Feed rate evaluation of mechanical toothed hopper-feeding device with ring orientator for parts, asymmetric at the ends

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Abstract. The paper considers an advanced design of a mechanical toothed hopper-feeding device with a ring orientator designed for automatic feeding of ax symmetric parts in the form of bodies of rotation, asymmetric at the ends, into technological machines. A new theoretical approach to the mathematical description of the feed rate of the mechanical toothed hopper-feeding device is proposed. This theoretical approach allows a correct assessment of the feed rate for various parts at the design stage of the technological machine, as well as to justify the rational parameters of the hopper-feeding device, in which its feed rate will be maximum.

1. Introduction
Mechanical hopper-feeding devices are designed for automatic feeding of piece parts in a given oriented position and with the required feed rate in technological machines, for example, for the assembly of multi-element products. The peculiarity of the functioning of mechanical hopper-feeding devices is that the process of piece-by-piece capture of parts by moving gripping organs from the total mass located in the hopper of the device is probabilistic.

In the majority of known works of domestic [1, 2] and foreign scientists [3, 4] mathematical descriptions of processes of piece capture of parts by moving gripping organs in mechanical hopper-feeding devices were under construction on the basis of laws of classical mechanics. As a result, the obtained mathematical dependences allowed us to estimate only the boundary values of the kinematic parameters of the moving gripping organs and the feed rate of the hopper-feeding devices. Therefore, to date, the main methods of studying the processes of functioning of mechanical hopper-feeding devices remain experimental studies based on the application of methods of probability theory, mathematical statistics and the theory of planning experiments [5]. They allow us to obtain empirical dependences of the device feed rate on the speed of its gripping organs and a limited number of physical and mechanical parameters of parts in the form of regression models.

For the first time, a theoretical approach to the mathematical description of the processes of capturing parts by moving gripping organs in mechanical hopper-feeding devices based on the theory of probability was presented in the domestic monograph [6]. The probability of capturing a part by moving gripping organs was described by the product of "conditional probabilities of transition" of the part from an arbitrary position in the hopper of the device to a position favorable for capturing and the absence of interference from other parts. A common drawback of the models was the incorrect description of the effect of the speed of the gripping organs on the probability of capturing the parts, which made it difficult for their practical application.

Based on the results of well-known works, the authors proposed a new theoretical approach, which consists in the combination of classical mechanics methods and probability theory, which allowed us to obtain the correct mathematical description of the process capture the parts of moving gripping organs and the feed rate of various designs of mechanical hopper-feeding devices [7 – 9].

This article describes the application of the proposed approach to the mathematical description and feed rate evaluation of the advanced design of the mechanical toothed hopper-feeding device with a ring orientator for axisymmetric parts in the form of bodies of rotation, asymmetric at the ends.
2. Formulation of the problem
In Figure 1 sketches are presented with the designation of geometric parameters of typical axisymmetric parts of the form of bodies of rotation with asymmetric ends: a conical (Figure 1, a) or spherical (Figure 1, b) end.

![Figure 1. Sketches of typical axisymmetric parts of the form of bodies of rotation with a conical (a) or spherical (b) end: l – the length of the part; \(d_1\), \(l_1\) – the diameter and the length of the cylindrical end of the part; \(x_c\) – the distance from the cylindrical end of the part to its center of mass; \(d_2\) – the diameter of the conical end of the part; \(r\) – the radius of the spherical end of the part.]

The scheme of the advanced design of the mechanical toothed hopper-feeding device with a ring orientator is shown in Figure 2. The main structural elements of the hopper-feeding device are a hopper, consisting of a shell 4 and a fixed base 1, and a rotating disk 5 with gripping organs in the form of pockets. Pockets are formed by teeth 3, evenly spaced along the circumference of the disk 5 (Figure 2, a).

![Figure 2. The scheme of the advanced design of the mechanical toothed hopper-feeding device with a ring orientator: a – a general view in the section; b, c – schemes capture and passive orientation of parts with a conical (b) or spherical (c) end: 1 – fixed base; 2 – ring orientator; 3 – teeth; 4 – shell; 5 – rotating disk; 6 – part.]

New in the advanced design of the mechanical toothed hopper-feeding device is that on the fixed base 1 under the rotating disk 5, a ring orientator 2 is installed. The groove of the ring orientator 2 repeats the profile of the asymmetric end of the part 6 (Figure 2, b, c) and has a different depth in the direction of rotation of the disk – from \(\frac{1}{4}\) of the height \(h = l - l_1\) of the asymmetric end of the part in the lower part of the hopper to the height of this end at the top of the hopper [10].

The device works as follows. Parts are filled into the hopper on the surface of the rotating disk.
During rotation of the disc parts are stirred and sink into the pockets between the teeth. If the parts
sink into the pockets of the asymmetric end down, they are immersed this end of the groove of the ring
orientator and securely held in the pockets (see Figure 2, b, c). During the rotation of the disc, they are
moved by pockets and teeth to the receiver (Figure 2 not shown) and fall out of the pockets into the
receiver. If the parts sink into the pockets with an asymmetric end up, they are supported by a
cylindrical end on the upper edge of the ring orientator and when moving the disk to the upper part of
the hopper fall out of the pockets under the influence of gravity back to the lower part of the hopper.

Thus, in this feeding device, a so-called passive method of orienting the parts is implemented, as a
result of which only the parts that are sunk by an asymmetric end into the groove of the ring orientator
are issued to the receiver, and the sunk by the cylindrical end are returned to the hopper.

The main characteristic of the hopper-feeding device is its feed rate. The construction of a correct
mathematical model of the feed rate of the hopper-feeding device, taking into account the interrelated
effect on its feed rate of the main kinematic and structural parameters of the device, geometric and
physical and mechanical parameters of the feed parts, will allow at the design stage of the
technological machine to justify the choice of rational parameters of the hopper-feeding device,
providing a given feed rate of the machine.

3. Theoretical part
Feed rate $F$ [parts / min] mechanical toothed hopper-feeding device is determined by the known
formula \[1, 2\]

$$F = knE,$$  \hspace{1cm} (1)

where $k$ – is the number of pockets; $n$ – the rotational frequency of the disk [rpm]; $E$ – capture factor
or efficiency.

Using the empirical expression for the coefficient $E$ obtained in [1] on the basis of statistical
processing of experimental data on the feed rate of more than 20 different designs of mechanical
hopper-feeding devices, the authors proposed the following formula

$$E = E_{\text{max}} \left[1 - \left(\frac{n}{n_{\text{max}}}\right)^4\right],$$ \hspace{1cm} (2)

in which the maximum value of the efficiency $E_{\text{max}}$ corresponds to the rotational frequency of the
disk with gripping organs close to zero, and the maximum value of the rotational frequency of the disk
$n_{\text{max}}$ corresponds to the condition that it is impossible to capture the parts by the gripping body. As
such a boundary condition, the phenomenon of collision of a part with the guide elements of a rotating
disk is considered, which leads to the ejection of the part from the pocket [7].

The product of two probabilities determines the maximum value of the efficiency in accordance
with the work [6]: $p_1$ – is characterizing the position of the part on the surface of the rotating disk,
favorable for capture, and $p_2$ – characterizing the absence of interference with the capture of other
parts

$$E_{\text{max}} = p_1 p_2.$$

A secure grip of the part is carried out by sinking its asymmetric end in a pocket (see Figure 2, b,
c). Therefore, the following positions of the part on the surface of the rotating disk will be favorable
for gripping: 1 – asymmetric end down, 2 – generating of cylindrical surface of the part (see Figure 1).
These possible positions of the part are characterized by probability $p_{11}$. To sink into the pocket from
the second position of the part must be rotated to the pocket asymmetric end, the possibility of which
characterizes the probability $p_{12}$. Using the well-known theorem of probability theory about the
probability of a complex event and by analogy with the work [6], we obtained an expression for
determining the probability $p_1$ in the form

$$p_1 = 1 - (1 - p_{11} p_{12_{\text{max}}})^3 (1 - p_{11} p_{12_{\text{min}}})^{2-3},$$ \hspace{1cm} (4)
where $p_{12\min}$ and $p_{12\max}$ -- is the minimum and maximum probability values $p_{12}$:

$$z = \frac{\pi(R - d_1) + 2R}{2l}$$ -- the maximum number of parts that can be placed in the capture zone (on the periphery of the disk at the bottom of the hopper); $R$ -- the radius of the rotating disk along the axis of the gripping organs (pockets).

The table 1 shows the mathematical formulas obtained by the authors. They describe the probabilities of expression (4) for two types of axisymmetric parts in the form of bodies of rotation: with a conical (see Figure 1, a) and spherical end (see Figure 1, b). The following symbols are accepted in the formulas of the table: $B = d_1 + \Delta t$ -- is the distance between the teeth or the width of the pocket; $\Delta t$ -- the gap between the part and the wall of the pocket, taken when designing in the range $0.05d_1 \leq \Delta t \leq 0.15d_1$; $\alpha$ -- the angle of inclination of the base and the rotating disk with gripping organs to the horizontal (see Figure 2, a); $\mu$ -- coefficient of friction sliding parts on the surface of the elements of the toothed hopper-feeding device.

**Table 1. Formulas to determine probabilities of expression (4)**

| Probability | Part with conical end (see Figure 1, a) | Part with a spherical end (see Figure 1, b) |
|-------------|----------------------------------------|------------------------------------------|
| $P_{11}$    | $p_{11} = \frac{1}{2} \frac{l - x_c}{d^2 + 4(l - x_c)^2}$ | $p_{11} = \frac{1}{2} \frac{x_c}{\sqrt{4x_c^2 + d^2}}$, |
|             | $\arctg\left(\frac{d_1}{2(l - x_c)}\right) - \arcsin\frac{\mu}{\tan\alpha}$ | $\arctg\left(\frac{d_1}{2(l - x_c - 0.5d_1)}\right) - \arcsin\frac{\mu}{\tan\alpha}$ |
| $P_{12}$    | $p_{12\max} = \frac{1}{\pi} \left(\arcsin\frac{B}{\sqrt{l^2 + d_1^2}} - \arctg\frac{d_1}{l}\right)$ | $p_{12\max} = \frac{1}{\pi} \left(\arcsin\frac{B}{\sqrt{l^2 + d_1^2}} - \arctg\frac{d_1}{l}\right)$ |
|             | $p_{12\min} = \frac{1}{\pi} \left(\arcsin\frac{B}{\sqrt{l^2 + d_1^2}} - \arctg\frac{d_1}{l}\right)$ | $p_{12\min} = \frac{1}{\pi} \left(\arcsin\frac{B}{\sqrt{l^2 + d_1^2}} - \arctg\frac{d_1}{l}\right)$ |

To calculate the probability $P_2$, characterizing the absence of interference in the capture of other parts, use the well-known expression [6]

$$p_2 = 1 - \frac{0.9 + 1.4 \frac{l}{d_1}}{1 + 2 \frac{l}{d_1}}.$$  \hspace{1cm} (5)

The frequency of rotation of the disk at which the part does not have time to completely sink into the pocket and is thrown out of it because of collision with the guide elements of the rotating disk, we define the expression

$$n_{\text{max}} = \frac{30 \sqrt{\left(d_1 + 0.8 \Delta t - 0.2 \sqrt{5d_1^2 - 4\Delta t^2}\right)}}{\pi R}.$$  \hspace{1cm} (6)

Substituting expressions (3) – (6) in formula (2), we obtain an expression for determining the efficiency in a mechanical toothed hopper-feeding device in the form
\[ E = \left(1 - (p_{11}p_{12\text{max}})^3(1 - p_{11}p_{12\text{min}})^3\right)^\frac{1}{3} \times p_3 \times \left[1 - \frac{\pi R n}{30 \sqrt{g \left(d_1 + 0.8 \Delta r - 0.2 \sqrt{5d_1^2 - 4\Delta r^2}\right)}}\right]^4, \quad (7) \]

where the probabilities \( p_{11}, p_{12}, p_3 \) are determined from the table.

Then the feed rate of the mechanical toothed hopper-feeding device is determined by a set of expressions (1) and (7).

4. Discussion of results
Mathematical descriptions of the efficiency (7) and feed rate (1), (7) of the mechanical toothed hopper-feeding device for axisymmetric parts in the form of bodies of rotation with a conical or spherical end have been implemented in the Mathcad software environment.

In Figure 3 the results of computer simulation in the form of graphs of the efficiency dependences are shown (Figure 3, a, b) and feed rate (Figure 3, c, d) hopper-feeding device from the rotational frequency of the disk in the range of its change 0 – 15 rpm and the coefficient of friction \( \mu \) in the range of 0.2 – 0.5 for the above two types of parts. Geometric parameters of parts (see Figure 1): \( l = 0.030 \text{ m}; \ d_1 = 0.01 \text{ m}; \ l/d_1 = 3; \ l_1 = 0.025 \text{ m}; \ x_c = 0.015 \text{ m}; \ d_2 = 0.0067 \text{ m}; \ r = 0.005 \text{ m}. \) Radius of the rotating disk on the axis of pockets \( R = 0.185 \text{ m}. \)

The graphs show that for parts with a conical end face, the highest value of the efficiency is from \( E_{\text{max}} = 0.135 \) for \( \mu = 0.5 \) to \( E_{\text{max}} = 0.298 \) for \( \mu = 0.2 \) (see Figure 3, a) and practically does not change at values of frequency of rotation of a disk from 0 to 6 rpm. Maximum feed rate toothed hopper-feeding device for such components reaches \( F = 55 \text{ parts/min} \) for \( \mu = 0.5 \) and \( F = 121 \text{ parts/min} \) for \( \mu = 0.2 \) (see Figure 3, c) at a rotational frequency of the disk of 9 rpm. The graphs show that with an increase in the friction coefficient by 2.5 times the efficiency and the feed rate of the toothed hopper-feeding device are reduced by 2.2 times (see Figure 3, a, c).

For parts with a spherical end face, the highest value of the efficiency is from \( E_{\text{max}} = 0.228 \) for \( \mu = 0.5 \) to \( E_{\text{max}} = 0.382 \) for \( \mu = 0.2 \) (see Figure 3, b) and practically does not change at values of frequency of rotation of a disk from 0 to 5 rpm. Maximum feed rate toothed hopper-feeding device for such components reaches \( F = 93 \text{ parts/min} \) for \( \mu = 0.5 \) and \( F = 155 \text{ parts/min} \) for \( \mu = 0.2 \) (see Figure 3, d) at a rotational frequency of the disk of 9 rpm. With an increase in the friction coefficient by 2.5 times the efficiency and the feed rate of the toothed hopper-feeding device are reduced by 1.7 times (see Figure 3, b, d).
Figure 3. Graphs of the efficiency (a, b) and the feed rate (b, d) of the mechanical toothed hopper-feeding device from the rotational frequency of the disk $n$ and the friction coefficient $\mu$ for parts with conical (a, b) and spherical (b, d) end.

Thus, the coefficient of friction has a significant impact on the value of the efficiency and the feed rate of the hopper-feeding device.

5. Conclusions

As a result of the research on the basis of the new theoretical approach proposed by the authors, mathematical models of the efficiency and feed rate of a mechanical toothed hopper-feeding device with a ring orientator are developed, which take into account the probabilistic nature of the process of capturing parts and the interrelated effect on the feed rate of the hopper-feeding device of its main kinematic and structural parameters, geometric and physical and mechanical parameters of the parts.

The correctness of the proposed mathematical models is confirmed by a qualitative comparison of the results of computer simulation of the efficiency and feed rate of the toothed hopper-feeding device with the results previously obtained by the authors for similar disk mechanical hopper-feeding devices [7 – 9].

The proposed mathematical dependences will allow at the design stage of the technological machine to justify for a specific axisymmetric part in the form of a body of rotation, asymmetric at the ends, the choice of rational parameters of a mechanical toothed hopper-feeding device with a ring orientator, ensuring its maximum feed rate.

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