

Раздел 1  
МАШИНОСТРОЕНЕ

Section 1  
MECHANICAL ENGINEERING

**POWER CHARACTERISTICS OF THE WIDE-RANGE ECCENTRIC  
KEYLESS CLAMPING CHUCK**

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**Abstract**

The article represents the results of theoretical research of power characteristics of the wide-range eccentric keyless grip socket synthesized with application of genetic-and-morphological approach. Authors show that the emergence principle is realized using the universal genetic operators of crossing and crossover. According with the emergence principle the union of subsets of converters in a new set (hybrid structure) allowed to receive new properties – increase of rapidity and a range extension at decrease in diametrical dimensions.

**Keywords:** genetic algorithm, instrumental clamping chuck, genetic code, genetic-and-morphological approach, force of clamping.

**1. INTRODUCTION**

The wide-range eccentric keyless clamping chucks (CC) with two inputs (kinematic chains) and three tightening elements were synthesized (Fig.1) with application of genetics-morphological approach [7-10] to reduce the auxiliary time for change of the cutting tools with a cylindrical shaft (drills, mills, taps, etc.) in metal-cutting machines and electric drills [3, 4], and also to eliminate the set of tightening grips.

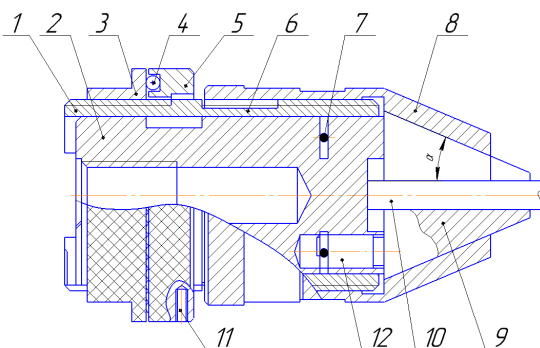


Fig. 1. The wide-range eccentric CC with two kinetic chains: 1 – tail part of a jacket; 2 – case; 3 –special nut; 4 – balls; 5 – ring; 6 – jacket; 7 – collar spring; 8 – the conical plug (a carving nut

with an internal cone); 9 – eccentric cam; 10 – tail end; 11 – lock screw; 12 – cylindrical tool-slide of an eccentric cam 9

These CC at the chromosomal level have a genetic code – a parental chromosome [9-12]  $M_{a1} - F_{r2}$  (the input is in the form of the moment  $M_{a1}$  around a rotation axis, and the output is in the form of the radial clamping force  $F_{r2}$ ). Taking into account two inputs their genetic formula at the population level taking into account the converters is as follows:

$$\begin{matrix} (M_{a1}^1 - SC^1) \\ (M_{a1}^2 - SC^2) \end{matrix} \text{ --- } \otimes \text{ --- } CB^1 - F_{r2} \quad (1)$$

where  $CB^1 = WD \times LV$  is a hybrid variant of the crossed converters (wedge-type  $WD$  and lever-type  $LV$ ) in one chuck. This clamping chuck realizes the emergence principal [1] using the universal genetic operators of crossing and crossover;  $SC^1$  and  $SC^2$  are the screw converters in the 1-st and 2-nd kinematic chains respectively.

The expansion of genetic formula of the synthesized CC for two power flows is:

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$$(M_{a1}^1 - SC^1 - WD \times LV - 3F_{r2}^1) + \quad (2)$$

$$+ (M_{a1}^2 - SC^2 - WD \times LV - 3F_{r2}^2)$$

**2. RESEARCH AT NONROTATING CHUCK**

The CC (Fig. 2) operates in 2 workflow phases [2,3]: 1-st phase – backlash adjustment between eccentric cam 9 tail end 10 and preliminary clamping with force  $F_{r2}^1$  due to rotation of collet 8 with the inside thread and cone  $2\alpha$ ; 2-nd phase – final clamping with force  $F_{r2}^2$  to the required force  $F_{r2}^1 + F_{r2}^2$  due to rotation of the special nut 3 on the left thread of the part 1 of the jacket 6 holding ring 5 fixed (with stop through balls 4) or together with the cone collet 8.

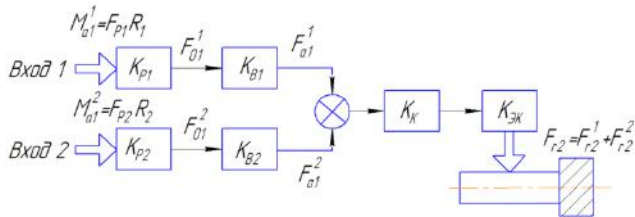


Fig. 2. Functional chart of CC with two force flows (2 inputs and 1 output)

At consecutive connection of chains, the strengthening coefficient of CC in the 1-st phase is as follows (1-st kinematic chain):

$$K_n^1 = \frac{F_{r2}^1}{F_{p1}^1} = K_{p1} \cdot K_{B1} \cdot K_K \cdot K_{\text{ЭK}} \quad (3)$$

The strengthening coefficient of CC in the 2-nd phase is as follows (2-nd kinematic chain):

$$K_{II}^2 = \frac{F_{r2}^2}{F_{p2}^2} = K_{p2} \cdot K_{B2} \cdot K_K \cdot K_{\text{ЭK}} \quad (4)$$

where  $K_{p1}$ ,  $K_{B1}$ ,  $K_K$ ,  $K_{\text{ЭK}}$ ,  $K_{p2}$ ,  $K_{B2}$  are respectively the strengthening coefficients of the next parts of chain: ‘conical plug 8 - jacket 6’; jacket 6 with strip-type screw; ‘conical plug 8 - eccentric cam 9’, leverage of

eccentric cam 9 with pivot axis by means of cylindrical tool-slide 12; ‘special nut 3 – part 1 of jacket 6’.

To evaluate each of strengthening coefficients  $K_{p1}$ ,  $K_{B1}$ ,  $K_K$ ,  $K_{\text{ЭK}}$ ,  $K_{p2}$ ,  $K_{B2}$ , the analytical models were used. (Fig. 3,4). The strengthening coefficient  $K_{p1}$  may be expressed from the following:

$$M_{a1}^1 = F_{p1}^1 \cdot \frac{D_B}{2} = F_{01}^1 \cdot R_1, \quad (5)$$

$$K_{p1} = \frac{F_{01}^1}{F_{p1}^1} = \frac{D_B}{2R_1}, \quad (6)$$

where  $D_B$  is the outside diameter of conical plug 8;  $R_1$  is the mid-radius of the square thread.

$K_{B1}$  strengthening coefficient may be expressed from the following:

$$F_{a1}^1 = F_{01}^1 \cdot \text{ctg}(\gamma + \varphi_1) \quad (7)$$

where  $\gamma = \text{arctg } t_{p1} / \pi \cdot d_{p1}$  – helix angle of strip-type screw of the jacket 6;  $t_p$  – pitch of thread;  $\varphi_1$  – friction angle in strip-type screw mating;  $d_{p1}$  – mid-radius of strip-type screw.

As it flows from the (7):

$$K_{B1} = \frac{F_{a1}^1}{F_{01}^1} = \text{ctg}(\gamma + \varphi_1) \quad (8)$$

From the Fig. 3 the balance of forces, acting in longitudinal section vertically on eccentric cam 9 is as follows:

$$F_{a1}^1 \cdot \text{ctg}(\alpha + \varphi_2) - F_{a1}^1 \cdot \text{tg} \varphi_3 - N_1 = 0 \quad (9)$$

$$F_{a1}^1 \cdot [\text{ctg}(\alpha + \varphi_2) - \text{tg} \varphi_3] = N_1,$$

where  $N_1$  – vertical lift component acting in contact point of eccentric cam 9 and plug 8 (point B);  $\varphi_2$  – friction angle in conical mating;  $\varphi_3$  – friction angle in flat contact of eccentric cam 9 and case 2.

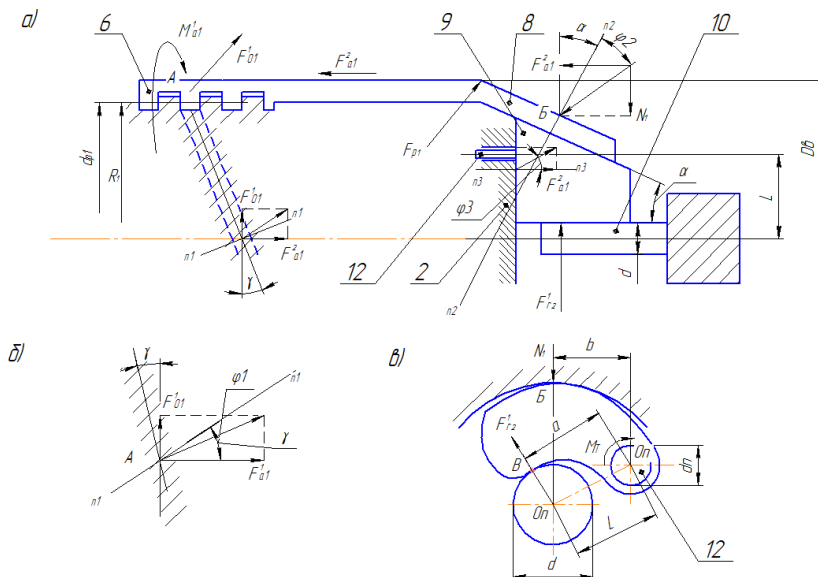


Fig. 3. Force flows design scheme at take-up of clearance and preliminary clamping (1-st phase): а – whole kinematic chain; б – strip-type screw 8 and jacket 6; в – when eccentric cam 9 interacts with tail end 10

As it flows from the (9):

$$K_k = \frac{N_1}{F_{a1}^1} = ctg(\gamma + \varphi_2) - tg\varphi_3. \quad (10)$$

From the Fig. 3 the balanced state of eccentric cam 9 at torques acting in plane normal to axis  $O_{II}$  of the cylindrical tool-slide 12 with  $d_p$  diameter:

$$N_1 \cdot b - F_{r2}^1 \cdot a - M_T = 0, \quad (11)$$

where  $a$  and  $b$  are respectively the arms of forces  $F_{r2}^1$  and  $N_1$  about the point  $O_{II}$ ;  $M_T = R_1 \cdot \frac{d_{II}}{2} \cdot f$  is the frictional torque in cylindrical tool-slide 12;  $R_1$  is the resulting force from  $F_{r2}^1$  and  $N_1$ , acting in the sliding contact of the cylindrical tool-slide 12 in the case 2;  $f$  is the coefficient of sliding friction of contact of the cylindrical tool-slide 12 in the case 2.

Considering the case of clamping the tool of the maximum diameter, when  $R_1 = N_1 - T_1$ , the expression (11) could be transformed as follows:

$$N_1 \cdot b - F_{r2}^1 \cdot a - (N_1 - F_{r2}^1) \cdot \frac{d_{II}}{2} \cdot f = 0. \quad (12)$$

As it flows from the (12)

$$K_{\Theta K} = \frac{F_{r2}^1}{N_1} = \frac{2b - d_{II} \cdot f}{2a - d_{II} \cdot f} \quad (13)$$

Considering  $d_{II} \cdot f \ll 2b$  and  $d_{II} \cdot f \ll 2a$ , we obtain

$$K_{\Theta K} \cong \frac{b}{a} \quad (14)$$

The gain constants  $K_{p2}$  and  $K_{B2}$  are defined similar to coefficients  $K_{p1}$  and  $K_{B1}$ :

$$K_{p2} = \frac{D_r}{2R_2} \quad (15)$$

$$K_{B2} = \frac{F_{a1}^2}{F_{o1}^2} = ctg(\beta + \varphi_4). \quad (16)$$

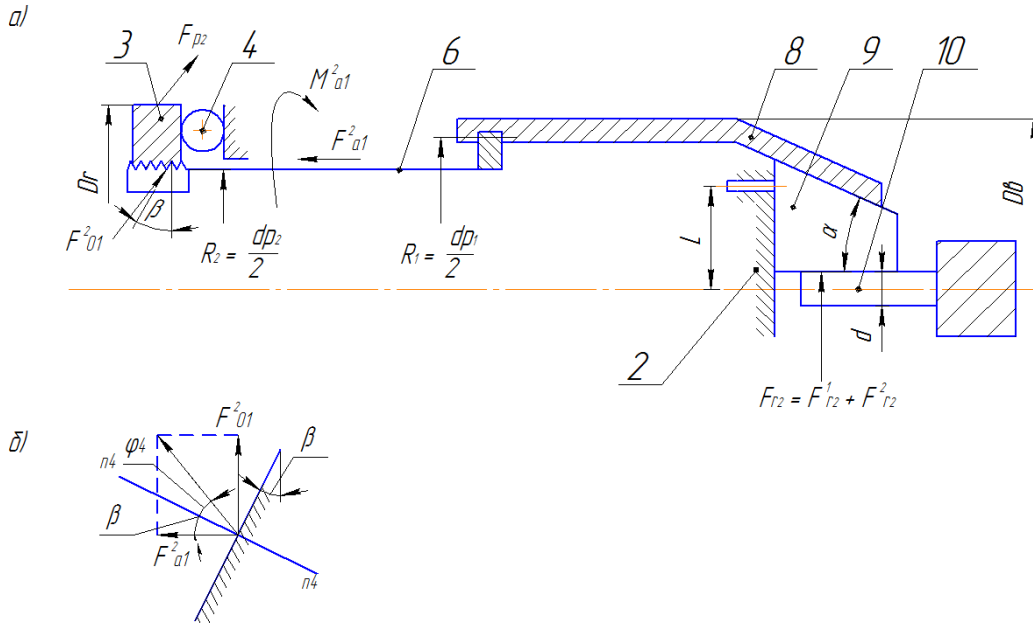


Fig. 4. Force flows design scheme at final clamping (2-nd phase): a – whole kinematic chain (special nut 3 – eccentric cam 9); b – additional screw (special nut 3 – jacket 6)

Substituting expressions (6), (8), (10), (14), (15), (16) in the expressions (3) and (4) there will be obtained:

$$K_{II1} = \frac{D_B}{2R_1} \cdot ctg(\gamma + \varphi_1) \cdot [ctg(\alpha + \varphi_2) - tg\varphi_3] \cdot \frac{b}{a}, \quad (17)$$

$$K_{II2} = \frac{D_r}{2R_2} \cdot ctg(\beta + \varphi_4) \cdot [ctg(\alpha + \varphi_2) - tg\varphi_3] \cdot \frac{b}{a}, \quad (18)$$

When  $\varphi_1 = \varphi_2 = \varphi_3 = \varphi_4 = \varphi$  and  $f = tg\varphi$  the resulting radial clamping force could be expressed as follows:

$$F_{r2} = F_{r2}^1 + F_{r2}^2 = K_{II}^1 \cdot F_{p1} + K_{II}^2 \cdot F_{p2} = K_{II}^1 \frac{2M_{a1}^1}{d_{p1}} + K_{II}^2 \frac{2M_{a1}^2}{d_{p2}} \quad (19)$$

or

$$F_{r2} = 2 \left[ \frac{M_{a1}^1}{d_{p1}} \cdot ctg(\gamma + \varphi) + \frac{M_{a1}^2}{d_{p2}} \cdot ctg(\beta + \varphi) \right] \cdot [ctg(\alpha + \varphi) - tg\varphi] \cdot \frac{b}{a} \quad (20)$$

### 3. RESEARCH AT THE ROTATING CHUCK

Let us consider the influence of CC rotation frequency on its power characteristics taking into account action of centrifugal forces of unbalanced clamping elements – cams 9 (Fig.5).

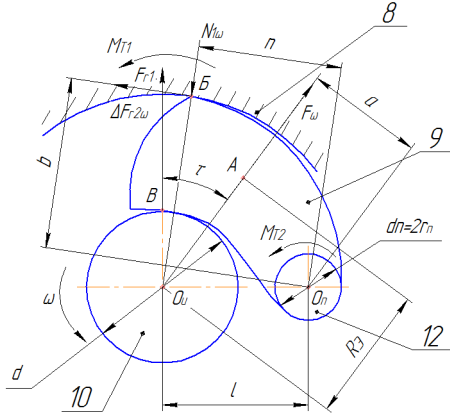


Fig. 5. Force flows design scheme of clamping element at rotation of CC

The cams 9 will have the balanced state in the case of  $n \approx l$ :

$$\sum M(O_n) = 0$$

$$-N_{1\omega} \cdot l + F_\omega \cdot a \cdot \cos \tau - M_{T1} - M_{T2} = 0, \quad (21)$$

where  $l$  is the distance between tool axis  $O_H$  and eccentric cam axis  $O_n$  (cylindrical tool-slide 12 with  $r_n$  radius);  $a$  – arm of centrifugal force from mass  $m_k$  of the cam 9, concentrated in its gravity centre (point A);  $\tau$  is the angle between centrifugal force  $F_\omega = m_k \cdot \omega^2 \cdot R_3$  and radial clamping force  $F_{r2\omega}$  in the contact point of eccentric cam with the tool surface (point B);  $M_{T1}$  is the frictional torque in point B when cam 9 rotates;  $M_{T2}$  is the frictional torque of the cylindrical tool-slide 12 in the case 2.

Considering the frictional coefficient equal, the frictional torques are:

$$M_{T1} = F_{T1} \cdot b = N_{1\omega} \cdot b \cdot f \quad (22)$$

$$M_{T2} = F_{T2} \cdot r_{II} = (F_\omega \cdot a \cdot \cos \tau - N_{1\omega}) r_{II} \cdot f \quad (23)$$

Substituting expressions (22), (23) into the (21) there will be obtained:

$$N_{1\omega} = F_\omega \cdot \cos \tau \cdot \frac{a(1 - r_{II} \cdot f)}{l + bl - r_{II} f}. \quad (24)$$

Considering the cam as absolutely rigid and taking into account the rigidity of the chuck  $C_{II}$ , the elastic displacement in the point B (Fig.5 and 6,a) is as follows:

$$\delta = \frac{N_{1\omega}}{C_{II}}. \quad (25)$$

During the displacement by amount of  $\delta$  in the contact point of the cam and tool (point B) the static radial clamping force, according to (19) and (20) and taking into account the rigidity of the tool  $C_{II}$  (Fig. 6,a), will be reduced by  $\Delta F_{r2\omega}$ :

$$\Delta F_{r2\omega} = \delta \cdot C_{II} \quad (26)$$

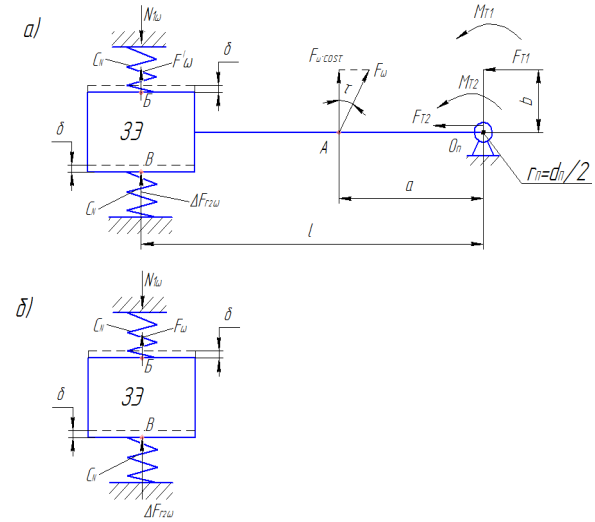


Fig. 6. Design schemes for comparing of reduction of dynamic clamping force in CC: a – with eccentric (wedge-lever-type) cam; b – with wedge-type cam

Substituting expressions (24), (25) into the (26) there will be obtained:

$$\Delta F_{r2\omega} = F_\omega \cdot \cos \tau \cdot \frac{a(1 - r_{II} f)}{l + bf - r_{II} f} \cdot \frac{C_{II}}{C_{II}} \quad (27)$$

Considering the fact that  $r_{II} f$  is essentially small quantity, the reduction of the radial clamping force is:

$$\Delta F_{r2\omega} = F_\omega \cdot \frac{C_{II}}{C_{II}} \cdot \frac{a}{l + bf} \cdot \cos \tau \quad (28)$$

In the case of wedge-type cam without pivot axis (Fig.6,b):

$$\Delta F_{r2\omega}^I = F_\omega \cdot \frac{C_{II}}{C_{II}} \quad (29)$$

Since  $a < l + bf$  and  $\cos \tau < 1$ , the reduction of the radial clamping force in rotating wide-range eccentric keyless CC is far less. For instance, when  $C_{II} = C_{II}$ ,  $l + bf = 2a$  and  $\cos \tau = 45^\circ \approx 0.7$

$$\frac{\Delta F_{r2\omega}^I}{\Delta F_{r2\omega}} = \frac{1 \cdot 0.7}{2} = 0.35, \text{ i.e. 3-times less. That means,}$$

that comparing with standard wedge-type drill chucks, the maximum rotating speed could be increased more then 3-times. The earlier studies [5,6] and calculations for CC with the 16 mm maximum diameter of tool (Fig.7) proved this statement.

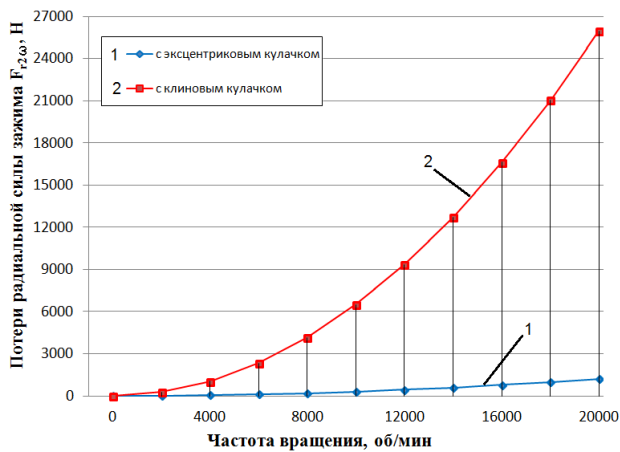


Fig. 7. The dynamic clamping force reduction versus rotating speed response characteristic

## CONCLUSIONS

The research showed that besides additional increase of clamping force by introduction of the second kinematic chain with the screw converter, the application of universal genetic operators of crossing and a crossover allows to realize the emergence principle (the integrity principle) [1,11,12], i.e. to obtain new properties. Such new properties are: reduction of influence of centrifugal forces caused by misbalanced clamping elements, increasing of the range and decreasing of diametric dimensions.

It is reasonable to use the synthesized wide-range eccentric keyless clamping chucks at high-speed processing on drilling, cutting and multi-functional CNC-machines, as well as on parallel kinematic structure machines.

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