The design model of the column simulates all kinds of deformation of a production version. Wall constraint upon displacements of the column structures for all configurations of edges is identical as the scheme of finite elements makes it possible to have a constant mesh for plate-like finite elements simulating column walls. The design data to do calculation: wall thickness is 0.04 m, edge thickness is 0.06 m, and column restraint along the lower contour is rigid.

If to denote, for instance, a turning angle of the front wall, which a spindle head slides along, by $\theta = \frac{|y_1 - y_2|}{L_{12}}$, where $y_1, y_2$ – linear displacements of point 1 and point 2, respectively, along $y$ axis, $L_{12}$ – distance between points 1 and 2 (Figure 7), $M$ – column mass, the following results are obtained:

| Configuration of Edges | $\theta, \%$ | $\theta/M, \%$ |
|------------------------|--------------|----------------|
| Figure 7, a            | 100          | 100            |
| Figure 7, b            | 70           | 70             |
| Figure 7, c            | 100.3        | 124            |

The results of calculation show that a production-release design of the column with longitudinal and transverse edges crossing throughout the height (Figure 7, a) is 24 % heavier than the column with edges crossing diagonally (Figure 7, c) at equal rigidity, and less rigid than the column with longitudinal and transverse edges crossing diagonally (Figure 7. b). Consequently, while designing a column slanted edges are far more preferable. As it is known, influence of longitudinal edges on shifts due to torsion is inappreciable, and due to bending it does not go beyond 10…20 %.

3. Conclusions
The results of calculation show that it is possible to improve configuration of supporting constructions, particularly a column, compared to a production version. For the admitted system of preferences, the
carrying system with the column with the sizes of cross section of 1.8 m (along x axis) and 2.6 m (along y axis) is stated to be the best.

Studies of the impact of the configuration of edges of rigidity show that a production-release design of the column with longitudinal and transverse edges crossing throughout the height is 24% heavier than the column with edges crossing diagonally at equal rigidity and less rigid than the column with longitudinal and transverse edges crossing diagonally. Consequently, while designing a column slanted edges are far more preferable.

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