WEAR AND MALFUNCTIONS OF GEARBOXES IN THE MINE LOCOMOTIVES FOR UNDERGROUND TRANSPORTATION

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ABSTRACT. The paper examines the impact loads on the wheels of the underground mining locomotives derived from the dynamic loads and the impact they exert on the gearbox. The dependencies for the emergence and change of the forces are shown. Ways of wear reduction are suggested.

Keywords: Underground mine locomotives, wear, impact loads.

ИЗНОСВАНИЯ И ПОВРЕДИ В РЕДУКТОРИТЕ НА РУДНИЧНИТЕ ЛОКОМОТИВИ ЗА ПОДЗЕМЕН ИЗВОЗ

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РЕЗЮМЕ. В статията се разглеждат ударните натоварвания върху колооста на рудничните локомотиви за подземен извоз, получени от динамичните натоварвания, и въздействието, което оказват върху опорните точки на редуктора. Изведени са зависимости за появата и изменението на силите. Посочени са пътищата за ограничаване на износването.

Ключови думи: локомотиви за подземен извоз, износване, ударно натоварване

Introduction

The trend in the development of underground mining locomotives is increasing the speed of movement and traction power (Mateev, 1961). Restrictions on the dimensions of mining locomotives remain virtually unchanged, the development and modernization of these locomotives remain primarily related to the possibilities of creating powerful and compact drives. The development of the non-adjustable power drives for individual propulsion in the arm-suspension of the traction motor naturally takes place in accordance with the general trends in the mining locomotive (Volotkovskii S., 1981). Contemporary electric locomotives for underground mine transport have an increased rotation speed of the traction motor rotor. This allows more powerful motors to be mounted in the limited space. In most cases, the motor is positioned longitudinally (Figure 1). Maintaining the allowed speed of movement of the locomotive (12 km/h) in this way requires a corresponding increase in the gear ratio. As a result most of the modern mining electric locomotives are equipped with two-stage cylindrical gears with transverse traction (4.5 АРП2М, 5АРП и 7АРП) and conical-cylindrical wheels - with the longitudinal traction motor (see Figure 1) with (К10М1У,К7М1 РКЛ-7А, РКЛ-10А, РАЛ-8А, АМ8Д, etc.) These gearboxes are enclosed into a solid massive shell, made from cast steel or welded structure. The gearboxes houses both the engagement and the centering of the electric motor, as well as the bearing and transmission of the torque of the driven wheel.

Exposition

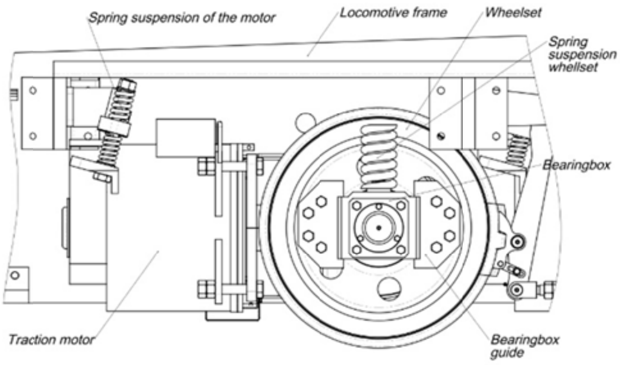


Fig. 1. Longitudinal power drive РКЛ-10А

In its movement the locomotive experiences longitudinal, transverse and vertical loads caused by road irregularities, curves and conicity of the wheel bracelets. These loads are also transmitted to the gearbox through the drive wheels. All electrically-operated undercarriages are equipped with gearboxes mounted on the wheelset and spring-mounted via the frame motor (Figure 1). There are permanent and impact loads on the gear unit. The last ones are obtained from the accelerations (positive and negative) of the locomotive and the passage of the wheelset through the rail joints, the different types of arrows and other unevenness on the road.

With this kinematic coupling scheme, which is dominant for almost all railroad electric transport, the most powerful load is absorbed by the wheelset and all its components directly connected to it - the wheels, the shafts, the gear housing and the bearingbox. The reason for this is that the kinematic element is unsuspended and absorbs all traction and braking loads and assumes the dynamic loads from the unevenness of the road and the available gaps in the kinematic scheme. Such gaps are the technological loopholes that are in the sliding elements - the guide and the bearingbox. The links between the rails are particularly influential. The difference in heights and the gap between the two rails causes a shock load.

Fig. 2. Wear bearing housing

My studies have shown that the wear and deformation of the gearbox bearings as well as their housing (Figure 2) takes the most damages. There are some differences when we have a rolling or a plane bearing. Studies show that, in the primarily used rolling bearings, the process begins with gap formation in the gearbox body. In my opinion, this gap is due to plastic deformations caused by the different types of loads and, most of all, the impact ones. The appearance of the gap creates conditions for obtaining internal impact loads in the gear housing as well. At the beginning, the material splashes and further increases the gap. In the end, it gets such dimensions that the loads reach values exceeding the strength of the shell of the bearing. This leads to its destruction (Figure 3). This is an emergency state that blocks the wheelset, and the locomotive respectively.

In the plain bearing, there is a technological gap in the bearing itself, which is designed to provide lubrication of the bearing. This gap, mainly due to impact loads, is progressively increasing. This leads to deterioration of the lubrication of the bearing. It also leads to a change in the tooth spacing and worsening of the grip in the tooth.

Fig. 3. Destruction wheelset roll bearing K10M

The process continues until the kinematic connection between the gears is broken. In this process, the bearing bush (Figure 4) and also part of the gear housing are worn to the end. Although such a level of weariness seems absurd, I have found a significant number of such cases in our practice.

Fig. 4. Wear of plain bearing bush AM8Д

The change in the center distance of the tooth pair leads to abnormal contact of the teeth and, respectively, to their intensive wear (Figure 5).

Fig. 5. Gear wear AM8Д

Forces that act on the bearings, respectively the gear housing, will mainly be divided into static, dynamic and percussion. The static loads are determined by the weight of the gearbox and the engine coupled to it, as well as by the constant force produced by the wheelset - traction or braking, taking into account the center of gravity, the position of the motor suspension and the position of the bearings. Bearings, by construction, are arranged to be evenly loaded by this force, which means that each bearing takes up half of the load. In the drives I have studied, the center of gravity is located very close to half the distance between the engine suspension and the gear axis. Under these conditions, each bearing will be loaded with the following force:

$$P_{cm} = \frac{\sum m_i}{2}, N; \qquad (1)$$

where: $\sum m_i$ is the sum of the masses of the involved elements.

The maximum traction effort is determined by the engine torque, the gear ratio and the traction coefficient (Mateev, 1961):

$$F_T = \psi \frac{2iM_{\text{дв}}}{D_k}, N; \qquad (2)$$

where: $M_{\text{дв}}$ - the rated torque of the engine;

i - the gear ratio of the power transmission;

D_k - the diameter of the traction wheel;

ψ - the traction coefficient.

Dynamic loads are caused by the change in speed in the longitudinal and transverse directions. Longitudinal accelerations are caused by the change in train speed. The accelerations obtained in the gearbox depend on the traction effort generated by the drive wheels and the total mass of the single drive, their size is being determined by the formula:

$$F = \frac{dv}{dt}\left(m_p + m_{\text{дв}}\right), N; \qquad (3)$$

where: m_p - mass of the gearbox;

$m_{\text{дв}}$ - mass of the traction motor.

The equation above is true in the absence of a technological and non-technological gap in the kinematic scheme of the gearbox and the machine (between the bearingbox and its guide - Fig. 6). In the presence of gaps (technological and non-technological) we have initial acceleration of the gear unit together with the motor until it is removed. When the gap is seized, the wheelset stops its movement, causing a stroke between the bearingbox and its guide. At the same time, the gearbox continues its movement until the technological and non-technological gaps in the bearing assembly are closed. Under these conditions, impact loads are obtained in the bearing housings of the hull and the bearings themselves. The magnitude of this impact is determined by the kinetic energy of the gearbox. Obviously, the wider the gap and the longer the drive's travel are, the higher the speed will be. It can be determined by the following equation:

$$E_{\text{уд}} = \frac{1}{2} v^2 \left(m_p + m_{\text{дв}}\right), J; \qquad (4)$$

The ultimate speed of the single drive is determined by the force applied to it and the gap between the jack and the driver. Assuming the motion is equally accelerated, it can be written:

$$a_p = \frac{F_T}{m_p + m_{\text{дв}}}, m/s^2; \qquad (5)$$

and the ultimate speed will be:

$$v = a_p t$$
$$F_{\max} = P_{\text{су}} \psi \qquad (6)$$

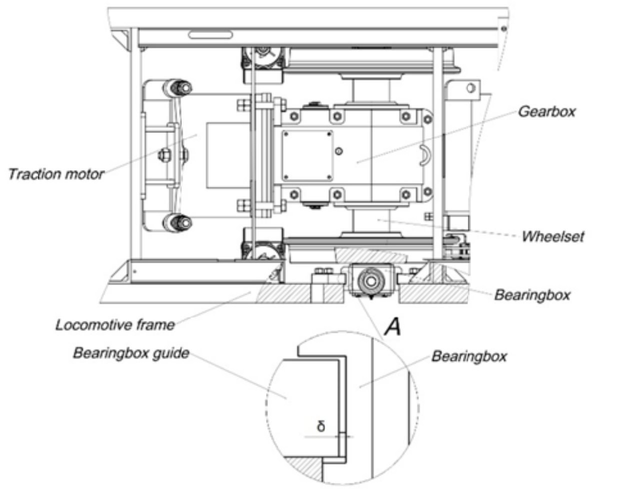


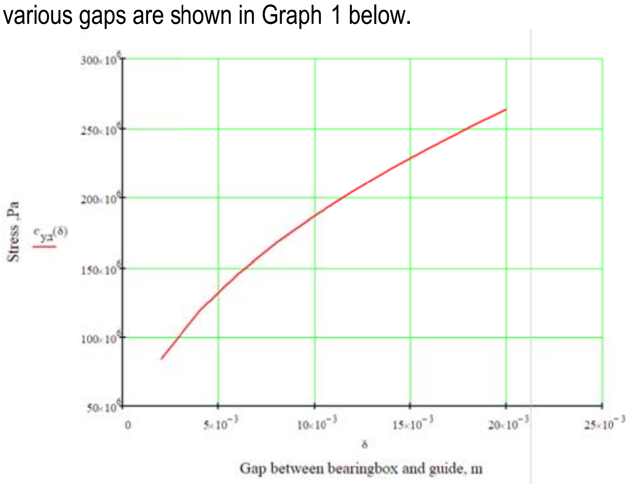
Fig. 6. Technological gaps of the power drive

Then the force of the impact, can be calculated based on (Kisyov, 1979):

$$k_{\partial} = 1 + \sqrt{1 + \frac{2\delta}{\Delta_{cm}} \frac{P_{\text{ез}}}{P_{\text{ез}} + G_{red}}} \qquad (7)$$

where: $\delta = \frac{v^2}{2a_{\max}}$ is the gap between the bearingbox and its guide.

The stresses resulting from the above dependency for various gaps are shown in Graph 1 below.



Graph 1. Relationship between the gap and stress

It can be seen from the graph that at normal values of the gap of 5-10mm, the stresses have acceptable values of 180MPa. Similar stresses are also produced at the gear unit bearing. Hertz's calculated tensions are one and a half to two times greater. These high stresses lead to compacting, and then to plastic deformations as shown in the above-mentioned figures. Additionally, the situation is complicated by obtaining a non-technological gap between the bearing and the bearing body. This clearance creates the conditions for an additional internal impact between the wheelset and all the elements attached to it and the gear housing. This impact stress initially leads to rapid increase of the gap and subsequently when the size is enough to destruction of the bearing.

Analogical impact loads are obtained by passing the wheelset through unevenness on the track, mainly the joints between the rails. These will be a topic of another paper.

Conclusion

This study aims to determine the magnitude of the impact forces that eventually lead to wear of the locomotive, and in some instances, to its failure. In my opinion, the limitation of the size of these forces is most likely to be achieved by extinguishing part of the energy of the impact.

In conclusion I can say that the grounded reason for getting gaps in the bearing assembly are the impact loads in a horizontal and vertical direction. The reduction of these loads in the horizontal direction can be constructively limited by the introduction of a damping element in the guiding drivers of the locomotive. The damping could be made of rubber elements to provide the necessary elasticity and mobility of the element, while attenuating the impact loads. To some extent, they reduce the impact loads in the vertical direction as they add the entire mass of the locomotive to the dynamics of the process. Unfortunately, these constructions have a short operating life, which limits the application. Attempts to absorb such elements are made by various companies. At this stage, these elements have a limited resource, finding them applied to smaller machines. In the R & D base of the University of Mining and Geology "St. Ivan Rilski" under my leadership a similar damping mechanism is being developed. It is designed to take on loads of seven-tonne locomotives that are common in Bulgarian mines. It is of great importance to increase the resources of the existing locomotives to maintain the margin in the boundaries defined by the constructor, and to increase their repairs over the limit.

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The article is reviewed by Assoc. Prof. Dr. Kristian Tzvetkov and Assoc. Prof. Dr. Ivan Minin.