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ENSURING THE RELIABILITY AND PERFORMANCE CRITERIAS OF CRANKSHAFTS

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Abstract

The issues of efficiency improvement of manufacturing crankshafts in order to ensure their reliability and performance criteria are the priorities in modern production of internal combustion engines. Using the capabilities of modern special grinding machines can improve the quality of machining and obtain the necessary running characteristics of crankshafts. In work the questions connected with development of a method of calculation of rigidity of crankshafts for increase of accuracy of their machining, reliability and performance criteria's are considered. Based on the proposed methodology, numerical calculations have performed and the possibility of determining the deflections and crankshafts rigidity in any section have been justified. The original construction of the following grinding steady rest for CNC grinding machines specified for machining the crankshaft main bearing journal and connecting rod journal is proposed. The construction design of the device allows for compensating the influence of the cutting force on the elastic strain of the part, depending on the change in its rigidity. The practical value of the research includes in develop recommendations for determining the optimal parameters for the round infeed grinding cycle of the crank pins from the point of view of productivity and accuracy.

Keywords: crankshaft, initial parameters method, reliability, grinding machine, rest.

1. INTRODUCTION

Modern machine industry is characterized by increasing requirements for quality assurance and precision manufacturing of critical machine parts. The reliability of operation of any machine is directly proportional to the break-even reliability of its components and mechanisms.

The crankshaft is considered one of the most critical parts of an internal combustion engine since it solves the problem of converting translatory motion from engine piston to rotational moment [1-4]. It should be noted that this element of the crank mechanism is a constructively and technologically complex member. This is why high requirements to the accuracy of manufacturing crankshafts are being put demanded. Among the main requirements should be noted:

- the accuracy of diametrical dimensions of the crankshaft main bearing journals and crankshaft rod journals IT6 (less often IT5);
- the form tolerance of the crankshaft main bearing journals and crankshaft rod journals are not more than 0.3 of the permissible error for the diameter of these crank pins;
- deviations from alignment of the location of the crankshaft main bearing journals not more than 0,02 mm, from the parallelism of the axes of the crankshaft main bearing journals and crankshaft rod journals not more than 0,015 mm on the length of the crank pin;
- hopping of the crankshaft main bearing journals relative to the axis of the center holes in the range 0,01 ... 0,03 mm;
- surface roughness of the crankshaft main bearing journals and crankshaft rod journals in the range Ra = 0.08 ... 0.32 μm.

When processing workpiece of crankshafts, the structure of the construction of the technological process operations and the choice of equipment depend on the volume of output products. At the same time, in any process, the state of the base surfaces and the finishing operations for the formation of high precision processing are given special attention [1]. To reduce deformations when processing flexible crankeds, distributing and other shafts, in addition to the rests, special machines with a central or double-sided drive are used.

As of today for final polishing and finishing processing CNC grinding machines have found the greatest application [2]. These machines implement high quality and precision of parts processing, but at the same time, their work as the dynamic systems has not been studied enough. This involves difficulties in the development of new equipment, increasing its accuracy and chatter stability. In this regard, studies and detailed calculations of the rigidity of all elements of the technological system become necessary. When developing a general model of metal-cutting machine, it is important to take into account the dynamics of each elements of the dynamic system. It is needed to know the rigidity of each element for determine their dynamic characteristics [5]. The total rigidity of the machine's dynamic system consists of a series and parallel connection of elements. This value depends on the rigidity of the workpiece, as often the rigidity of the workpiece is much less than the rigidity of other elements of the machine's dynamic system.

From the analysis of modern information sources, the actual task of ensuring the accuracy and quality of manufacturing critical parts of machines [6-8] such as a crankshaft that affect the reliability of the overall mechanism during working was defined.

Author of paper [6], taking into account the qualitative characteristics of the crankshaft, have presented the results of the analysis of how sea waves affect the angular speed of a propulsion. In the article, the method of assessing the state of the ship's engine was considered. Implementation of the method is possible, including with the appropriate qualitative characteristics of the crankshaft.

In article [7] there is described the details of the method designed to determine parameters of vehicle’s internal combustion engine with compression ignition (CI) during road tests. The method requires simultaneous measurements for the crankshaft rotation frequency, fuel pressure in the injector, pressure in the combustion chamber, air pressure and temperature in the intake system [7]. It is not possible to obtain an adequate mathematical model using this method if the crankshaft does not meet the criteria for quality and accuracy.

In the article [9] authors have considered the crankshaft (taking into account the fact that the part corresponds to high quality and accuracy characteristics) and presented the model of the crankshaft taking into account the coupling of bending and torsional vibrations. The paper is summarized by giving conclusions about the analysis of crankshaft systems with a dynamic eliminator of vibrations and their modelling.

The authors of paper [10] noted that crankshaft-springing characteristics are one of the most important from the ships' main engines reliability point of view. The adequacy of this characteristic depends on the quality and accuracy of the crankshaft machining on the working machine. The authors noted that development of the analysis methods of crankshaft's stiffness characteristics is the first step of planned monitoring system that will be able to verify crankshaft-springing characteristics by continuous measurements of the crankshaft free-end's axial deformations [10].

Thus, many researchers investigate the behavior of the crankshaft during the working of the complete mechanism and try to increasing the operational characteristics both the details and the mechanism as a whole. However, the actual task remains to make initially high-quality engineering product. For this, it is important to accept and process the specifications for the production of the crankshaft.

Thus, the purpose of this article is to propose methods for calculating the rigidity of crankshafts for improving the accuracy of their processing, reliability and performance indexes.

2. CALCULATION OF THE CRANKSHAFT RIGIDITY USING THE INITIAL PARAMETERS

To determine the deformations in multi-diameter parts that work under the conditions of applied loads, it was decided to apply a modified initial parameters method [1, 7, 11-12]. The essence of the method consists in changing the stepped part with an equivalent piece of constant rigidity.

The analytical model for determining the deformation movements and the crankshaft rigidity is shown in Fig. 1.

The characteristic property of the crankshafts is that the connecting rod journals, as well as the webs connecting them with the crankshaft main bearing journals, have relevance an eccentricity to the axes of the crankshaft main bearing journals and the shaft as a whole. At the same time, their unbalanced masses will cause significant dynamic loads during grinding. Therefore, to study the dynamic system, it is important to determine the value of the crankshaft rigidity. The load from the cutting force in the form of the resultant force *P* will be applied in the middle of the crank pin (see Fig.1, cond.I).

We divide the crankshaft into three equal parts (see Fig.1, cond.I-III), applying the corresponding force factors *Q* (lateral force) and *M* (bending moment) in the cut points, which in our case can be found by the formula (1):

on condition $x < a$:

$$Q_i = \frac{P\cdot(L-a)}{L}, \text{ and } M_i = \frac{P\cdot(L-a)\cdot I_i}{L},$$

on condition $x > a$: (1)

$$Q_i = \frac{P\cdot(L-a)}{L} - P, \text{ and } M_i = \frac{P\cdot a}{L\cdot(L-I_i)},$$

where

a – coordinate of force application, (mm);

I_i – coordinate of the corresponding section, (mm).

After that, it is necessary to multiply the obtained values Q_i and M_i by the corresponding reduction factors determined by the formula (2):

$$\beta_i = \frac{I_0}{I_i}, \tag{2}$$

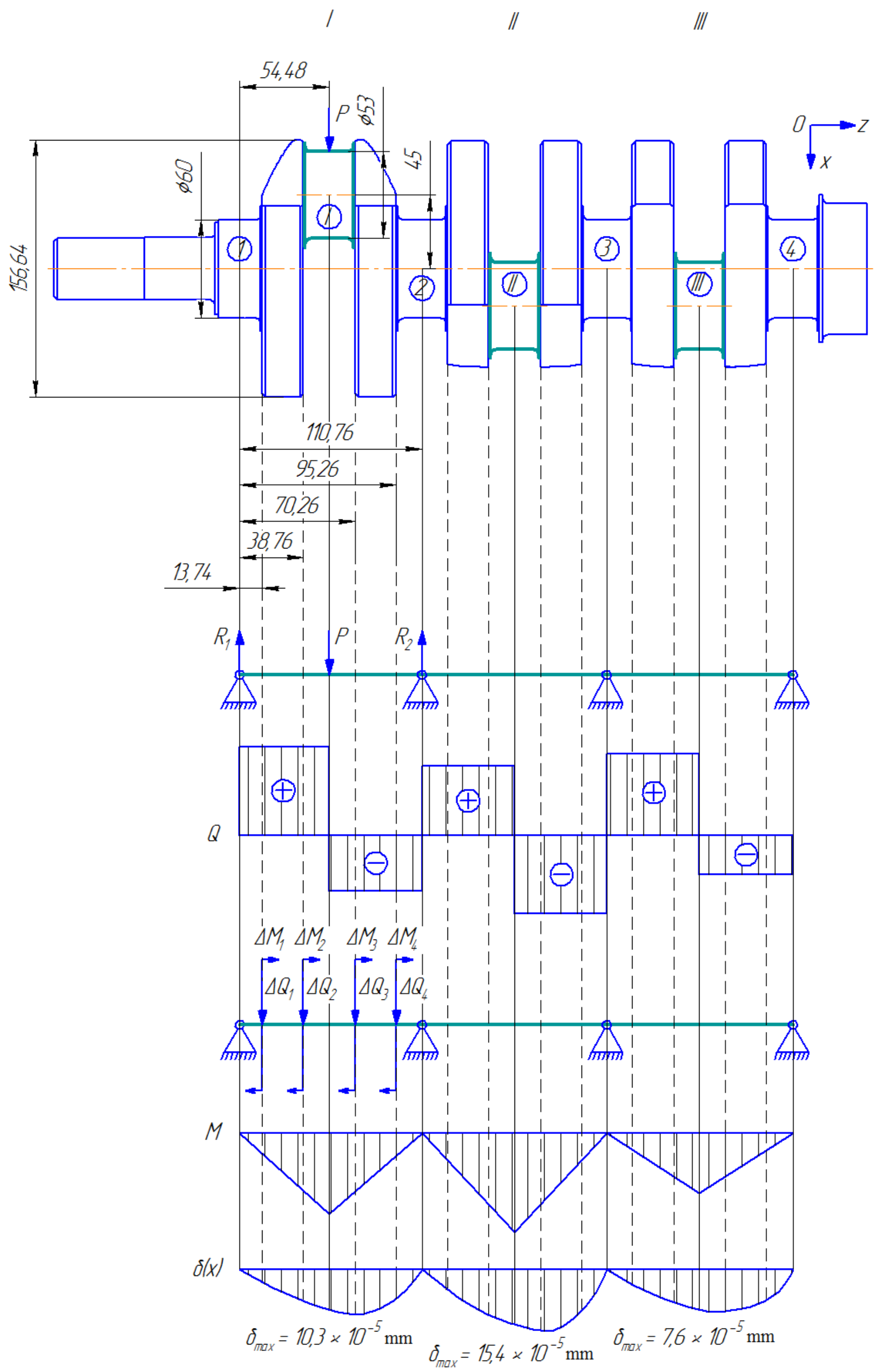

Fig. 1. The crankshaft analytical model

where

I_0 – the equivalent moment of inertia, equal to the moment of inertia of one of the sections to which the reduction takes place, (mm^4);

I_i – inertia moment of the corresponding area, (mm^4) [11].

For cylindrical sections I_i it corresponds to:

$$I_i = \frac{\pi \cdot d_i^4}{64}, \qquad (3)$$

where

d_i – diameter of the corresponding section, (mm).

For eccentric surfaces of crankshaft connecting rod journal I_i will be determined as:

$$I_i = \frac{\pi \cdot d_i^4}{64} \cdot e^2 \cdot \frac{\pi \cdot d_i^4}{4}, \qquad (4)$$

where

e – eccentricity between the axis of the connecting rod bearing journal and main bearing journal, (mm);

d_i – diameter of the crankshaft pin, (mm).

For webs connecting the crankshaft neck and crankshaft pin *I* can be defined as:

$$I_i = \frac{b \cdot h^3}{12} \cdot \cos^2 \alpha + \frac{h \cdot b^3}{12} \cdot \sin \alpha + e_l^2 \cdot \frac{\pi \cdot d_i^2}{4}, \qquad (5)$$

where

b – width of web, (mm);

h – height of web, (mm);
e_1 – eccentricity between the axis of the crankshaft neck and the web, (mm);
α – corresponding rotation angle of the crankshaft;
d_i – diameter of the crankshaft neck (main bearing journal), (mm).

From the formula (5) it is seen that the inertia moment of the webs varies depending on the rotation angle of the crankshaft. Its value is shown in Table 1.

Table 1. Values of inertia moments of crankshaft webs

α, (°)	I·10⁻⁷, (mm⁴)	α, °	I·10⁻⁷, (mm⁴)	α, °	I·10⁻⁷, (mm⁴)
20	4,175	140	3,84	260	3,21
40	3,84	160	4,175	280	3,21
60	3,459	180	4,308	300	3,459
80	3,21	200	4,175	320	3,84
100	3,21	220	3,84	340	4,175
120	3,459	240	3,459	360	4,308

As a result of joining-up the split parts we obtain a beam of constant cross section, which is loaded with the given external loads. At the junction points, it is important to take into account the additional forces *ΔQ* and the moment *ΔM*, whose values are determined by formula (6):

$$\begin{aligned}\Delta M_i &= M_i\cdot(\beta_i-\beta_{i-1}),\\ \Delta Q_i &= Q_i\cdot(\beta_i-\beta_{i-1}),\end{aligned} \tag{6}$$

In order to determine the displacements in the resulting equivalent beam, an universal elastic-line equation has been used [11], which has the form (7):

$$\delta(x)=w_0+\theta_0 x+\frac{1}{El_i}\times \times\left[\sum_{n=1}^{i}M_i\cdot\frac{(x-l_i)^2}{2}+\sum_{n=1}^{i}Q_i\cdot\frac{(x-l_i)^3}{6}+R\cdot\frac{x^3}{6}\right] \tag{7}$$

where
w_0 – initial deflection in the left support, (mm);
θ_0 – initial rotation angle in left support, (°);
x – running coordinate, (mm);
E – Young's modulus, (N/mm²);
M_i – bending moment at the corresponding section, (N·mm);
l – coordinate of the relevant section, (mm);
Q_i – transverse force at the relevant section, (N);
R – reaction in the left support, (N).

The crankshaft is not a rigid part, and the requirements for accuracy and roughness of its surfaces are quite high. Given this fact, two intermediate supports in the form of rests were proposed to meet these quality requirements (Fig. 1).

At the first stage of solving the task of ensuring the quality of crankshaft processing, the equation of the elastic line for the first part of work-piece have been written (8):

$$\begin{aligned}\delta(x)=w_0+\theta_0 x+\frac{1}{El_i}\times\\ \times\Bigg[&\frac{P\cdot(L-a)\cdot l_1}{L}\cdot\frac{(x-l_1)^2}{2}+\frac{P\cdot(L-a)\cdot l_2}{L}\times\\ &\times\left(\frac{l_1}{l_2}-\frac{l_1}{l_1}\right)\cdot\times\frac{(x-l_2)^2}{2}+\frac{P\cdot a}{L\cdot(L-l_3)}\times\\ &\left(\frac{l_1}{l_3}-\frac{l_1}{l_2}\right)\cdot\frac{(x-l_3)^2}{2}+\frac{P\cdot a}{L\cdot(L-l_4)}\cdot\left(\frac{l_1}{l_4}-\frac{l_1}{l_3}\right)\times\\ &\times\frac{(x-l_4)^2}{2}+\frac{P\cdot(L-a)}{L}\cdot\frac{(x-l_1)^3}{6}+\\ &+\frac{P\cdot(L-a)}{L}\cdot\left(\frac{l_1}{l_2}-\frac{l_1}{l_1}\right)\cdot\frac{(x-l_2)^3}{6}+\\ &+\left(\frac{P\cdot(L-a)}{L}-P\right)\cdot\left(\frac{l_1}{l_3}-\frac{l_1}{l_2}\right)\cdot\frac{(x-l_3)^3}{6}+\\ &+\left(\frac{P\cdot(L-a)}{L}-P\right)\cdot\left(\frac{l_1}{l_4}-\frac{l_1}{l_3}\right)\cdot\frac{(x-l_4)^3}{6}+\frac{P}{2}\cdot\frac{x^3}{6}\Bigg]\end{aligned} \tag{8}$$

Note that the equation of the elastic line for the second and third part of the crankshaft will be similar to the equation of the elastic line of the first part of the crankshaft, the difference will be only in the values of the lengths of some sections.

The rigidity of the workpiece is determined without taking into account the rigidity of the supports, as well as the spindle, stock head, deadhead and rests. Therefore, the initial rotation angle in the left support is found from the initial conditions $x=L$; $w_0=0$; $E=2\cdot105$ N/mm^2; $P=157$ N. From here, the equation (9) was obtained:

$$\begin{aligned}\theta_0=-\frac{1}{El_iL}\times\\ \times\Bigg[&\frac{P\cdot(L-a)\cdot l_1}{L}\cdot\frac{(L-l_1)^2}{2}+\frac{P\cdot(L-a)\cdot l_2}{L}\times\\ &\times\left(\frac{l_1}{l_2}-\frac{l_1}{l_1}\right)\cdot\frac{(L-l_2)^2}{2}+\frac{P\cdot a}{L\cdot(L-l_3)}\times\\ &\times\left(\frac{l_1}{l_3}-\frac{l_1}{l_2}\right)\cdot\frac{(L-l_3)^2}{2}+\frac{P\cdot a}{L\cdot(L-l_4)}\cdot\left(\frac{l_1}{l_4}-\frac{l_1}{l_3}\right)\times\\ &\times\frac{(L-l_4)^2}{2}+\frac{P\cdot(L-a)}{L}\cdot\frac{(L-l_1)^3}{6}+\\ &+\frac{P\cdot(L-a)}{L}\cdot\left(\frac{l_1}{l_2}-\frac{l_1}{l_1}\right)\cdot\frac{(L-l_2)^3}{6}+\\ &+\left(\frac{P\cdot(L-a)}{L}-P\right)\cdot\left(\frac{l_1}{l_3}-\frac{l_1}{l_2}\right)\cdot\frac{(L-l_3)^3}{6}+\\ &+\left(\frac{P\cdot(L-a)}{L}-P\right)\cdot\left(\frac{l_1}{l_4}-\frac{l_1}{l_3}\right)\cdot\frac{(L-l_4)^3}{6}+\frac{P}{2}\cdot\frac{L^3}{6}\Bigg]\end{aligned} \tag{9}$$

At this stage, it is possible to determine the amount of deflection of the crankshaft at any place in the longitudinal and in the cross sections, depending on the angle of rotation. To find the deflection in the general equation of an elastic line, it is necessary to take only those components of the force factors that lie to the left of the required cross-section.

After finding the deflections, the rigidity of the corresponding sections was founded, in accordance with formula (10)

$$C_x = \frac{P}{\delta(x)}. \qquad (10)$$

The results of calculations of deflections and roughness's of connecting rod bearing journals depending on the rotation angle of the crankshaft have presented in Table 2.

Table 2. The calculated values of deflections and stiffness's of connecting rod journal

α, (°)	δ·10^5, (mm)	C·10^{-5}, (N/mm)	α, (°)	δ·10^5, (mm)	C·10^{-5}, (N/mm)
First crank pin					
1	2	3	4	5	6
20	10,07	15,49	200	10,08	15,48
40	10,15	15,37	220	10,16	15,35
60	10,21	15,28	240	10,22	15,26
80	10,23	15,25	260	10,24	15,23
100	10,19	15,31	280	10,18	15,32
120	10,10	15,45	300	10,10	15,45
140	10,02	15,57	320	10,01	15,58
160	9,986	15,62	340	9,672	16,13
180	9,980	15,63	360	9,985	15,62
Second crank pin					
20	14,93	10,45	200	14,98	10,41
40	14,94	10,44	220	14,95	10,43
60	15,04	10,37	240	15,05	10,37
80	15,21	10,26	260	15,20	10,26
100	15,27	10,22	280	15,33	10,18
120	15,35	10,16	300	15,38	10,14
140	15,29	10,20	320	15,33	10,18
1	2	3	4	5	6
160	15,21	10,26	340	15,20	10,26
180	15,09	10,34	360	15,07	10,35
Third crank pin					
20	6,542	23,85	200	6,544	23,84
40	6,001	26,00	220	5,982	26,08
60	5,766	27,06	240	5,780	26,99
80	6,030	25,87	260	6,041	25,82
100	6,624	23,55	280	6,621	23,56
120	7,203	21,66	300	7,221	21,60
140	7,537	20,70	320	7,558	20,64
160	7,519	20,75	340	7,557	20,64
180	7,165	21,77	360	7,168	21,76

3. DETERMINATION OF THE CRANKSHAFT RIGIDITY BY THE SIMULATION APPROACH

With the invention of modern computer technology, the analysis of complex details has become simpler [13-16].

To determine the deflection of the connecting rod journal of the crankshaft under the action of the cutting force during grinding the simulation method was used. This was made possible by minimizing the simplifications of the mathematical model and increasing the accuracy of the calculation of force factors. The model of the crankshaft with the specified fastening and the action vectors of the cutting forces is shown in Fig. 2.

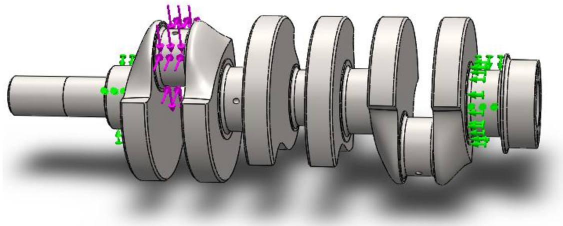

Fig. 2. Three-dimensional model of the crankshaft

Initial conditions for modeling:

- material of the part – steel 40HNMA;
- fixing type – rigid;
- the cutting force was decomposed into two components: P_y = 142,6 N, P_z = 64,8 N.

The diagram of the crankshaft model deflections under the influence of the cutting force is shown in Fig. 3.

The total results of the deflections values and rigidities of connecting rod journal (Table 3), depending on the angle of the crankshaft rotation, were determined using the simulation method.

4. ANALYSIS OF SIMULATION RESULTS

As can be seen from the tables 2 and 3, the calculation of the rigidity and elastic deformation of crankshafts by the initial parameters method ensures a sufficiently high convergence with the simulation results.

For further analysis, the data obtained at the stage of simulation modelling were accepted. This made it possible to plot the out-of-roundness profile of the cross section strain of the crankshaft pin under the influence of the cutting force. In Fig. 4 shows the out-of-roundness profile for the most dangerous, from the point of view of the form accuracy, the second crank pin.

From Fig. 4 it can be seen that, depending on the crankshaft rotation angle during grinding, its strain has different characteristics (by a factor of 1.5), thereby directly affecting the accuracy of the shape of the machined crank pin.

Taking into account possible changes in the cutting conditions (namely, their intensification), additional studies were carried out to obtain an index of the cross section strain of the crankshaft rod journal δ depending on the resultant cutting

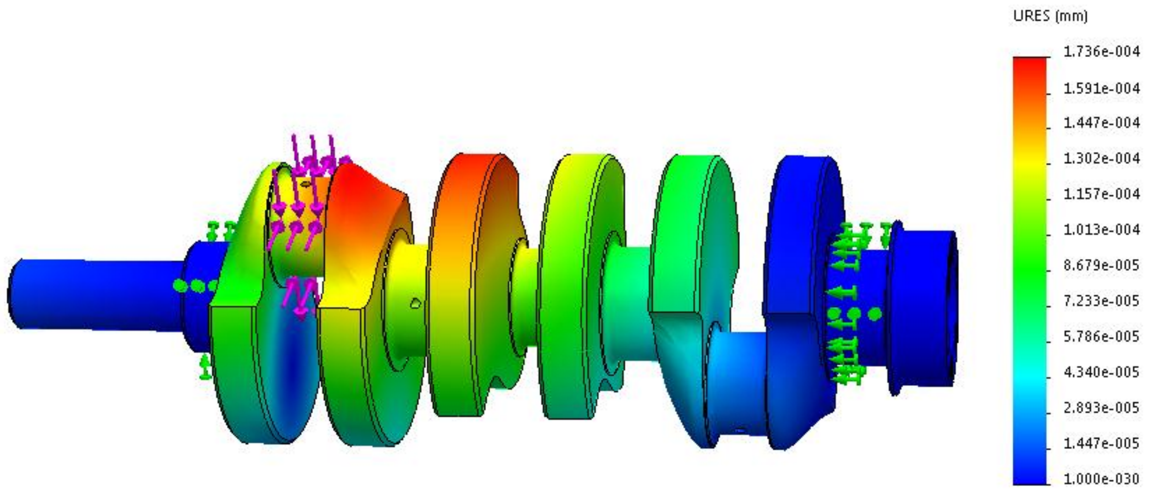

Fig. 3. Strain diagram of the crankshaft model

Table 3. The total results of the deflections values and rigidities of connecting rod journal

First crank pin					
α, (°)	$\delta\cdot10^5$, (mm)	$C\cdot10^{-5}$, (N/mm)	α, (°)	$\delta\cdot10^5$, (mm)	$C\cdot10^{-5}$, (N/mm)
1	2	3	4	5	6
20	15,81	9,930	200	15,83	9,92
40	15,93	9,854	220	15,95	9,842
60	16,04	9,790	240	16,04	9,787
80	16,05	9,780	260	16,08	9,767
100	16,00	9,814	280	15,99	9,821
120	15,85	9,906	300	15,85	9,906
140	15,73	9,979	320	15,72	9,986
160	15,68	10,01	340	15,65	10,03
180	15,67	10,02	360	15,68	10,02
Second crank pin					
20	33,88	4,635	200	36,79	4,267
40	40,11	3,915	220	40,15	3,910
60	65,20	2,408	240	65,25	2,406
80	89,20	1,760	260	89,27	1,759
100	105,16	1,493	280	105,11	1,494
120	109,86	1,429	300	109,84	1,429
140	102,80	1,527	320	102,82	1,527
1	2	3	4	5	6
160	85,10	1,845	340	85,19	1,843
180	60,19	2,608	360	60,27	2,605
Third crank pin					
20	1,03	152,9	200	1,03	152,8
40	0,94	166,6	220	0,94	167,2
60	0,91	173,4	240	0,91	173,0
80	0,95	165,9	260	0,95	165,5
100	1,04	151,0	280	1,04	151,0
120	1,13	138,8	300	1,13	138,5
140	1,18	132,7	320	1,19	132,3
160	1,18	133,0	340	1,19	132,3
180	1,12	139,6	360	1,13	139,5

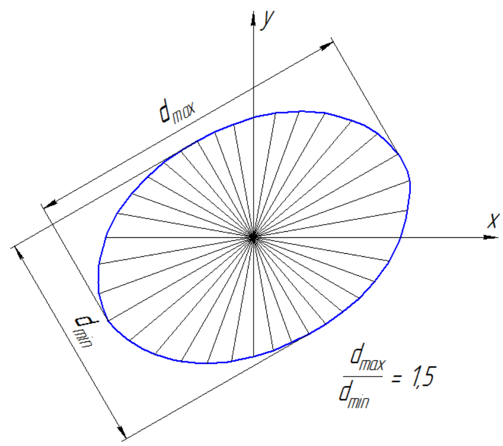

Fig. 4. The out-of-roundness of the cross section strain of the second crank pin

force *P* (Fig. 5). On the basis of the results obtained earlier, the second crank pin of the crankshaft, which is rotated by 120°, was accepted as a dangerous section. The results of the analysis are presented in Table 4.

Table 4. The cutting forces and strain of the crankshaft pin

P, (N)	$\delta\cdot10^5$, (mm)	$C\cdot10^{-5}$, (N/mm)
50	34,99	1,429
100	69,97	
150	105	
200	139,9	
250	174,9	
300	209,9	
350	244,9	
400	279,9	

It should be noted that the elastic strain of the crankshaft pins under the action of cutting forces are significant, which will not allow the necessary shape accuracy.

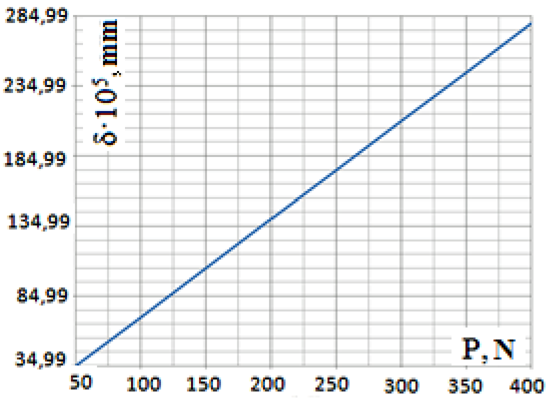

Fig. 5. Dependence of the cross section strain of the crank pin δ from the resultant cutting force P during grinding

Therefore, the normative condition to ensure the necessary accuracy of machining crankshafts is the use of grinding steady rests of various designs [17].

5. DEVELOPMENT THE CONSTRUCTION OF THE GRINDING STEADY REST

To improve the quality and accuracy of the machined connecting rod journal (crank pin), as well as to level the harmful effect of the cutting force on the above parameters, the original design of the following grinding steady rest was designed (Fig. 6).

The principle of action of the rest is shown in Fig. 7.

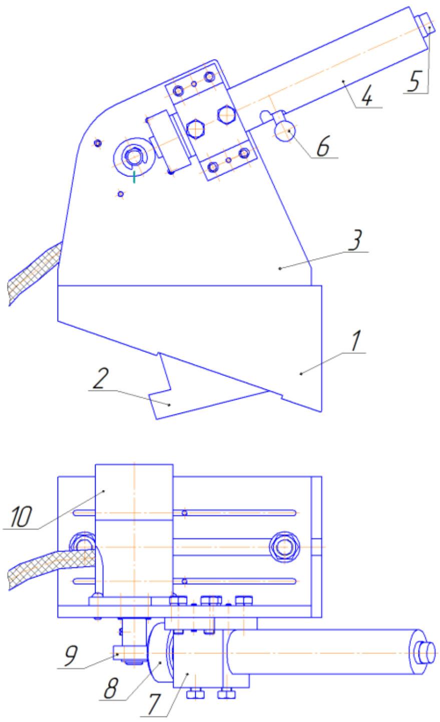

Fig. 6. Construction design of the following grinding steady rest: 1 – bearing plate; 2 – clamp strap; 3 – rest body; 4 – barrel; 5 – plunger; 6 – pin lock; 7 – flange; 8 – screw-nut; 9 – profile cam; 10 – servomotor.

The rest is fastened to the machine bed with the help of a clamp strap 2, installed from below the bearing plate 1. A movable rest body 3 is fastened on the bearing plate 1. A flange 7 with a barrel 4 and a servomotor 10 are mounted on the body. A preliminary retraction of the plunger 5 of the rest when changing the workpiece is carried out using a pin lock 6.

The main detail in the design of the reste is the profile cam 9 (Fig. 6).

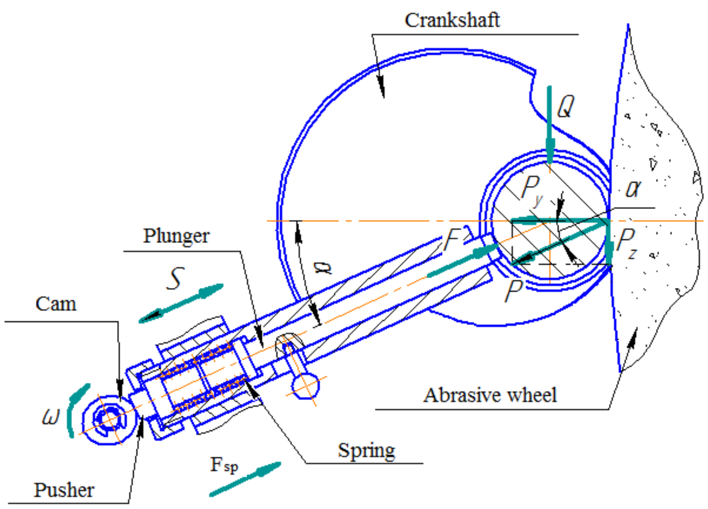

Fig. 7. Diagram of the work of the following grinding steady rest: *Py, Pz* – cutting force components; *P* - closing cutting force; *Q* - crankshaft weight; *F* - balancing force; *S* - plunger movement; F_{sp} – initial spring strain force; ω - angular velocity of the cam rotation; α – angle between the closing cutting force *P* and the vertical plane

The cam profile completely repeats the out-of-roundness profile of the crank pin strain depending on the grinding force applied to it (Fig. 4).

The cam rotates with the help of an servomotor 10, while moving the pusher, which through the spring moves the plunger 5 with the carbide plate. The plunger applies a balancing force *F* to the crankshaft pin thereby reducing its elastic strain. The return of the plunger to its original position, as well as the constant contact of the pusher with the cam provided by the spring. The spring stiffness and the initial pressing force are regulated by the screw-nut 8. An important condition for effective work of the rest is the establishment of the initial position of the cam with respect to the crank pin (connecting rod journal) and the subsequent rotation of them with the same frequency.

The angle α between the closing cutting force *P* and the vertical plane can be determined by the formula (11)

$$\alpha = \arctan\frac{P_z}{P_y}\,. \qquad (11)$$

An important step in the project is to determine the dimensions and spring stiffness.

The spring pressing force for maximum strain can be determined by the formula (12):

$$F_3 = \frac{F_2}{(1-\delta)}\,, \qquad (12)$$

where

F_2 – spring force at strain working;

δ – relative inertia gap.

The critical speed of displacement of the spring movable end is determined by formula (13):

$$V_c = \frac{\tau_3\cdot(1-\frac{F_2}{F_3})}{\sqrt{2\cdot G\cdot\rho}\cdot 10^{-3}}\,, \qquad (13)$$

where

τ_3 – maximum shear stress of the spring;

G – modulus of elasticity in shear;

ρ – dynamic material density.

The next step is to check the uptime condition for $1\cdot 10^4$ hours (14):

$$\frac{V_{\max}}{V_c} < 1\,, \qquad (14)$$

where

$V_{\max}$ – maximum conveying speed of the spring movable end.

The correctness of the spring choice will be the fulfillment of the condition (14).

And at the end to ensure the rigidity of the rest design and qualitative of the crankshaft processing, it is important to determine the spring stiffness (15):

$$c = \frac{F_2 - F_1}{h}\,, \qquad (15)$$

where

h – working stroke of the spring;

F_1 - force of prior deformation.

Adequate design calculation will allow to make the original construction of the following grinding steady rest.

Such an machine device will allow to automate the process of grinding the crankshafts taking into account the compensation of the problems of the technological process, which are caused by the peculiarities of the profile of the crankshaft component part. And also this rest will improve the accuracy of processing and reduce its cost.

CONCLUSIONS

The crankshafts of internal combustion engines are responsible and stressed parts that operate under the influence of dynamic loads. The operating conditions of the crankshafts and associated engine parts require precise performance of the dimensions and the correct mutual position of the individual elements. To meet such challenges the following has been accomplished:

- The considered methods of initial parameters and simulation modelling. The possibility to determine the deflections and rigidity of crankshafts in any section is proved. Under such arrangement to determine the optimum cutting parameters for the round infeed grinding cycle has been possible.
- The design of the following grinding steady rest is developed. This machine device allows to compensate the cutting force effect on the elastic strain of the crankshaft, depending on the change in its rigidity. It is considered that the rigidity varies depending on the angle of rotation of the part.

Further research is planned to priorities time with the modeling issues and parametric optimization of the following grinding steady rest design for this type of parts.

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