DESIGN OF CONSTRUCTIONAL OPTIMISATION DETERMINED FOR MIXER TRUCK GEARBOX

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Abstract: When designing and constructing machines, it is very important to ensure that machines that are subjected to different types of stresses and different operating conditions, bear the loads to which they are subjected and also withstand operating times. The transmission by gear wheels are the most commonly used transmission mechanisms in practice. The paper deals with the analysis of the mixer truck gearbox damage that occurred during the operating time. The result of the optimization is a change in the geometrical parameters of the gear wheels. The constructional optimisation of this gearbox is designed to eliminate the failure rate.

Keywords: gearbox, spur gear, optimization, strength calculation of gearing, safety factor.

1 Introduction

Gearing are the most commonly used transmissions mechanisms in practice. Gear wheels appear in history in the form of jewelry in the past, for 2000 years BC. Already 300 years BC, primitive forms of gearing were used, especially in water mill drives for pumping water or lifting heavy loads. Great development occurred in the development of windmills, which were among the largest machines in the period (9th to 18th century). Later, the first theoretical works on gearing by Leonardo da Vinci (1452 - 1519) appear in which the wear of the sides of the teeth has already been mentioned. The need to devote more attention to the gear geometry was related to the increasing demand for gears to operate at ever higher speeds and higher loads [1].

The gearbox is a technical implementation of the gear mechanism in the form of a separate machine subsystem. It represents the most widespread and most important type of gear mechanism, we encounter in all areas of technology. Stepped gearboxes are mainly used to drive vehicles and wherever machine workpieces need to be driven at different speeds at different loads [2, 3]. The transmission mechanism transmits and optionally distributes the energy supplied from the drive machine to the working machine.

Due to the characteristics of most types of engines (in particular combustion piston) transport and mobile machinery, the stepped gearbox must fulfill the following functions:

1. Change the value of the gear ratio to match the speed and force modes of input and output, that is, to vary the driving power on the wheels and the speed of the vehicle in a relatively wide range with relatively low torque and engine speed.
2. Change the sense of the gear ratio as needed to reverse the movement.
3. Enable gearbox neutral, in which the input and output shafts are open and the engine can operate with a stationary vehicle.

The gear mechanism consists of master gear or planet gear. The gear mechanism consists of several main parts, which include shafts on which other important parts of the gearbox are mounted, namely gears or sprocket, carriers that are connected to each other and the frame by kinematic pairs (gearing and bearings) and temporarily connected by control elements [4].

When designing and constructing machines, it is very important to ensure that machines that are subjected to different types of stresses and different operating conditions, bear the loads to which they are subjected and also withstand operating times [5 - 7]. In today's modern day, thanks to computational technology to incorporate CAD programs, we can achieve, in a very short time, the development of structures, their analysis, and calculations.

The paper describes the analysis of the existing damaged gearbox on the basis of which structural modifications were designed to maintain the required characteristic parameters. The purpose of this design modification is to eliminate the failure of a given gearbox during operation.

2 Characteristics of gearbox

The gearbox is mounted on a mechanical mixer of truck (Fig. 2) determined for prepare of the concrete before use. It is a single-stage gearbox with helical gearing. The original gearbox operates in 8-hour dual-shift operation 5 days of week. The failure occurs at approximately three-month intervals. The gearbox did not meet the current load condition. Drawing documentation is not known, so values for this particular gearbox are given on the label:

- power P_1= 7.5 kW,
- input speed n_1= 1450 min^{-1},
- output torque M_{k2}= 156 Nm,
- output speed n_2= 480 min^{-1},
- gear ratio i= 3.15,
- weight m= 113 kg,
- quantity of oil Q=0.5l.

3 Damage of helical gearing

Gear wheel classification due to cause of damage is of very importance because it allows to determine the operating conditions from which the damage occurred. Damage of gear wheels are very different [8-11].

For dimensioning of the gear wheels, those failures that are fatigue and seizing at higher speeds or at high slip speeds are important. Damage to gear wheels due to damage is divided into two groups:

- tooth surface damage,
- damage to gear wheels by fracture of teeth [12].

There, the surface parts of the teeth between the pinion and the gear wheel contact each other when the gears are rotated. The actual contact area is smaller because the contact of the two tooth surfaces with the specified roughness occurs only between the highest projections of the uneven surfaces. These surfaces produce large pressures and the material surface is deformed from the load. Therefore, the contact area increases, the temperature increases and the surface tension and the surface of the tooth flanks are large. This can lead to different types of tooth damage [13].

In the case of strength calculation, the bending and contact resistance is normally considered. In the case of the gear wheels with hard toothed edges, the fatigue fracture is the limiting state, especially in case of cemented and surface hardened teeth. The fatigue wear of the surface layer (pitting) tends to be the ultimate condition for gear wheels with heat-treated condition and soft teeth. Due to the severity of the break-out accident, higher reliability is often required, which is due to the level of fatigue fracture safety. Therefore, the load-bearing criteria cannot be clearly defined. In addition to fatigue damage, damages caused by unsuitable ratios can also arise, especially during lubrication,

impact stress and also caused by material errors, structural and technological errors [14-18].
The gearing of the gearbox is shown in Figure 1.

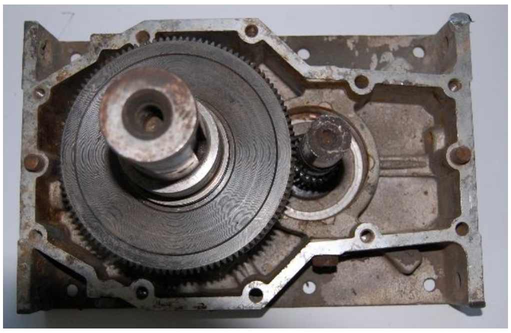

Figure 1. The gearing of the gearbox.

The helical pinion is designed as part of the input shaft. All pinion teeth are damaged (Figure 2). The teeth are only damaged in the part of the contact with gear wheel. A fatigue fracture in the foot of the tooth, which resulted in complete abrasion of the teeth, occurred on the pinion. The tooth profiles were completely abraded.

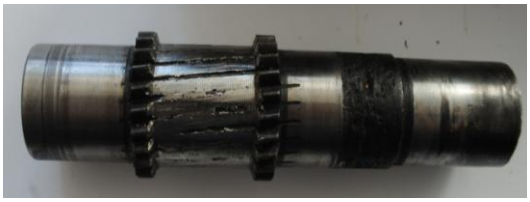

Figure 2. The pinion damage.

The helical gear is pressed on the shaft (Figure 3).

Figure 3. Gear wheel pressed on the output shaft.

The gear wheel is damaged by breaking of the tooth element. On each tooth there are signs of damage to the sides of the tooth by seizing, which can occur during overloading for several hours or even minutes. The physical and chemical properties of the lubricating oil play an important role in its formation. On the gear wheel, a fatigue fracture of the tooth part occurred and the abrasion that can be seen in Figure 4.

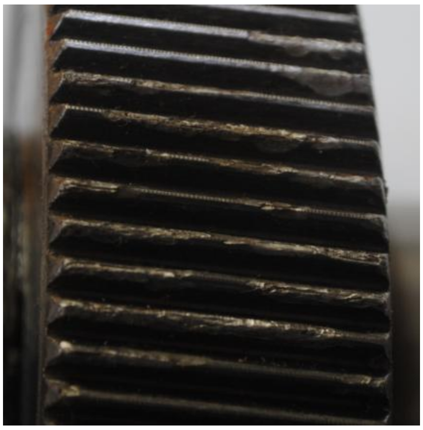

Figure 4. Damage to the teeth of gear wheel.

The pinion is more damaged than meshing gear wheel. This damage could have occurred for the following reasons:

- one-time overload,
- there was no regular check (in the time of increased noise),
- there was no proper lubrication or proper lubricant was not used,
- fragments of damage teeth that remained in the gearbox after previous damage could cause complete damage.

The cause of the fracture is ultimately due to the influence of external or internal stresses, which in this case only exceeded the breaking strength or fatigue limit of used material. If the material is brittle, it breaks and creates a crack or fracture. The effects that lead to failure are diverse, may be inappropriate construction, unsuitable material, improper transport, or due to time change in material properties (fatigue).

4 Design of gearbox constructional optimisation

Due to the extent of damage and the unsatisfactory condition of the gearbox, it has been proposed to solve this problem by designing a new gearing that will meet the operating characteristics of the particular gearbox.

The design was based on data on the gearbox label where the ratio number is i = 3.15. To maintain the required ratio number, the number of teeth of the new pinion z_1 = 35 and the number of teeth of the new gear wheel z*2* = 110 were selected. For the new gearbox we have to keep the gear ratio i = 3.15 and the axial distance a = 100 mm.

In practice, the pinion of the hardened material and the meshing gear wheel of unhardened material are most often chosen. The pinion gearing must be of hardened material because the pinion must bear more loads and thus the gearing is then more stressed to the touch and bend.

Steel 15 241 was chosen for the pinion gear material. It is a noble premium steel suitable for surface hardening, for which the tensile strength is R_m = 980MPa, the slip stress is R_e = 850MPa, the limit, the fatigue limit of the contact in contact σ_{HLimb} = 1160MPa and the bending fatigue limit σ_{FLimb} = 528MPa. For pinion teeth, the hardness in the core of the tooth is JHV = 315 and the hardness of the side of the tooth is VHV = 600 ~ 675, according to [13].

For the gear wheel material, a steel of 12 050 was chosen for which the tensile strength is R_m = 640MPa, the slip stress is R_e = 390MPa, the limit, the fatigue limit of the touch at σ_{HLimb} = 520MPa and the bending fatigue limit σ_{FLimb} = 410MPa.

The first step was to design the gear module according to [13]. According to this standard, the modul of bending is determined by:

$$m_n = f_F \cdot \sqrt[3]{\frac{K_F M_{k1}}{\psi_m z_1 \sigma_{FP}}} \tag{1}$$

where f_F is the bending coefficient for bevelled teeth, M_{k1} is the input shaft torque, ψ_m is the tooth width coefficient, z_1 is the number of pinion teeth, σ_{FP} is the permissible bending stress for the disappearing load.

The modul value of the contact stress was determined by the relationship [11]:

$$m_n = f_H \cdot \sqrt[3]{\frac{K_H M k_{k1}(i+1)}{\psi_m \cdot z_1^2 \cdot i \cdot \sigma_{HP}^2}} \tag{2}$$

where f_H is the coefficient for bevelled teeth subjected to contact, M_{k1} is the input shaft torque, ψ_m is the tooth width coefficient, z_1 is the number of pinion teeth, i is the ratio number, σ_{HP} is the permissible contact voltage for the disappearing load.

In the design of the modification, the standard [13] was used, which in the first step calculated the normalized modul value for the pinion in bending (m_n = 1.15mm) and in contact (m_n = 1.06mm). Based on this calculation, a normalized module value of m_n = 1.25mm was selected. To maintain the original axial distance of a = 100 mm, helix angle of β = 25 ° was proposed.

The second step was the complete strength check of the pinion and the gear wheel according by STN 014686 [13]. According to this standard calculation is based on the control of bending strength and contact stress. For calculations, the load input values were used as indicated on the original gearbox label.

In bending strength calculation, the fatigue fracture of the teeth, starting from the root transition area on the active side of the teeth, is monitored as a limit state. According to this standard, the bending capacity can be proved by calculation the safety factor for bending failure in the root, for which [13] applies:

$$S_F = \frac{\sigma_{Flimb} \cdot Y_N \cdot Y_\delta \cdot Y_X}{\sigma_F} \geq S_{Fmin} \tag{3}$$

where S_F - safety factor for bending failure in the root, σ_{Flimb} - bending fatique life for the intended way of load (MPa), Y_N - coefficient of durability, Y_δ - coefficient of nick sensitivity, Y_X - coefficient of dimension, σ_F - bending stress in the critical cross section of root (MPa), S_{Fmin} - the minimum value of the factor: $S_{Fmin} = 1.4$.

In contact stress calculation, the progressive surfaces fatigue damage (pitting) of the teeth is monitored as a limit state. According to this standard, the contact capacity can be proved by calculation the, for which applies:

$$S_H = \frac{\sigma_{Hlim} \cdot Z_N}{\sigma_H} \cdot (Z_L \cdot Z_R \cdot Z_V) \geq S_{Hmin}, \tag{4}$$

where S_H - safety factor against fatigue damage of tooth side, σHlim - fatigue limit in contact (MPa), Z_N - coefficient of durability, Z_L - coefficient of lubricants, Z_R - roughness coefficient of tooth side before meshing, Z_V – coefficient of peripheral speed, σ_H - Hertz stress in pitch point (MPa), S_{Hmin} - the minimum value of the factor: $S_{Hmin} = 1.1$.

Fatigue limit values for gearing materials in accordance with this standard are given for grinding teeth sides. The influence of roughness of untreated and hardened teeth on their load-bearing capacity is taken into account in the strength calculation by the roughness coefficient of tooth side before meshing. The load-bearing capacity of the teeth in the bend reduces manufacturing deficiencies, such as the decarburized surface and cracks in the area of the tooth's root that may occur during quenching. These effects should be avoided in the manufacture of gears, since the calculation according to standard does not take into account such deficiencies.

This standard specifies the values of the durability of ma-terials covered for 5.10^7 load cycles. For other load cycle values, the standard sets the reduction factor.

The results of the gearing strength calculation and the basic geometric parameters of gearing are shown in Table 1.

Table 1. Some parameters of strength calculations.

Parameter	*Pinion*	*Gear*
Number of teeth	35	110
Pressure angle (˚)	20	
Helix angle (˚)	25	
Normal module (mm)	1.25	
Rotation speed (min^{-1})	1450	460
Torque (Nm)	49.44	150
Pitch diameter (mm)	48.273	151.715
Centre distance (mm)	100	
Material of the gear	15 241	12 050
Safety factor S_H	1.48	1.33
Safety factor S_F	3.82	2.48

The calculated safety factors of bending and contact for the pinion and gear wheel are for the required service life of 10,000 hours is satisfactory.

The pinion was designed as part of the input shaft (Figure 5).

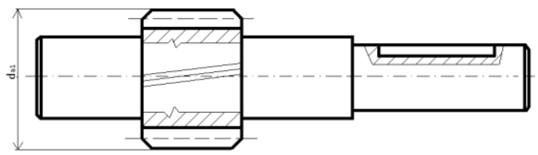

Figure 5. Designed input shaft with pinion.

The gear wheel is designed as a separate wheel and is pressed onto the output shaft (Figure 6). The shaft and gear wheel have been selected with system of limits and fits an H7 / r6 to transmit the transmitted torque.

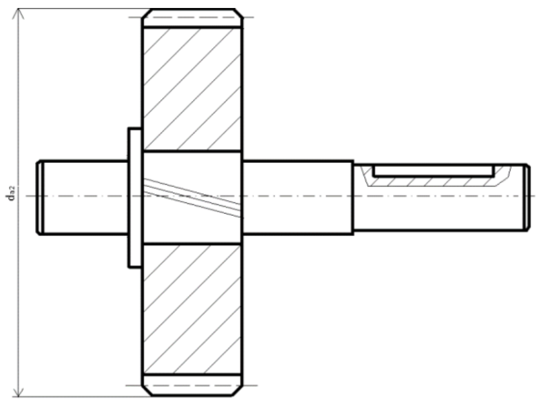

Figure 6. Designed input shaft with pinion.

In Figure 7 is a CAD model of a transmission gearbox with structural modifications to eliminate operational failure.

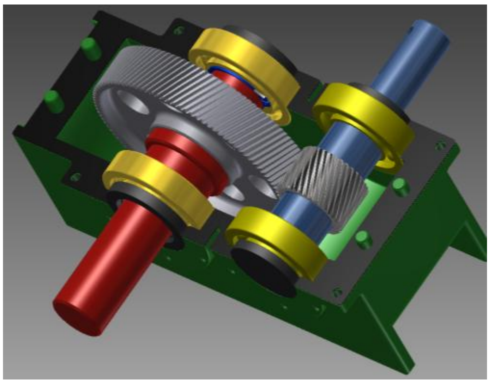

Figure 7. 3D gearbox model with designed changes.

4 Conclusion

When investigating the damage to a particular gearbox, it was found that the appropriate heat treatment was not applied to the original gearing, because on the pinion all the damage was devastating - total tooth breakage. The teeth of the pinion do not meet the condition of hard surface and tough core. A new material and a new heat treatment for the pinion and gear wheel were selected. For pinion it was 15 241 steel and for gear wheel 12 050 it was steel, hardened and cemented. Since the bearings were not damaged, the design modifications involved the replacement of the entire gear set with a suitably selected material and heat treatment. At the same time, new numbers of teeth were selected but the ratio number was kept. The standardized module value was calculated and designed according to the standard. The helix angle of the teeth was chosen to maintain the original axial distance.

Subsequently, the geometric parameters of the new gear were calculated and the safety factors calculated according to STN 01 4686. Because the pinion is designed as part of the shaft, only the dimensions of the gearing are changed, all other shaft parameters are retained. The gear wheel is designed as a separate and on the output shaft is fixed by pressing. The interference fit was designed and strength checked. All designed parameters are suitable for a specific gearbox so that it can operate in a fault-free state for the required service life.

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