

## Static Stiffness Modeling of a Novel PKM-Machine Tool Structure

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### Abstract

This article presents a new configuration of a 3-dof machine tool with parallel kinematics. Elastic deformations of the machine tool have been modeled with finite elements, stiffness coefficients at characteristic points of the working area for different cutting forces have been calculated.

Keywords: parallel kinematics, multipurpose machine tools, stiffness, FEM analyses

### 1. Introduction

There is a growing research interest in parallel kinematics structure machines, as they are used in high-speed machining (HSM) which is increasingly used in drilling / milling / boring multi-task machine. HSM puts high demands on speed, acceleration and positioning accuracy of the machine, which cannot be accommodated with machines with traditional serial kinematics. The reason for this is that in a typical configuration machine a sequential movement of slides relatively to each other provides the traversing path for the main body (spindle assembly). In such kind of system forces cause bending of the supporting elements. The effect of these deformations (total compliance of the machine) is to reduce significantly machining accuracy. Reserves of decreasing the compliance of these machines by optimizing the configuration factors mostly exhausted, and to further enhance rigidity (decrease compliance) the mobile elements mass increasing is used dramatically reducing the machine feeds velocity and acceleration limits.

An effective solution to these problems is the development of machines using parallel kinematic structures. Such structures, in one case, provide a coordinated change of lengths of the rods that hold the cutting tool; or –in a coordinated manner reposition the constant length rods, moving on framework guides. This principle has been used by many machine tool companies in U.S.A., France, Germany, Spain, Turkey and other countries [1,2].

An example of PKM machine tool with constant length rods is the V100 of INDEX (Germany) [3].

A jointed arrangement of rods develops only compressive stresses in them. With the exception of bending they can dramatically increase the overall stiffness of the system while maintaining a low moving parts mass. This makes it possible to reach high feed accelerations and speeds during machining.

Existing machines with parallel kinematics have an increased speed but still lack rigidity compared to classical configurations. Depending on the design features, their stiffness varies from 5 to 200 N/ $\mu\text{m}$  [4, 5].

Therefore research on machine tool configurations based on parallel kinematics, which would possess a high level of stiffness, is a promising direction in research.

### 2. Machine tool kinematics structure

On Fig. 1 a drawing of a 3-dof PKM milling machine tool structure is shown [6].

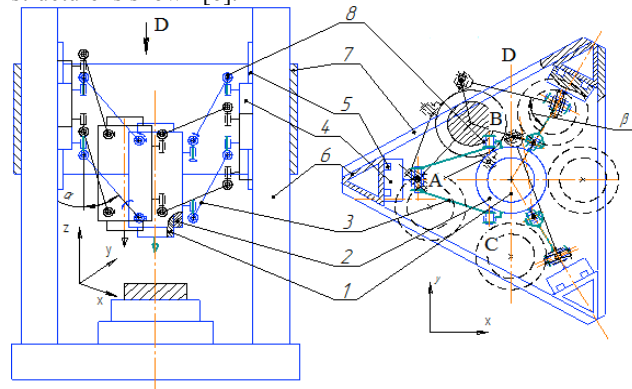


Fig. 1. PKM machine tool structure. 1 – milling spindle; 2 – mobile platform; 3 – rod; 4 – carriage; 5 – guides; 6 – framework; 7 – strut; 8 – joint.

A distinctive feature of this configuration is that the constant length rods (3) are arranged in pairs in the normal plane, forming a rectangular structure which provides a linkage between spindle (1) and columns (6) through joints (8), carriages (4) and guides (5). One of the carriages is connected with the spindle by two pairs of rods forming the triangle ABC. Furthermore, columns (6) at a height of

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guides (5) are connected by struts (7) also forming a triangle.

These differences provide increased rigidity of the machine as a whole, because all loaded structural elements are the sides of a triangle, characterized by high stiffness.

Thus, the kinematic structure of the machine consists of three kinematic chains connecting spindle with columns. One of the chains has a PUR – structure (P – prismatic active joint; U – universal joint; R – revolute joint). Two other chains have PUU – structure. This structure provides 3 DOFs to spindle: linear movement along Z axis, revolution around PUR-chains carriage vertical axis and linear movement along the revolution vector. This provides the possibility of spindle movement in three Cartesian coordinates.

### 3. Stiffness modeling

The objective of this work is to study a new 3-DOF PKM milling machine static stiffness with finite elements. The elastic deformations were calculated with SolidWorks simulation software.

The finite element analysis method was chosen in order to account for all part compliances while producing accurate results.

On Fig. 2 a 3D model of a machine is shown whose moving parts have a total weight of 380 kg.

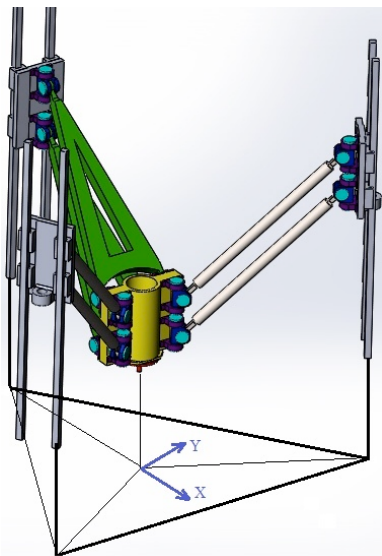


Fig.2. – 3D model of a machine

This study assumed that the support columns 6 (Fig. 1) have an absolute bending stiffness. The rods length is 1000 mm each.

Fig. 3 shows the universal joints design.

Its distinguishing feature is the axes that do not intersect but cross at a right angle and are located at a certain distance from each other. This design choice was made to provide a wider range of rod orientation angles without significantly increasing the axial dimensions of the universal joint. Normal and axial stiffness of tapered roller bearings are taken to be 200 N/μm and 100 N/μm, respectively [7].

Stiffness of guide sliders assumed to be 1000 N/μm in both normal and tangential directions [8,9].

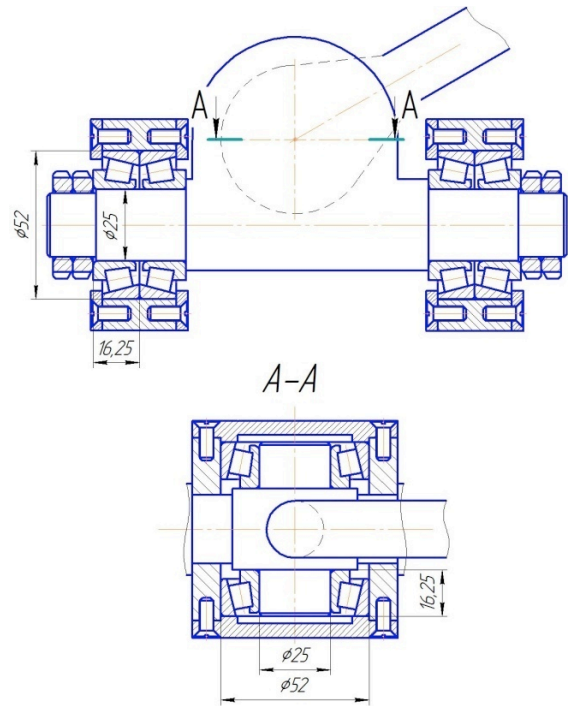


Fig. 3. – U-joints design

The axial stiffness of the ball screw (Fig. 4) was calculated as follows:

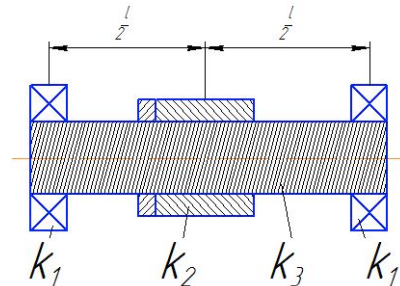


Fig. 4. Ballscrew

If the screw is axially locked on both sides, the overall systems stiffness will be:

$$k = \frac{4k_1k_2k_3}{k_1k_2 + 2k_2k_3 + 4k_1k_3}$$

where  $k_1 = 150 \text{ N}/\mu\text{m}$  – axial stiffness of the support;  
 $k_2 = 1000 \text{ N}/\mu\text{m}$  – axial stiffness of the nut;

$$k_3 = \frac{ES}{l} = \frac{2.1 \cdot 10^5 \cdot \frac{40^2}{4} \pi}{1200} = 220 \cdot 10^3 \text{ H/mm}$$

– axial stiffness of the screw.

Thus, substituting (2), (3) and (4) into (1) yields:

$$k = \frac{4 \cdot 150 \cdot 1000 \cdot 220}{1.5 \cdot 10^5 + 4.4 \cdot 10^5 + 1.32 \cdot 10^5} = 182.8 \text{ N}/\mu\text{m}$$

The study was completed for six points of the working space (Fig. 5). Positions 4'; 5'; 6' are similar to 4; 5 and 6 due to the symmetry of the structure relative to the X axis.

For each point three tests were performed:

1 – Acting force parallel to the X axis and attached to the point located at 50 mm distance from the spindle face and at 5 mm distance from the spindle axis along the Y axis;

2 – Acting force parallel to the Y axis and attached to the point located at 50 mm distance from the spindle face and at 5 mm distance from the spindle axis along the X axis;

3 – Acting force parallel to the Z axis.

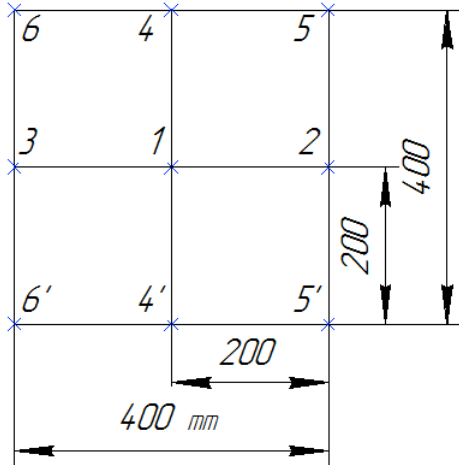


Fig. 5 – Working area arrangement

Load value in all experiments was set to 1000 N and applied to the spindle face as a "remote load".

#### 4. Modeling results

Table 1 shows the tools displacement values in each axis direction for the different cutting forces.

Table 1. Tools displacement for cutting forces ( $\mu\text{m}$ )

№		$F_x$	$F_y$	$F_z$
1	$\delta_x$	24.60	0.211	0.307
	$\delta_y$	0.173	20.98	0.024
	$\delta_z$	1.088	0.138	7.610
2	$\delta_x$	15.76	0.138	2.133
	$\delta_y$	0.114	21.38	0.076
	$\delta_z$	0.312	0.280	9.84
3	$\delta_x$	28.54	0.180	0.835
	$\delta_y$	0.072	16.37	0.026
	$\delta_z$	1.091	0.088	6.95
4	$\delta_x$	31.53	14.58	11.01
	$\delta_y$	0.793	24.15	6.602
	$\delta_z$	8.626	8.746	17.95
5	$\delta_x$	17.00	7.936	5.595
	$\delta_y$	0.266	23.60	5.280
	$\delta_z$	4.234	6.208	21.05
6	$\delta_x$	49.64	27.95	16.09
	$\delta_y$	2.329	26.90	9.314
	$\delta_z$	11.91	13.04	16.60

Graphs of the total stiffness of the machine at characteristic points of the working area for different directions of the cutting forces are shown below.

Table 2 shows the values of the angular stiffness of the machine around the X and Y axes for different directions of the force for six points mentioned above.

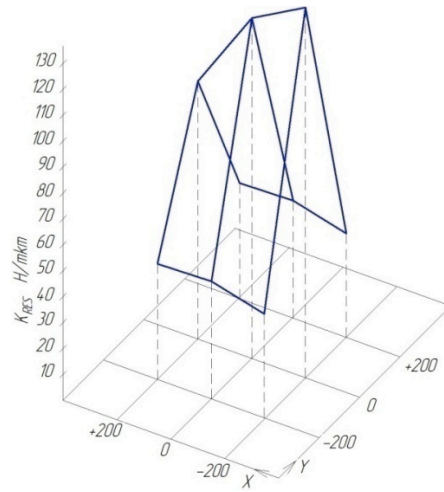


Fig. 6. Stiffness of machine for a cutting force  $F_z = 1000 \text{ N}$

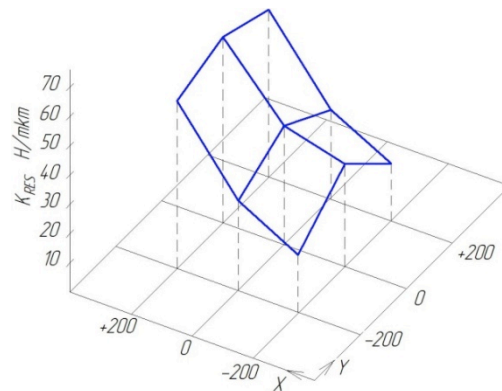


Fig. 7. Stiffness of the machine for a cutting force  $F_x = 1000 \text{ N}$

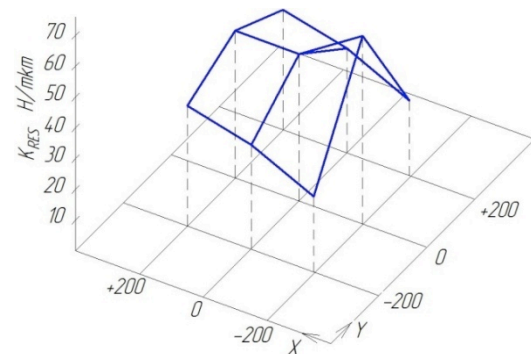


Fig. 8. – stiffness of the machine for a cutting force  $F_y = 1000 \text{ N}$

Table 2. Angular stiffness of machine (H/arc sec.)

№		$F_x$	$F_y$	$F_z$
1	$J_x$	952	91	20000
	$J_y$	65	833	357
2	$J_x$	1613	72	909
	$J_y$	94	1492	364
3	$J_x$	495	465	1449
	$J_y$	11111	9090	1639
4	$J_x$	100	46	62
	$J_y$	56	191	971
5	$J_x$	203	48	59
	$J_y$	86	223	1111
6	$J_x$	91	41	74
	$J_y$	41	166	292

The linear and angular stiffness of the proposed configuration has a distinct non-uniformity within the working area (see Figs.6-8) because one of the rods pair is part of a triangle. On the other hand, the average values of stiffness are considerably higher in comparison to those of a hexapod machine [3] found with experiments (linear – 4-7 times higher; angular – 2-10 times higher).

The linear stiffness values calculated here are similar to those of some three-axis machines with the serial kinematic structure, which have a total moving mass several times higher value.

The presence of points characterized by angular rigidity values exceeding  $10^3$  *Narc sec.* may indicate that there is a scope for further optimization of the proposed machine configuration using the angular stiffness criterion, which has

a big importance in five-axis machining of aircraft parts where manufacturing accuracies are required.

## 5. Conclusion

Results show that designing parallel kinematic structured machine tools for high speed machining of complex parts is promising for its ability to combine an acceptable level of stiffness and low inertia, and thus reach high speeds and accelerations of the cutting tool.

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