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OPTIMISATION OF THE DESIGN OF THE BASIC ELEMENTS OF THE BEARING SYSTEM OF A HEAVY CNC MACHINING CENTRE

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Abstract

By the example of a three-dimensional model of the bearing system of a heavy multi-purpose CNC machine, the influence on the deformation displacements of the structural design of the basic elements, the location of the supports is considered, which made it possible to identify weak points in the structure and outline ways to improve the processing accuracy.

Keywords: multi-purpose machine, spindle, axis, accuracy, modeling, loading, stiffness, deformation movements, modification of structures.

ОПТИМИЗАЦИЯ КОНСТРУКЦИИ БАЗОВЫХ ЭЛЕМЕНТОВ НЕСУЩЕЙ СИСТЕМЫ ТЯЖЕЛОГО МНОГОЦЕЛЕВОГО СТАНКА С ЧПУ

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Реферат

На примере трехмерной модели несущей системы тяжелого многоцелевого станка с ЧПУ рассмотрено влияние на деформационные перемещения конструктивного оформления базовых элементов, расположения опор, что позволило выявить слабые места в конструкции и наметить пути повышения точности обработки. Расчеты выполнены с использованием метода конечных элементов.

Ключевые слова: многоцелевой станок, шпиндель, ось, точность, моделирование, нагружение, жесткость, деформационные перемещения, модификация конструкций.

Preamble

Processing accuracy is significantly influenced by deformations of basic elements of the machine bearing system which result in readjustment of mounting surfaces carrying a billet and a tool, that is, geometrical accuracy parameters. The multipurpose machine is by its design a complicated system with considerable electric power availability compared to common machines and the load and heat sharing in it is problematic to describe analytically. Deformations of machine units and parts are of a complex spatial nature and depend upon many factors [1]: structure and arrangement of machine units, machine operating modes, sequence and duration of operation in this or other mode, thermo-physical properties of materials of which the machine is made, geometric form of the machine units and many other parameters. The result of basic component displacement caused by elastic deformations of the technological system is spatial deviation of the spindle axis. This deviation results in processing error increase and in the absence of machine accuracy reserve the size got at surface processing may exceed the bounds of tolerance zone, which is a parametric failure [2]. Spindle deviation in consequence of deformation of machine major units and its entire bearing system will be a variable and primarily depends upon machine operating conditions.

If in case of light and medium machines the main influence on spindle axis displacement is equally exercised by force and thermal factors of action on the spindle unit and the surfaces it is based on [3], in case of heavy and unique machines the following matters deserve special attention: basic components' weight, gravity centre shifts of moving elements while in operation, residual voltages.

Conducting experimental researches of such deviations under different operating conditions is a long-term and labor-intensive process and in case of multipurpose machines (especially heavy machines) it produces a need for transition to computer modelling of deformation processes during diagnosis of elastic deformations of machine bearing system and to forecasting of its geometrical accuracy change. The forecasting is possible if the behavior of spindle axis time shifting is known. Instead of experimental determination of such dependence one may use an elastic model of the multipurpose machine basic component.

The use of modeling of deformation processes occurring in the processing equipment with the help of computers and special-purpose software substantially reduces the laboriousness of similar problems solution on changing the subject of research, because once produced model can be adapted to new conditions or a new subject in the shortest possible time. Modelling makes it possible to solve the problems of design optimization with the purpose of increase in stiffness and consequently in accuracy of the entire machine.

Workobjective

The objective of this work is assessment of the impact of deformations of the basic elements of the heavy multi-purpose CNC machine's bearing system on the spindle shift and assessment of a possibility of optimization of these elements' design.

Body

The research technique set forth in this work [4] was applied for modelling the assessment of displacements caused by elastic deformations. A heavy horizontal multipurpose CNC machine was considered as the subject of research. The machine is intended to processing of large parts weighing up to 50 tons. The machine structure includes a spindle head with a moving slide and horizontal bar spindle, a rack along which a spindle head moves, a sledge with a fixed rack, a frame along which the sledge with the rack move, a swivel table and a stationary table placed apart from the frame. Also the machine is equipped with an automatic tool changing apparatus (ATC apparatus). The maximum working travels of: the rack horizontally along the axis X – 8000 mm; the spindle head vertically along the axis Y – 3000 mm; the slide along the axis Z – 1500 mm. The machine overall dimensions are length x width x height – 17000 x 10800 x 6970 mm. Examination of the main part of the machine assembly (without reference to the stationary and longitudinal swivel tables) weighing 103960 kg is performed. The total height of the machine geometrical model is equal to 6500 mm. The weight of prismatic parts is shown in Figure 1.

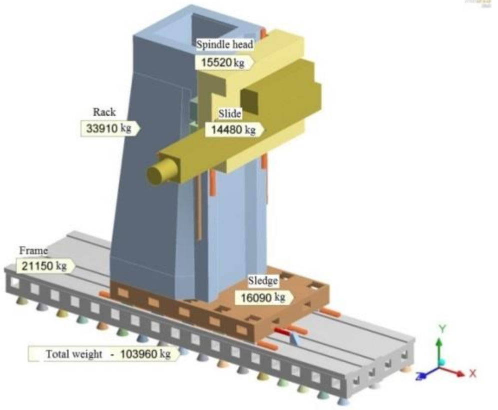

Figure 1 – Weight of prismatic parts of the machine

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Rails, carriage support bearings, drive screws, shoes are presented in the model in a stylistic way. Three rails X (the distance between the end rails is 2000 mm) carry 15 bearings. The stiffness of 3000 N/mkmis attributed to each bearing. Two rails have three carriage support bearings each in respect to the coordinate Y, their stiffness is also 3000 N/mkm. The distance between the axes of these rails is 1650 mm.

The linear guideways along the axis Z carry only the slide and their standard size is less than that of linear guideways along the axes X and Y. That is why the spindle head carries 12 carriage support bearings Z with stiffness 2000 N/mkm each. They are connected to three rails on the slide.

The frame rests on three rows of bearings running along the axis X with a step 500 mm. The bearings are presented in a stylistic way as hollow truncated cones (shoes). It is assumed that the vertical stiffness of each bearing is equal to 1000 N/mkm. The example of representation of the stylized rails and bearings is given in Figure 2.

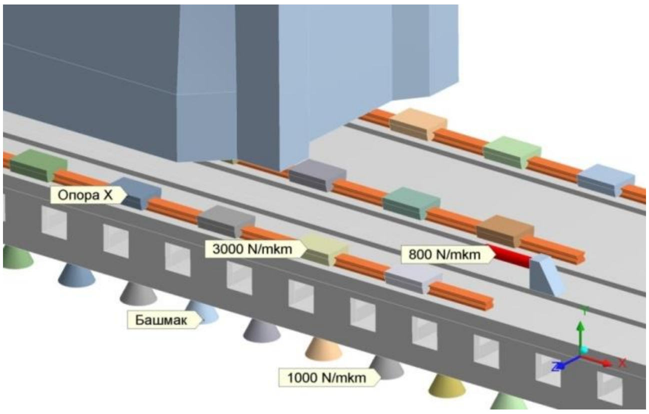

Figure 2 – Stylized representation of rails and carriage support bearings illustrated bythe carriage support bearing along the axis X

There are contact fining elements present in the model. They are located on the borders of all connected bodies. The so-called contact pairs are formed. Initially all contact elements are in a "bonded" state, that is, the contact pairs are blocked against slipping and releasing.

Modelling is performed in the linear setting. Geometrical nonlinearities are not taken into consideration because considerable deformations of the structure (compared to its size) are not expected. Physical nonlinearities associated with plastic deformation are not considered either, since the loads on the system are small with regard to its carrying capacity.

We have considered three versions of locking and loading of the machine basic units.

Figure 3 shows the first version of locking and loading of the rack assembly. It is assumed that all the parts are linked to each other with the help of contact elements. They jointly base themselves on the shoes. All the shoes (51 piece in this model) are fixed on the lower butt ends (marker A – "Fixed Support"). Here the rigid footing virtually begins. Marker B – "Standard Earth Gravity" – indicates the gravity direction. The structure is loaded with its own weight only.

The second version of loading consists in application of the testing force to the top of the rack. Two testing forces 500 N each act on the rail ends Y in the direction X. The total force 1000 N tries to shift right and bend the machine rack. If the bearings X are not blocked, the force is transmitted to the drive X as a result. Such loading characterizes stiffness of the rack and all the underlying parts. The testing force may be directed both along X and along Z. The spindle head and the slide do not participate in loading. The force of gravity is considered to be absent.

The third version of loading is application of the testing force tothe spindle butt end on the slide. The force value is 1000 N (in a linear modelthe force value is notessential). It is alternately directed along the axes X, Y, Z. In this way it is possible toreveal supplenessof all system parts and primarily of the slide, spindle head and their carriage support bearings. This testing force simulates the cutting force.

Preliminary analysis of machine loading versions has shown that the main impact on deformation displacements ofthe basic parts is caused by their weight, gravity center shifts of moving elements, residual voltages. The cutting forces applied to the spindle butt end produce only a slight impact. That is why we leave the first version of loading as the basic one.

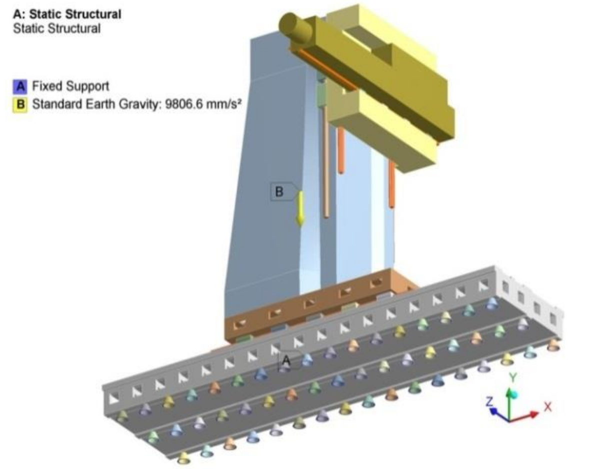

Figure 3 – The first version of locking and loading (A – bottom of the shoes, B – force of gravity) of the rack assembly.

In the work [5] the pattern of total deformation displacements USUM in the rack assembly in the basic version of loading is presented. The pattern of to tal deformation displacements USUM is divided into horizontal displacements along the axis X and vertical displacements Y. Assessment of the obtained results is conducted by the techniques submitted in [6].

Results of researches and discussion

With the reference to analysis of the obtained results it can be seen that the rack performs a deformation clock-wise rotation under the action of a heavy 30-tonspindle head. The rack's turn is related to a deformation "break" of the sledge. The sledge bends exactly under the front wall of the rack. The spindle heads weight is passed down along it. The sledge deflection is also caused by sagging of the under lying carriage support bearing along the axis X. Visually the deformation of the rack itself is not big. When turning, its profile gets moderately distorted. Also the frame on the bearings looks stiff enough. As for the slide, it is rather supple. Displacement patterns examined below are obtained by FEM-calculation at the blocked guides.

The general manifestation of deformation is "a dive" of the rack (clockwise turn). The suppleness of sledge and its carriage support bearings are responsible for it in the first place. Also a can tilevered bending of the extended slide has an impact. These deformations lead to vertical and horizontal shifting of the spindle butt end axis.

The mode of "dive" is illustrated by the section in Figure 4. It shows a deformation situationinside the sledge. The rack locally weighs on the sledge with its front wall by sagging to "248.03" mkm. The sledge bottom is relatively thin (bottom thickness $h_D = 40$ мм and bulges over the bearings X (for example, marker "186.72"). Sledge vertical ribs go into inter bearing gaps. The bearings X also sag under load (see Figure 4). This happens quite non-uniformly. The bearings under the front wall of the rack go down to about 104 mkm, and at the sledge edges – only to 42-48 mkm.

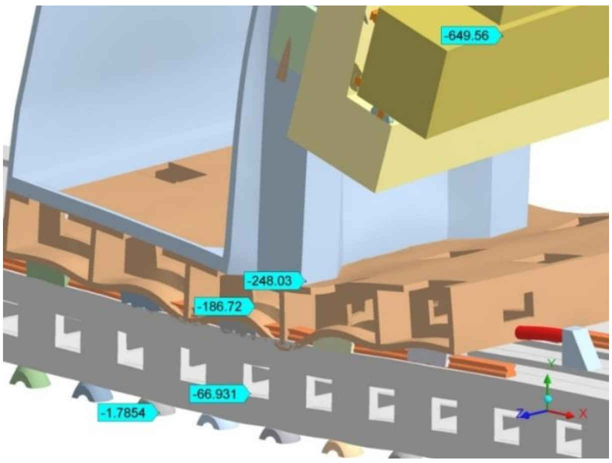

Figure 4 – Vertical displacements (mkm) under gravity in the section in the sledge area (×1200)

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Table 1 and Figure 5 show vertical reactions that act on the carriage support bearingsin the directionX (see Figure 2). Let's take the next to us rail as the first one. Ordinal numbers will be assigned to the bearings on each railfrom right to left. The nearest bearing in Figure 5 will be designated as bearing 1–1, and the farthest bearing – as bearing 3–5 (the first digit is a number of a rail). The highest overall reaction is present on the bearing 1-3. This bearing is close to the front wall of the rack that forms a load. The bearings at the edges of each railare loaded about three times as weak. It can be observed that the maximum load accrues to the first rail of the third row. The minimum reaction accrues tothe angle bearing 3-5. Forces in these bearings differ by a factor of 7, 9.

Following the results of modeling described above the sledge design was virtually modified. Firstly, the sledge bottom thickness h_D was increased. Secondly, the search was tried out for the optimal layout of carriage support bearings (roller bearings) with smaller number of bearings and their more uniform loading. Also the model material was changed in order to increase its stiffness.

Table 1 – Vertical reactions R_y in the bearings X in rows and rails (sledge bottom thickness is 40 mm), in N

	Rail 1	Rail 2	Rail 3	Portion - rows
Row 5	35728	19879	12870	8.6%
Row 4	70004	46281	48374	20.6%
Row 3	101740	84603	84767	33.9%
Row 2	74580	67189	66910	26.1%
Row 1	29957	30638	25497	10.8%
Portion – rails	39.0%	31.1%	29.9%	100%

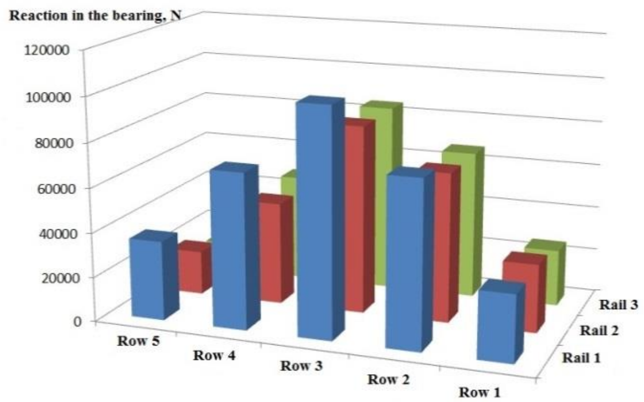

Figure 5 – Vertical reactions in the bearings X under the weight of rack assembly

Different versions of structural design (SD) are accumulated in Table 2. The case of SD1 refers to the basic version. Sledge bottoms were by turn enlarged to the thickness 80 and 120 mm (SD3 and SD4 respectively).

The maximum deformation displacement by gravity along the spindle axes X and Y is observed in the top position of the spindle headon the frame (SD1). At this point the maximum angle of obliquity is observed at sagging of the sledge on the bearings. The minimum displacement along the axis X is observedat the maximum descending of the spindle head (SD2). The problem may be solvedby increase in the bottom thickness to 40 mm.

ForacaseSD3 сh_D = 80 мм there can be observed a positive effect – spindle end hanging decreased by 9% - from 939 mkm up to 859 mkm (Table 2). "The dive" to the right decreased approximately by a factor of 1, 2. The cause of it is a smaller bulging of the sledge bottomover the bearings. The sledgebends to a lesser extendunder the front wall of the rack. The rack leans clockwise to a lesser extend.

Table 2 – Aggregate displacements of the spindle for different structural cases of the test (load – own weight)

Structural case	Description	Spindle, displacement X		Spindle, displacement Y	
		mkm	%	мкм	%
SD1	h_D = 40 мм	505	100,0%	939	100,0%
SD2	h_D = 40 мм and spindle head descending to 2 m	206	40,8%	952	101,4%
SD3	h_D = 80 мм	423	83,8%	859	91,5%
SD4	h_D = 120 мм	397	78,6%	821	87,4%
SD5	h_D = 80 мм and 11 bearings	411	81,4%	885	94,2%
SD6	h_D = 80 мм, and 11 bearings	393	77,8%	873	93,0%
SD7	h_D = 80 мм and sledge material stiffness increased up to	344	68,1%	790	84,1%
SD8	h_D = 80 ммand bearing X material stiffness increased	391	77,4%	821	87,4%

In the case SD 4 there is observed a slight difference in displacements from SD 3. Though the growth of the walls up to 120 mm results in a substantial increase of weight of the structure, it does not bring a substantial increase in stiffness of the machine.

Duetonon-uniformed loading of the bearings a version was modulated with decrease in the number of bearingsup to 11 (SD5) with simultaneous increase of bottom wall thickness up to 80 mm. The bearing row 4 and the bearing 1-2 are cut down. Displacements on the spindle have hardly changed compared to the version SD3. Loading of the bearings increases, for example, of the bearing 1-3 by 40%, though the reaction forceremains with in the limits of dynamic load rating of the bearings. The increase in sledge wall thickness to (SD6) does not make difference for spindle displacements compared to the version SD5.

The considered version with the increase in bearing thickness SD8 has shown a slight increase in stiffness compared to the version SD3 and worse values compared to the version SD7.

Thus, by degree of labor inputs and reduction of values of spindle displacement in the directions X and Y it is possible to offer the design change according to the versions SD3 and SD7.

Figure 6 showsthe degree of impact of the basic elements of the bearing system of the machine concernedat the structural cases of modeling SD1, SD3, SD7. If in the basic version the maximum impact on the deformation "dive" is made by the sledge and the bearings, in the offered versions the main impact is made bythe rack.

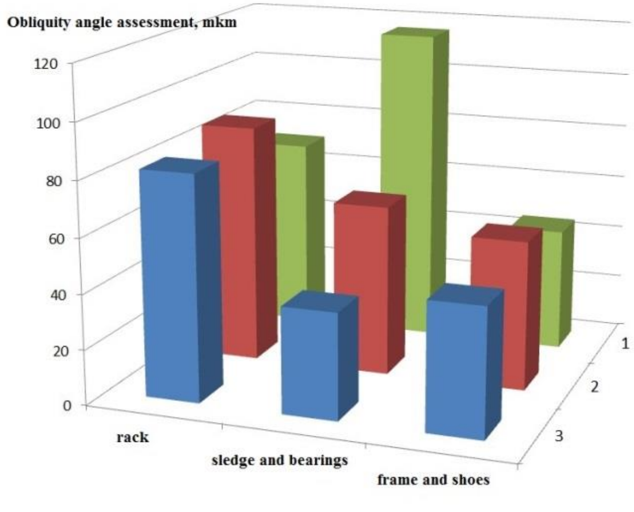

Figure 6 – The role of the rack assembly partsin forming the deformation "dive" (1 – SD1, 2 – SD3,3 – SD7)

The results of virtual variation of stiffness of the main parts of the machine assembly are presented in Table 3. Stiffness has been changed for each part by turn, twice: both downward and upward. For that end one operated the elastic modulus of the corresponding model material. We shall speak about a stiffness coefficient which in the initial state is equal to one f_j= 1 for design SD3, at the decrease in stiffness it is f_j = 0,5, at the increase it is f_j = 2. The result of modeling is displacement of the spindle end along the axis X ("dive").

Table 3 – Spindle "dive" along the coordinate X depending on the stiffness coefficient of the part, in mkm

Stiffness coefficient	Shoe	Bearing	Frame	Sledge	Rack
0,5	443	458	447	562	572
1	423	423	423	423	423
2	401	391	396	344	343

It is evident from Table 3 that the greatest impact on "the dive" is made by stiffness of the sledge and the rack. After increase of the sledge stiffness the further lowering of "the dive" should be attained by elaboration of the rack design. The frame, the shoes and bearings X have roughly the same and moderate effect on "the dive". Expenditures for their reinforcement are unreasonable.

Conclusion

It is evident from the obtained results of modelling that the greatest deformations of the machine are due to sledge and rack deflection along the axis X, the reat the deflection increases when the spindle head goes upward. To increase the sledge stiffness, the most preferred method will be to increase its bottom thickness to 80 mm. Also, in order to achieve an even load distribution on the frame guides, it is necessary to change the installation diagram of roller bearings. To reduce the impact of the rack deflection on processing accuracy by using a mathematical model describing deviations, it is possible to negate errors with the help of correction ofthe control programme in the machine CNC system.

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