The strength study of the rotating device driver indexing spatial mechanism

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Abstract. The indexing spatial mechanisms are widely used in automatic machines. The mechanisms maximum load-bearing capacity measurement is possible based on both the physical and numerical models tests results. The paper deals with the driven disk indexing spatial cam mechanism numerical model at the constant angular cam velocity. The presented mechanism kinematics and geometry parameters and finite element model are analyzed in the SolidWorks design environment. The calculation initial data and missing parameters having been found from the structure analysis were identified. The structure and kinematics analysis revealed the mechanism failures possible reasons. The numerical calculations results showing the structure performance at the contact and bending stresses are represented.

Key words: rotary mechanism, cam mechanism, orthogonal mechanism

1. Introduction

The driven sprocket indexing spatial cam mechanism at the driving disk constant angular velocity is considered. Similar constructions are used, for example, in advertising installations [1]. The study subject (figure 1) consists of the driving disk (figure 1, reference 2) and the driven sprocket (figure 1, reference 1) and represents the driving mechanism providing the stop in three positions.

The disk-sprocket mechanism is the transmission comprising the complex spatial interaction between the driving and driven links with the point contact moving along the links surface [2]. Therefore, the numerical simulation in the Solidworks environment was chosen for the study. The similar models examining in the computer environment provides the study subject comprehensive information [3].

Most present day researches focuse on the spatial mechanisms kinematics studying [4, 5, 6] without taking into consideration the acting loads, that does not confirm their performance.

The presented in papers [7, 8, 9] structures analysis reveals that the main disadvantage is the parts working surfaces overwhelming sliding friction. This type of friction wears the parts surface thus reducing the thickness. Parts shape changing increases backlashes and gaps in the structure caused by wear that has a negative impact on the operation smoothness and develops the additional dynamic moments [4].

The research objective is to study the driving mechanism under the increased acting loads and to identify measures for reducing harmful factors emerging in operation.
2. Problem statement

The aims of the study include the following:
- the analysis of the structures on the wear criteria, of the performance at high loads, of the mechanism seizure and jamming inability;
- the strength calculation according to the interacting surfaces contact and bending stresses.

The driving mechanism structure analysis was carried out. The driving mechanism structure analysis objective is to identify the factors influencing the parts operation.

Circumferential speed is defined by the following equation:

\[ V = \frac{\pi \cdot d_{c1} \cdot n_1}{60000} = \frac{\pi \cdot 68 \cdot 12}{6000} = 0.042 \text{ m/s} \quad (1) \]

where \( d_{c1} \) is the sprocket contact point diameter. On the assumption of the speed the transmission is low-speed one, lubrication conditions are without lubrication or grease lubricant can be applied.

The backlashes occurring can be caused by the material compression at the environmental temperature changing. Furthermore, teeth jamming may occur at the temperature increase.

### Table 1. The examined model material.

| Properties                              | Arnamid | PA SV 30-2T |
|-----------------------------------------|---------|-------------|
| Density \( \rho \), kg/m\(^3\)          | 1380    |              |
| Melting temperature, °C                 | 220     |              |
| Tensile strength, MPa                   | 165     |              |
| Bending stress at maximum load, MPa     | 233     |              |
| Flexural modulus, MPa                   | 8000    |              |
| Heat deflection temperature is 1.8 MPa, °C | 200     |              |
| Water absorption in water (23 °C, 24 h), % | 0.95    |              |
| Transversal shrinkage in casting, %     | 0.7…0.9 |              |
| Longitudinal shrinkage in casting, %    | 0.1…0.2 |              |
| Shrinkage temperature, °C               | 260…270 |              |
3. Theory
For calculating the loads, the computer model making possible to obtain the systems behaviour approximate evaluation by performing computational experiments is used. Moreover, it is also necessary to define the missing initial data used in calculations and required for the model future research.

The driving disk shaft torque is set \( M = 7.53 \text{ N} \cdot \text{m} \).

Taking into account the operating conditions possible changes the safety coefficient \( C_s = 1.3 \) was introduced. Then the transmitted moment by the mechanism is \( M_1 \approx 10 \text{ N} \cdot \text{m} \).

The inertia moment is identified for the part mounted on the mechanism output shaft \( J = 0.03248 \text{ kg} \cdot \text{m}^2 \).

The driving disk and sprocket rotation frequency is \( \omega : 72 \text{ °/s} = 1.256 \text{ rad/s} \).

Since the angular velocity real change over time is unknown, then the assumption that the angular acceleration is \( \varepsilon = \frac{d\omega}{dt} \), where \( \omega \) varies from 0 to \( \omega \) will be used.

In this case, taking into account the load application time 0.001 seconds (the non-linear analysis step in SolidWorks) the following equation is derived for the peak load:

\[
\varepsilon = \frac{d\omega}{dt} = \frac{1.256 - 0}{2 \cdot 0.001} = 628 \text{ rad/s}^2
\]

Then the transmitted dynamic torque is:

\[
M_{\text{dyn}} = -J \cdot \varepsilon = 0.03248 \cdot 628 = 20.4 \text{ N} \cdot \text{m}.
\]

While working for 5 seconds and having a standstill for 15 seconds, each tooth takes the stress once per minute. The service life in the stresses variation cycles for a period of one year is designated as \( N_{k1} \) and equals to: \( N_{k1} = 525600 \); of 3 years is: \( N_{k3} = 1576800 \).

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4. Experimental results
The model studying is to define the stresses (figures 2, 3) and resource (figure 4) for the driving disk and driven sprocket.

In the case of pure shear, at von Mises stresses, the destruction occurs if \( \tau_{\text{max}} = 0.577 \sigma_t = 0.577 \cdot 139 = 80.2 \text{ MPa} \).

The force limiting value is identified on the basis of the dynamic torque obtained value. The dynamic torque value is assumed to be transferred by the mechanism in operation, since the pair parts manufacturing poor accuracy results in the fact that the new mechanism initial dead range can already be compared with the permissible one [2]. The maximum force not causing the model shearing is: \( F = 600 \text{ N} \). The force obtained value corresponds to the transmitted torque of 20.4 N \cdot m.

Figure 2. Sprocket stress diagrams on the basis of the maximum distortion energy theory (von Mises).
5. Results discussion
The sprocket stresses value (figure 2) approximately equals to 75 MPa which is based on the assumption of the shearing absence. Hence, the limit contact force obtained as a result of modelling in accordance with the sprocket working surfaces to be not more than 600 N.

The constant amplitude material fatigue event is completely determined by the alternating stress and load cycles number. The sprocket load cycles number (figure 4) is 256117 cycles which corresponds to 6 months of the part operation.

The sprocket breakage point under the contact stresses influence (figure 2) coincides with the fatigue failure area (figure 4).

The engaged driving disk stresses value (figure 3) is somewhat less (≈63 MPa) than the sprocket one. However, the values (figure 4) obtained for the resource indicate disk surface failure approximately in half a month. These results suggest the surface failure and premature fatigue wear as physical modelling results do not imply the mechanism breakdown as a result of the disk failure [2].

6. Conclusions
As a consequence of the conducted research, the results of contact stresses were obtained, the parts strength and durability were checked.

The obtained modelling results show the model application for the given initial data to be unacceptable. Thus, it is manifested in the parts short service life for the most severe operating conditions.
conditions. The working surfaces overwhelming sliding friction has the greatest influence. However, the model structure perfection is impossible to be improved, as the sprocket cranked beam forms cannot be avoided.

Although the simulation results for the specified life do not satisfy the required, they have shown good agreement with the experiments results obtained on the basis of the physical model [2]. The parts minimum model contacting surfaces wear life is half a month, but it should be specified by experimental methods. It should also be noted that the model service life based on the parts damage assumption for the most severe operating conditions, is rather short.

The rotary mechanism under consideration functionality enhancing is possible to be achieved by using the other forms of the indexing transmission and avoiding the sliding friction in favour of the rolling one.

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