Determination of design parameters of block linkage mechanism of the drive of machine for processing of details with the compound motion of working reservoir

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Abstract
Significant amount of fine polymeric parts of various industries including finding products of consumer industry are formed by mechanical operation. After that considerable microroughnesses and lack of luster are observed in the surface of products. Such details undergo finishing operation, which consists in grinding and polishing of their surface with loose abrasive in the form of granules in the working reservoir which carries out compound spatial motion.
Today the urgent task is intensification of treatment of polymeric products surface with loose abrasive in the form of granules; it can occur due to rationally established nature of the granular medium motion in the working reservoir, and in turn, is provided with the machine drive.
Synthesis of the hinged four-link block linkage mechanism of the 2nd class, the 2nd order and the 2nd type is performed. It is a part of the machine drive for reproduction of the necessary law of change of angular speed and carrying out further kinematic analysis in SAPR SolidWorks. Consistent patterns of impact of design data of the mechanism on reproduction of the law of change of angular speed of the driven crank are determined. On the basis of synthesis of block linkage mechanism, rational ratios of lengths of links are established, dependences allowing to create mechanisms of various large-scale standard sizes are obtained. Thus, machine drive, where block linkage mechanism is used, is capable to implement the law of change of angular speed in driving

shaft of machine in order to create conditions for ensuring grinding and polishing with high quality of polymeric products surface.

The suggested drive design with block linkage mechanism can be applied in machines with compound spatial motion of working reservoir for implementation of necessary technological modes that, in turn, will provide implementation of grinding and polishing of parts with high quality.

Key words: BLOCK LINKAGE MECHANISM, LAW OF CHANGE OF ANGULAR SPEED

Problem statement

It is necessary to implement the cascade mode of the granular medium movement, to create conditions under which dynamic load of working medium will decrease for the purpose of possibility of use of the equipment with compound spatial motion of working reservoir for technological processes of grinding and polishing of polymeric products surface with loose abrasive in the form of granules. Collision of details with walls of working reservoir occurs at lower kinetic energy of their motion; moreover, the products will move between opposite end faces of working reservoir with an equal intensity in both directions. It is known that it is possible to provide such nature of movement of working medium only if the driving shaft is provided with pre-established cyclically uneven law of change of angular speed [1]. The special drive [2], in a design of which the block linkage mechanism is used, can reproduce the necessary law of change of angular speed. However, in the paper [2], all design data of the machine drive are not held, mathematical dependences, according to which it is possible to determine necessary ratios of lengths of links of the block linkage mechanism, are not provided.

Analysis of the last researches and publications

The analysis of the published papers on means and methods of processing of details by loose abrasive in the form of granules has shown that papers of foreign scientists [3-5] on research of processes of mixing of loose substances in such equipment with the compound spatial motion of working reservoir are known. However, in the equipment used for mixing, the drive special designs providing to the driving shaft with the cyclic law of change of angular speed are not applied. The drive shaft of such machines rotates with uniform angular speed and, thus, conditions providing fast and intensive mixing of loose fine substances are formed. But such nature of motion is unsuitable for processes of grinding and polishing.

The equipment with the compound spatial motion of working reservoir [6, 7, 8] is also known; due to special designs of drive it is adapted to implementation of high-quality processes of grinding and polishing. However, all types of such equipment are original on the structure and have distinctions in calculation of design data.

Information on detailed calculation of design data of the drive of equipment according to source [2] is illustrated superficially and requires more detailed research and the analytical argument.

Main results of research

The machine with the compound spatial motion of working reservoir is equipped with a special design of the drive with block linkage mechanism. Its model is presented in Figure 1.

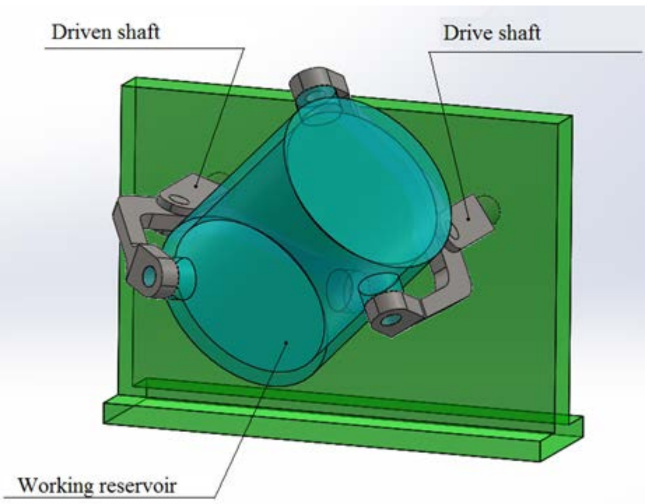

Figure 1. The machine for processing of details with the compound spatial motion of working reservoir

The spatial kinematic scheme of the machine with the drive is presented in Figure 2.

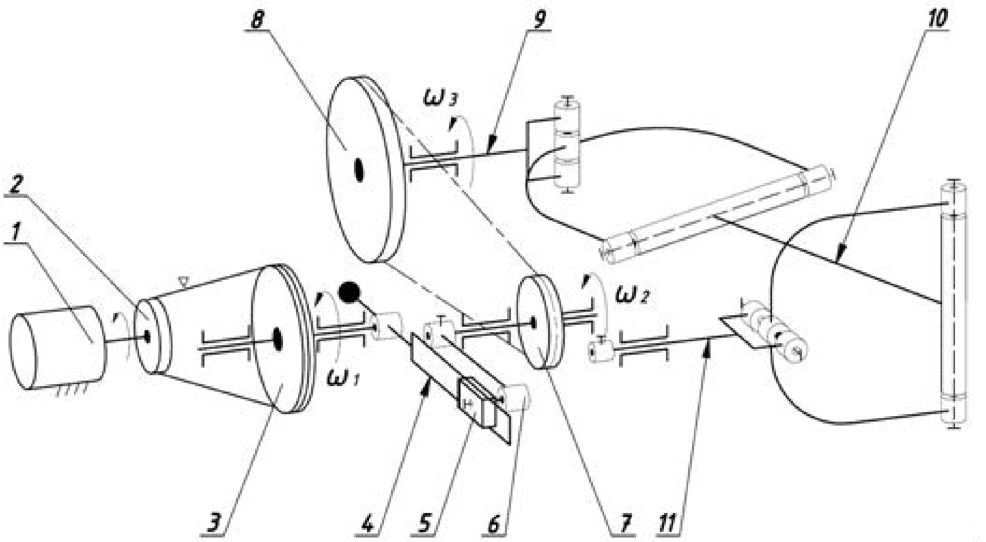

Figure 2. The kinematic scheme of equipped with the special drive machine for processing of details

Where: 1 - electric motor, 2 - driving pulley of belt drive, 3 - driven pulley of belt drive, 4 - driving crank, 5 - slide block, 6 - driven crank, 7 - driving sprocket, 8 - driven sprocket, 9 – shaft, which is both driven shaft of the drive and driving one, 10 - conventional value of working reservoir, 11 - driven shaft of machine.

It is known that the law of change of angular speed can be implemented due to block linkage mechanism of the 2nd class, the 2nd order and the 2nd type (1cl → 2cl 2od 2t) with the driving link of crank [9] which block diagram is provided in Figure 3.

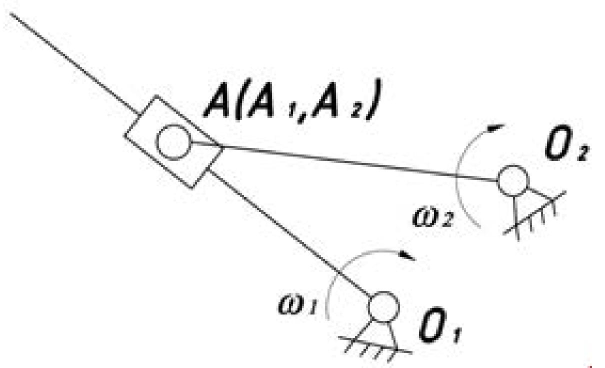

Figure 3. Structure diagram of block linkage mechanism

In the scheme of the mechanism, the following elements are presented: O_1A - driving crank, O_2A - driven crank, A - slide block, A_1 - sliding kinematic couple, A_2 - rotating kinematic couple, O_1, O_2 - centers of rotation of the corresponding cranks.

The block linkage mechanism, which is a part of drive, is synthesized by means of establishment of all rational ratios of its lengths of links so that the necessary law of change of angular speed could be implemented. For this purpose, the kinematic scheme of the block linkage mechanism with randomly chosen sizes of links in 12 states is built. It is provided in Figure 4.

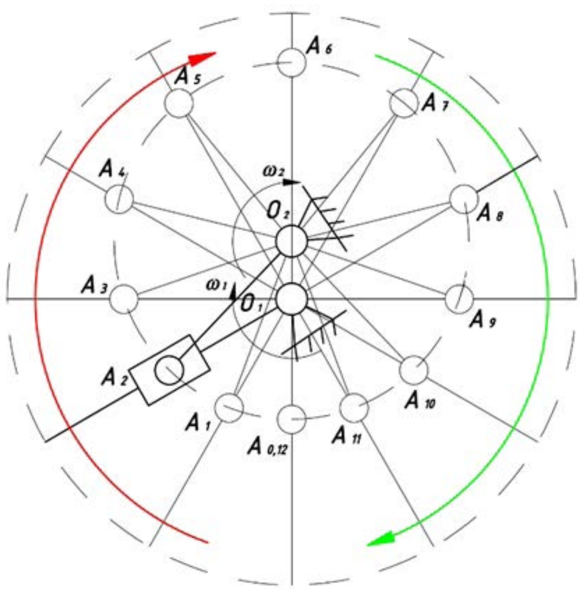

Figure 4. Kinematic scheme of block linkage mechanism

In case of synthesis of the block linkage mechanism, it is necessary to consider previously obtained data in paper [1].

It is necessary to provide implementation of the law of change of angular speed in driving shaft of machine:

$$\omega_3 = \omega_1 - \left(\frac{\omega_1}{3}\right)\sin\left(2\varphi + \frac{\pi}{2}\right) \qquad (1)$$

where ω_1 − uniform angular speed of the driving crank which is arithmetic-mean value of the law of change of angular speed ω_3 in driving shaft of machine, φ − angle of rotation of the driving shaft of machine.

According to expression (1), the law of change of angular speed in the driving shaft of machine should have two periods, that is for one turnover of the driving shaft of machine, value of angular speed will reach the minimum values 2 times and will reach maximum values 2 times.

And as block linkage mechanism is capable to provide the law of change of angular speed only with one period, additional application of gear transmission with gearing ratio *i=2:1* is necessary. It is also necessary to consider that gear transmission with gearing ratio *i=2:1* will reduce amplitude values of angular speed of driven crank twice, thus, it is necessary that the law of change of angular speed of the driven crank had extrema which are increased twice. Thus:

$$\omega_3^{MAX} = 2\omega_2^{MAX} \tag{2}$$

where ω_3^{MAX} − amplitude (maximum) value of angular speed in driving shaft of machine; ω_2^{MAX} − amplitude (maximum) value of angular speed in the driven crank.

Analyzing the kinematic scheme shown in Figure 3, it is evident that mechanism positions 6 and 12 (0) will correspond to extrema of angular speed in the driven crank. As the distance O_1A_6 is the largest, angular speed in this position will be maximum, and vice versa, as the distance O_1A_{12} is the smallest, and angular speed in this position will be minimum. The distance between cranks rotation centers O_1O_2 will affect the amplitude values of angular speeds.

Instant value of angular speed of the driven crank will be determined as:

$$\omega_2 = \frac{\overline{V}_{O_2A}}{l_{O_2A}} \tag{3}$$

where $\overline{V}_{O_2A}$ - linear speed of point A around point O_2; l_{O_2A} - length of crank O_2A;

Moreover, linear speed $\overline{V}_{O_2A}$ will be equal to linear speed $\overline{V}_{O_1A}$ in each position of mechanism:

$$\overline{V}_{O_2A} = \overline{V}_{O_1A} \tag{4}$$

Let us determine the linear speed $\overline{V}_{O_1A}$:

$$\overline{V}_{O_1A} = \omega_1 l_{O_1A} \tag{5}$$

where ω_1 - value of uniform angular speed of rotation of crank which is rigidly connected to the driven pulley;

l_{O_1A} - distance from the center of rotation of crank to the position of kinematic couple of sliding block for any position of the mechanism. l_{O_1A} is possible to be determined on the basis of the theorem of cosines considering the conditional triangle O_1O_2 formed by links of block linkage mechanism in each position:

$$l_{O_1A}^2 = l_{O_1O_2}^2 + l_{O_2A}^2 - 2l_{O_1O_2}l_{O_2A}\cos(l_{O_1O_2}{}^\wedge l_{O_2A}) \tag{6}$$

$$l_{O_1A} = \sqrt{l_{O_1O_2}^2 + l_{O_2A}^2 - 2l_{O_1O_2}l_{O_2A}\cos(l_{O_1O_2}{}^\wedge l_{O_2A})} \tag{7}$$

Let us substitute value of expression (7) into the equation (5):

$$\overline{V}_{O_1A} = \omega_1\sqrt{l_{O_1O_2}^2 + l_{O_2A}^2 - 2l_{O_1O_2}l_{O_2A}\cos(l_{O_1O_2}{}^\wedge l_{O_2A})} \tag{8}$$

Let us substitute value of expression (8) in a formula (3) and we obtain the general equation for determination of instantaneous value of angular speed of the driven crank ω_2:

$$\omega_2 = \frac{\omega_1\sqrt{l_{O_1O_2}^2 + l_{O_2A}^2 - 2l_{O_1O_2}l_{O_2A}\cos(l_{O_1O_2}{}^\wedge l_{O_2A})}}{l_{O_2A}} \tag{9}$$

For determination of distance l_{O_1A} in the 6th position of mechanism, it is possible to write down the following expression:

$$l_{O_1A} = l_{O_1O_2} + l_{O_2A} \tag{10}$$

Let us substitute the value of formula (9) into expression (4):

$$\overline{V}_{O_1A} = \omega_1(l_{O_1O_2} + l_{O_2A}) \tag{11}$$

Taking into account the equality of linear speeds (4), the equation for determination of maximum value of angular speed of the driven crank in the 6th position of the mechanism due to substitution of formula (11) into the expression (3) will be of the form:

$$\omega_2^{MAX} = \frac{\omega_1(l_{O_1O_2} + l_{O_2A})}{l_{O_2A}} \tag{12}$$

We can set the following basic data: the arithmetic average value of law of change of angular speed ω_1, necessary maximum value of angular speed of the driven crank ω_2^{MAX}, and also can randomly accept the length of the driven crank l_{O_2A}. Further, we can express the value $l_{O_1O_2}$ from the equation (12) and, having substituted all output data in expression (13), we obtained length of interaxle mechanism for ensuring the maximum value of angular speed in the 6th position of the mechanism in the driven crank.

$$l_{O_1O_2} = \frac{\omega_2^{MAX} l_{O_2A}}{\omega_1} - l_{O_2A} \qquad (13)$$

However, it is necessary to be convinced that the minimum value of angular speed of the driven crank in the 12th position of mechanism will be provided at such interaxle distances. Thus, the expression for determination of minimum instantaneous angular velocity of the driven crank of the mechanism in the 12th position is developed by similar principle.

Similar to expression (3) instantaneous minimum value of angular speed of the driven crank will be determined as:

$$\omega_2^{MIN} = \frac{\overline{V}_{O_2A}}{l_{O_2A}} \qquad (14)$$

Let us determine the linear speed:

$$\overline{V}_{O_2A} = \overline{V}_{O_1A} = \omega_1 l_{O_1A} \qquad (15)$$

Let us determine the distance l_{O_1A} for 12th (0th) position of mechanism:

$$l_{O_1A} = l_{O_2A} - l_{O_1O_2} \qquad (16)$$

Let us substitute the value of expression (16) into (15), we obtain:

$$\overline{V}_{O_1A} = \omega_1 (l_{O_2A} - l_{O_1O_2}) \qquad (17)$$

Let us write down the expression (14) considering (16):

$$\omega_2^{MIN} = \frac{\omega_1 (l_{O_2A} - l_{O_1O_2})}{l_{O_2A}} \qquad (18)$$

Let us establish in which positions of the mechanism the angular speed of driven crank coincides with value of constant angular speed of the driving crank.

On the basis of expression (4) and the fact that in such positions of the mechanism the angular speed of the driven crank must be equal to constant angular speed of the driving crank, it is possible to draw a conclusion that in these two positions of the mechanism distance from the center of rotation of crank to sliding block will be equal to length of the driving crank, $l_{O_2A} = l_{O_1A}$.

Correspondingly, having used the expression (7), it is established that it will be the third and ninth positions of the mechanism when the crank holds the horizontal position. And consequently, the angular speed of the driven crank at clockwise rotation will increase from zero position of the mechanism and will reach the maximum value in the position 6, and also, at rotation from position 6 angular speed will decrease and will reach the minimum in position 12. Such regularity will be also correct at counterclockwise rotation. According to the phase, angle of acceleration of the driven crank will be equal to a phase angle of deceleration:

$$\alpha_{aceleration} = \alpha_{deceleratin} = \frac{360^\circ}{2} = 180^\circ \qquad (19)$$

Besides, in case of rotation of the driven crank in any direction when passing of points, which coincide with positions of mechanism 3 and 9, its instantaneous angular speed will coincide with the constant angular speed of the driving crank.

For check of correctness of accomplishment of all previously developed expressions, we will consider a specific case by acceptance of amplitude values of established law of angular speed change (1) of driving shaft of machines $\omega_3 = [4{,}2 \div 8{,}4] rad/s$. According to results obtained in paper [1], the arithmetic mean value should correspond to the uniform angular speed with which the driving crank rotates $\omega_1 = 6{,}3 rad/s$. However, considering existence of gear transmission with gearing ratio *i=2:1* in the drive, the angular speed of driven crank must change in limits $\omega_2 = [\omega_2^{MIN} \div \omega_2^{MAX}] = [8{,}4 \div 16{,}8] rad/s$. At the same time, it sets uniform rotation to the driving crank with an angular speed $\omega_1 = 12{,}6 rad/s$, and it is rigidly connected to the driven pulley. Besides, let us accept distance $l_{O_2A} = 20mm$. Let us substitute all necessary data into expressions (13) and (18):

$$l_{O_1O_2} = \frac{\omega_2^{MAX} l_{O_2A}}{\omega_1} - l_{O_2A} = \frac{16{,}8 \cdot 20}{12{,}6} - 20 = 6{,}67mm$$

$$\omega_2^{MIN} = \frac{\omega_1 (l_{O_2A} - l_{O_1O_2})}{l_{O_2A}} = \frac{12{,}6(20 - 6{,}67)}{20} = 8{,}4 rad/s$$

Thus, with the link length $l_{O_2A} = 20mm$, the interaxle distance $l_{O_1O_2} = 6{,}67mm$, providing achievement of necessary extreme values ω_2^{MIN} and ω_2^{MAX} of the law of change of angular speed in the driven crank was determined and, as a result, the necessary law of change of angular speed in driving shaft of machine will be implemented.

As we have accepted the length of link $l_{O_2A} = 20mm$ for the initial parameter of counting of lengths of all links of block linkage mechanism, there is a need of test of the mechanism for the maximum allowable value of angle of pressure Θ in the kinematic couple A. In order to avoid blocking of the flat hinged mechanism, extreme value of angle of pressure [10] in any kinematic couple should not exceed 60 °, and for long operation of the mechanism the angle of pressure should not exceed 40 °.

Proceeding from these reasons, let us develop the expressions for determination of angle of pressure in kinematic couple A. It is known that pressure angle in kinematic couple is an angle between vector of force and vector of speed of the mechanism links, which are interconnected by the corresponding kinematic couple. In our case, the vector of force is directed perpendicularly to link O_1A, and a speed vector is perpendicular to the link O_2A. Thus, pressure angle in each position of the mechanism corresponds to O_2AO_1 angle formed by mechanism links.

According to the theorem of sine we can write down:

$$\frac{O_1O_2}{\sin(l_{O_2A}{}^\wedge l_{O_1A})} = \frac{O_2A}{\sin(l_{O_2O_1}{}^\wedge l_{O_1A})} \quad (20)$$

From the theorem (20) we can express value of angle of pressure Θ:

$$\Theta = \arcsin(l_{O_2A}{}^\wedge l_{O_1A}) = \frac{O_1O_2 \sin(l_{O_2O_1}{}^\wedge l_{O_1A})}{O_2A} \quad (21)$$

Having analyzed a formula (21), we have established that the maximum values of pressure angle can arise in the 3rd and 9th positions of the mechanism. Test according to the formula (21) has shown that pressure angles in such positions are equal.

Therefore, randomly chosen parameter of length of link O_2A corresponds to conditions [10] of operability of the hinged mechanism.

Besides, let us check our theoretical hypotheses by kinematic analysis of the block linkage mechanism with all previously determined ratios of lengths of links in CAD SolidWorks. In Figure 5, the diagram of dependence of angular speed of the driven crank on an angle of rotation of the driving crank is presented.

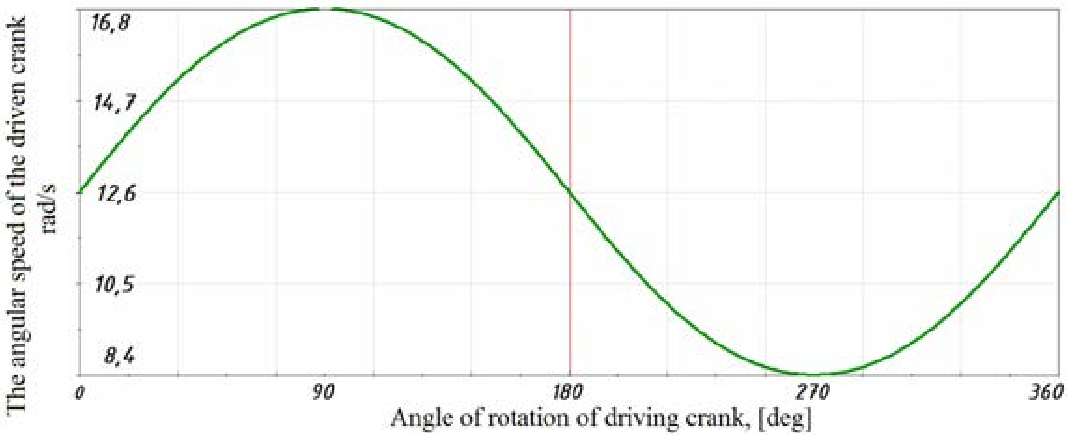

Figure 5. The diagram of dependence of angular speed of the driven crank on an angle of rotation of the driving crank

After substitution of all data in expressions (12,18) and the analysis of the diagram of the law of change of angular speed of the driven crank, which is provided in Figure 4, it is possible to draw a conclusion that the block linkage mechanism of the 2nd class, the 2nd order and the 2nd type (1cl → 2cl 2od 2t) completely provides reaching of instantaneous maximum and minimum values the law of change of angular speed on the driven mechanism crank. Moreover, the phase angle of slowdown of the driven crank is equal to the phase angle of acceleration. Thus, in case of additional application of chain gearing with the gearing ratio *i = 2:1* in driving shaft of machine, the sinusoidal law of change of angular speed must be implemented.

One typical size of linkage block mechanism with the set ratios of lengths of links that is a part of the machine drive is capable to implement the law of change of angular speed in the driven crank for various values of fixed angular speed of the driving crank. This could easily be verified due to permanence of lengths of links of the mechanism after substitution of other numerical amplitude values of the established law of change of angular speed in expressions (12), (13) and (18); these values are different from specific cases, which are considered.

Moreover, proportional change of all lengths of links of the block linkage mechanism at the same angular speed of the driving crank does not influence the strain of law of change of angular speed of the driven shaft. The law is invariable regardless of proportional change of lengths of all links of block linkage mechanism. Such statement can be confirmed due to substitution of the changed values of lengths of links of the mechanism expressions (12) and (18) in proportion, then the maximum and minimum value of law of change of angular speed will be invariable. Thus, the drive is universal construction and can provide the set operating mode of machine for processing of details with compound motion of working reservoir at any angular speed and various geometrical parameters of the machine.

Conclusions

1. Design and principle of operation of the special drive, where the block linkage mechanism of machine is used for processing of details with compound motion of working reservoir, were analyzed.

2. The block linkage mechanism, which is a part of drive and provides implementation of the cyclic sinusoidal law of change of angular speed with necessary amount of the periods for one turnover of the driving shaft of machine, was synthesized.

3. Consistent patterns of impact of design data of block linkage mechanism for reproduction of the law of change of angular speed of the driven crank were determined.

4. The obtained dependences allow creation of the mechanism of any size in case of identical scale of all elements of design.

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