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BWE TRACTION UNITS FAILURES CAUSED BY STRUCTURAL ELASTICITY AND GEAR RESONANCES

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Original scientific paper
Extreme powerful gear drives have a relatively elastic structure of housing components. The supports and the connections with the carrying structure are not rigid enough either, including this structure itself. The paper analyses the gear unit supporting bearings failure of bucket wheel excavators. The failure of BWE SchRs-1600 occured after a very short service life, in the form of progresive and catastrophic pitting. The presented analysis shows that this phenomenon appeared due to multiple synchronized influences which are a direct consequence of the structure elasticity, the reducer and output shaft masses eccentricity, the moment lever rigidity, and the bearing area local deformations. An improved design solution as well as the mentioned analyses are presented. Additionally, gear transmission units can operate properly in the resonance conditions but they cause very unfavorable effects on other components, such as the bearings. The article contains gear resonance identification and its influence on rapid failure of bearings and other gears of the SRs-1301bucket wheel excavator. Thereafter follows the correction of design parameters of the critical gear pair in order to avoid its resonance and excitation of natural frequencies of gearbox housing.

Keywords: *bearings, elasticity, gear drive, pitting, resonance*

Kvarovi vučne jedinice rotornog bagera prouzročene elastičnošću konstrukcije i rezonancijom zupčanika

Izvorni znanstveni članak
Zupčasti prijenosnici velikih snaga imaju relativno elastična kućišta. Također oslonci i veze s nosivom konstrukcijom nisu dovoljno krute, uključujući i samu strukturu prijenosnika. Članak sadrži analizu razaranja ležaja reduktora rotornih bagera. Kod rotornog bagera SchRs-1600, kvar se dogodio nakon vrlo kratkog rada u formi progresivnog i vrlo razarajuće rupičavosti (pitinga). Prikazana analiza ukazuje na to da se ovaj fenomen pojavio zbog višestrukih sinhroniziranih utjecaja koji su direktna posljedica elastičnosti strukture, ekscentričnosti reduktora i masa izlaznog vratila, krutosti momentne poluge i lokalnih deformacija u okolini ležaja. Poboljšano konstrukcijsko rješenje skupa sa spomenutim analizama prikazani su u radu. Osim toga, zupčanici u zupčastim prijenosnicima mogu ispravno raditi u uvjetima rezonancije, ali oni izazivaju vrlo nepovoljne uvjete za rad drugih komponenata kao što su ležaji. Članak sadrži identifikaciju rezonancije zupčanika kao i analizu njihovog utjecaja na oštećenja ležaja i drugih zupčanika, koja nastupaju vrlo brzo, kod rotornog bagera SRs-1301. Iza toga slijedi prikaz usklađivanja parametara konstrukcijskog rješenja kritičnog zupčastog para kako bi se izbjegla njegova rezonancija i pobuda vlastitih frekvencija kućišta.

Ključne riječi: *elastičnost, ležaji, reduktor, rupičavost (piting), rezonancija*

1 Introduction

In the course of Bucket Wheel Excavator (BWE) exploitation, accidental damages frequently occur on the places (local areas - joints) where elasticity and strength are not adequately synchronized. These are frequent causes of accidental damages, redesigns and reparations investigated in a number of papers. Some of them are quoted in a reference list. In work [1] is given an overview of destruction causes such as design faults, manufacturing, operating and environmental faults (during exploitation). Usually, the causes of failure are all the above mentioned influences. In general, failure presents a great financial loss and serious risk for workers` safety. From a diagram representing the relation between service life and digging capacity of BWE structures it is evident that with digging capacity increase, service life decreases. The main topic of this article is failure study of bucket wheel boom tie-rod welded joint. The causes of fracture were internal flows which are not allowed for high class weld. In the next work [2], the failure of bucket wheel slewing platform is studied. Pronounced stress concentration was identified as the main cause of failure. Design solution for the reconstruction chosen based on comparative analysis of stress-strain state of alternative solutions has been proposed. Numerous studies have been conducted, based on numerical and analytical analysis of bearings. Authors of paper [3] made a numerical and analytical model of bearing for the presentation of spatial and time distribution of stress and strain on roller bodies and outer bearing ring with simulation of local sliding effect on roller bodies. In paper [4] the authors performed numerical analysis of actual bearing capacity under rolling contact of large roller bearings with hardened race to determine location and time of appearance of rolling race damage (micro cracks). Influence of load character at bearing limit capacity is elaborated in paper [5]. In [6] is presented the study of the influence of cooling fan and driving wheel unbalances on loads and life of bearings due to mass eccentricity that has caused load fluctuations and life reduction. Research on micro-pitting [7] may be divided into two approaches, experimental and theoretical. Experimental approach is used for the development of test rigs and test procedures to better understand micro-pitting behaviour in controlled conditions and to identify the parameters that control the problem such as load, hardness, sliding speed, temperature and lubricants. Theoretical approach is used to study the mechanism of micro-pitting predominately by evaluating pressures and tractions in rough contacts. In work [8] is given the presentation of these failures in the form of pitting and fractures of bearings parts which caused failures of railway axle. Paper [9] searches for fracture causes of railway locomotive traction shaft and identifies that the bearings are one of the main causes.

Researches in the field of gear vibration have been mainly oriented to sub-critical range of teeth mesh frequency, but not frequent in supercritical one [10]. The aim of these researches is to determine the influence of transmission errors on dynamic characteristics [11]. Researches of gears behaviour in the range of resonance are not frequent. The operation of gears in resonance frequency range is possible, but undesirable. Identification of different types of resonance [12, 13, 14] has major importance for recognition of resonance effects and harmonised design parameters in order to avoid resonance [15].

2 Bearing failure description

2.1 Gear drive unit of BWE SRs-1600

After only 1350 service hours, the failure of roller bearing has appeared in the reducer of bucket wheel excavator, although the calculated nominal service life of this bearing is 60 000 hours. Fig. 1 shows 3D model of the reducer. The reducer case consists of two wholes, planetary PL and no planetary NP connected with a screw joint SB, which envelopes satellite carrier of the last transmission stage PL. Satellite carrier represents an overhang supporting the reducer. By conical rings it is directly connected to rotor shaft which leans over two roller bearings. The rotation of a reducer around the rotor shaft axis is prevented by the lever for fixation of FB.

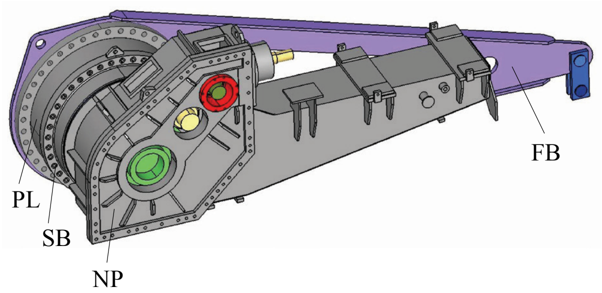


Figure 1 3D model of traction gear unit

Fig. 2 shows cross-section of the reducer and satellite carrier SK connected with a bucket wheel shaft. The reducer, with its own weight, leans over bearings 1 and 2 on the satellite carrier SK and on the rotor shaft overhang. In supports 1 and 2, roller bearings of corresponding dimensions are mounted. By the effect of force F_g=541kN, due to the reducers' net weight, bearing 1 is subjected to the force F_1=904 kN, and bearing 2 also to the vertical force F_2=361 kN. Calculated service life of bearing 1 under this load should be 60 000 hours, but it was destroyed after 1350 hours of operation.

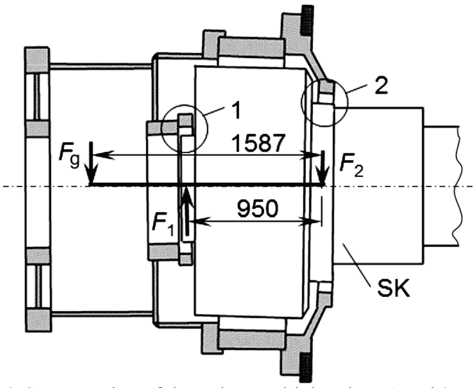


Figure 2 Cross-section of the reducer with bearings 1 and 2 positions

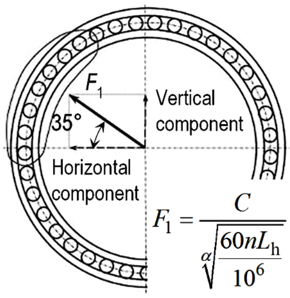
There are several traces pointing out that failure has appeared due to long-lasting surface fatigue of the bearing. The surface layer of a rolling path on the outer bearing ring was progressively roughened by the separation of particles. Due to pitting, bigger cracks have penetrated into the depth of the rings' wall, rapidly spreading and breaking off bigger pieces of ring until the biggest part has not been taken away. Besides bearing ring, rolling bodies have been also damaged, while some of them have been smashed into pieces or metal dust. Broken parts and damaged rolling bodies are shown in Fig. 3. A pile of metal sawdust gathered by oil flows for forced lubrication.

Figure 3 Failure of external bearing ring and pitting dust collected by oil flow

Failure is a consequence of surface fatigue which is confirmed by a small pile of sawdust found in a reducer. Although bearings transfer only the reducer weight, failure has appeared on one smaller part of rings shape as a cause of great force action, whose direction and course are closer to horizontal than vertical plane (Fig. 4).

Reverse bearing calculation shows that the magnitude of force which has led to failure is 2800 kN in the course of complete service life of L_h = 1350 hours.

Dynamic carrying capacity of the bearing is C=2×10^6 N in the course of 10^6 revolutions and speed of rotation is 3,9 rpm. The static carrying capacity is also much higher than the nominal magnitude of force F_1= 904 kN.



$$F_1=\frac{C}{\sqrt[\alpha]{\frac{60nL_h}{10^6}}}=\frac{2\cdot 10^6}{\sqrt[3,33]{\frac{60\cdot 3,9\cdot 1350}{10^6}}}=2,8\cdot 10^6\,\mathrm{N}$$

Figure 4 Failure position and destructive force value

Observation of the damaged area on the bearing outer stationary ring shows that the force which has caused failure has direction under the angle of nearly 35 degrees to horizontal plane, i.e. acts under the slope, see Fig. 4. This means that in vertical direction the force of 1606 kN acts, and horizontal component is 2293 kN. Since the nominal value of vertical component of reducers weight is 904 kN, and horizontal component should not exist, it is necessary to determine the causes for these additional forces appearance.

2.2 Gear drive unit of BWE SRs-1301

The input gearbox at bucket wheel of bucket wheel excavator SRs-1301 has five stage transmissions with fixed shaft axes. At the second stage, there is a pair of spiral bevel gears, and all other stages consist of helical gears. The output shaft is connected with bucket wheel shaft by means of a flange. The bucket wheel shaft is one of the supports of the gearbox (Fig. 5a). The gearbox is driven by electromotor with the power of 400 kW and speed of rotation n = 1480 rpm. Nominal torque at the input shaft is T_1 = 2581 N·m.

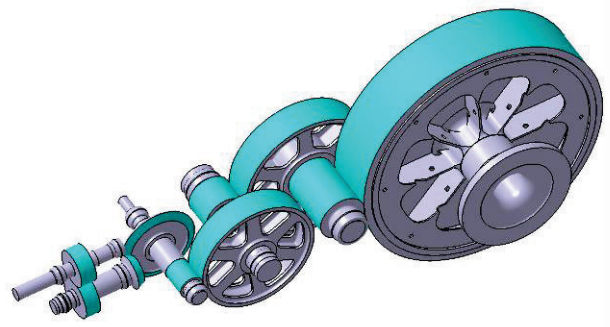


Figure 5a 3D model of traction gear unit of BWE SRs-1301

In the course of exploitation, the bearings of the second shaft were prone rapidly to failure. At once, there was even the failure of the teeth of spiral bevel gears that were torn apart. It was determined by calculation that all of the gears and bearings should not fail due to the service loads, even if they were intensively overloaded. Figure 5b shows the assembly of the second shaft and bearings. The damage occurred previously at spherical roller bearings. Due to the worn out bearings, internal clearance was extremely enlarged. Axial displacement of the shaft became huge. All these facts indicated large axial forces with variable direction.

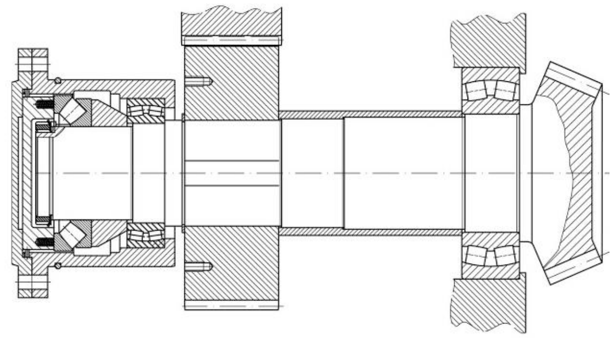


Figure 5b Assembly of the second shaft

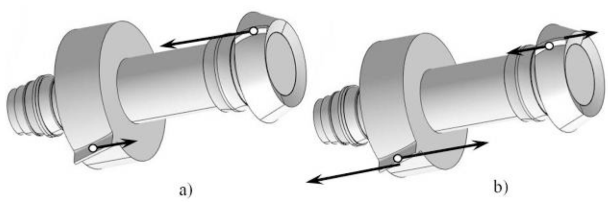


Figure 6 Axial forces at the second shaft: a) at stationary operation, b) at resonance

Axial forces applied to the second shaft for a given rotation direction are directed opposite to each other (Fig. 6a), and their difference is accepted by the axial bearing at the left support. Due to the resonance of helical gear pair, displacements of these gears are in anti phase. Teeth collide with each other very hard, vibrations are intensive as well, and axial force is with alternated direction (Fig. 6b). Spherical roller bearings are very sensitive to axial loads, and they fail quickly. Rapid failure of bearings is aided by axial supporting of the outer ring, which enforces the bearing to accept the axial force oriented to spiral bevel gear. When axial clearance in spherical roller bearings becomes higher than the lateral clearance between teeth of spiral bevel gears, axial resonance force from helical gears becomes accepted by bevel gears. Because of spiral shape of the teeth, the axial resonant force from helical pair acts on the teeth of bevel pair in alternative directions, leading to their failure.

Causes of the failure of bearings could be also additional loads induced by elastic deformations of shaft, bearing supports, housing walls or by natural frequencies of housing. These reasons led to systematic analysis of these components by FEM. Fig. 7 presents a 3D model of shaft meshed by tetrahedron elements, with extremely enlarged deformations. Fig. 8 shows deformations of the bearing nave. Stresses and displacements of all components are determined assuming the forces on gears that correspond to the nominal power increased by the magnitude of short-time overload (application factor K_A = 1,75). Shaft deformations are very small (highest displacement is 12 μm), so it could be assumed that those deformations did not affect self-adaptive bearings. The way of supporting of the bearing nave contributes to unequal displacement, which reflects on unequal distribution of radial force on two bearings in this support. Analysing the stiffness and natural frequencies of the housing by means of FEM, it is determined that the housing is rigid enough and is not the cause of the bearing failure.

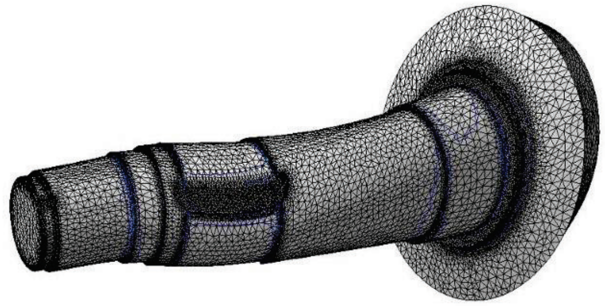


Figure 7 Elastic deformation of shaft caused by gear forces

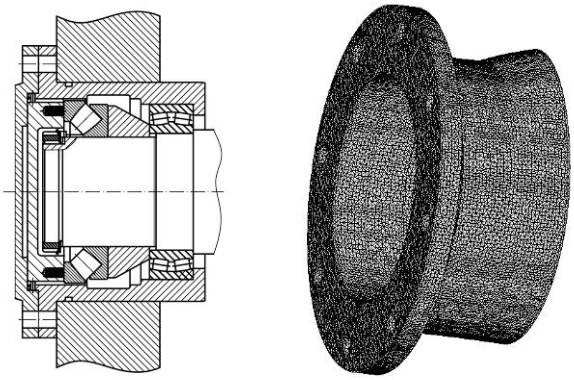


Figure 8 Supporting nave deformation

3 Effect of structure elasticity

3.1 Effect of gear unit housing elasticity

For the bearing calculation, it was assumed that the reducer case is a rigid body. Elasticity of structure can lead to the change of force direction turning it away to horizontal plane and structure conditions causing the increase of force magnitude. This fact is supported by the eccentric position of reducers mass with motor drive to the bearing meridian plane (Fig.1). Mass eccentricity generates additional elastic force on bearings. In Fig. 9 are shown deformations on the place of bearing 1 and 2, on the condition that there is no bearing and that on the place of bearing 2 there is a clamp. Due to these facts, reaction force in bearing 1 has been increased and inclined a little from vertical to horizontal direction. As a confirmation of this assumption, by applying FEM analysis, there have been calculated the magnitude and direction of force induced by those movements.

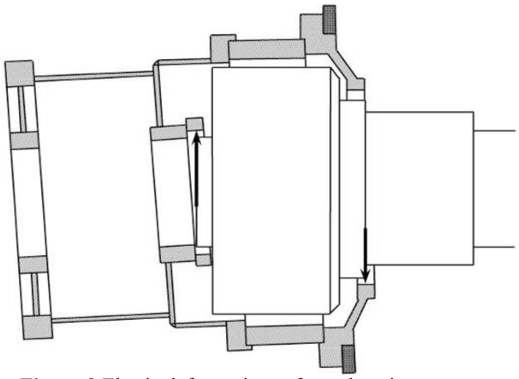


Figure 9 Elastic deformations of gear housing structure

3.2 Effect of torque fixing structure

To prevent reducer rotation around the exit shaft and the rotor shaft, the reducer is fixed by moment lever – fixing bar FB (Fig. 1) and 3 in Fig. 10. The lever is a strengthened and facilitated plate connected by screws with the reducer case (Fig. 10). With one free end, the lever is connected to the BWE carrying structure by crank connection that consists of teeter of two axles and two lateral plates (Fig. 1). The lever is to prevent the reducer rotation but, at same time, not to interfere with the freedom of adjustment to shafts in other directions and angles. If freedom is insufficient, the lever can become a source of additional load to the bearing. At dismantling of a reducer, the lever is found blocked in one of its marginal positions. In case of the deviation of rotor shafts position the free end of moment lever should move enough to the left side and enable the reducer to take the proper direction. In case of deficiency of unconstrained movement at free end of the lever a force will appear inducing the force which attacks bearings on satellite carrier in the horizontal plane. The magnitude of this force depends on the lever's rigidity and the size of free ends movement, which is prevented to happen. Under very big service load, rotor shaft endures elastic deformations under the effect of cutting forces on the rotor buckets. The shift size of the shaft overhang (satellite carrier) can be determined using FEM analysis for the known cutting resistance. The overhang movement must be also enabled by the lever free end. Under influences of forces of inertia the lever should also allow horizontal movements of a reducer under sudden oscillations of the whole system to the left or right side. This effect could be disabled by sufficient level of the horizontal movement freedom at the end of the lever, as noticeable in Fig. 10. If connection 4 does not allow freedom of lateral movement, reducer 1 cannot adjust to shaft 2 and by means of lever 3 horizontal force on the bearing is produced.

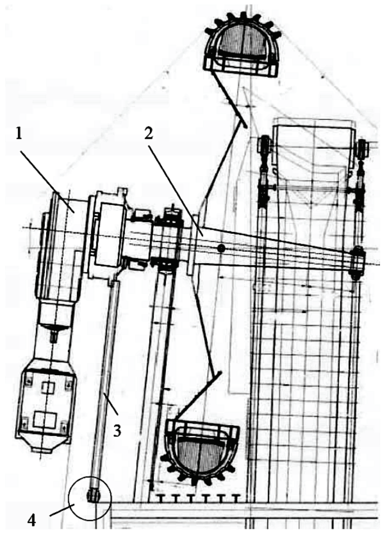


Figure 10 Reducer 1 on BWE with moment lever 3

3.3 Effect of elasticity in the bearing seat area

The local stiffness of structure, where the bearing is mounted, apart from the mentioned influences, can have effect on the bearing load. The analysis of the bearing seat area is done using FEM analysis. As a result, elastic deformations of bearing rings in reducers case and on satellite carrier are determined as well as the possibility of excentricity and oval shapes appearance. Also, a significant deviation in the parallelity of ring lines, between which the bearing is mounted, and ovality effects on loads distribution inside the bearing are all identified. Excentricity can be rectified only by additional force which additionaly burdens the bearing. In continuity of this analysis there follows the representation of the state inside the bearing caused by the interaction between the bearing internal parameters and external effects. The load that the bearing transfers is distributed on the rolling bodies behind the meridian plane perpendicular to the force direction (Fig. 11).

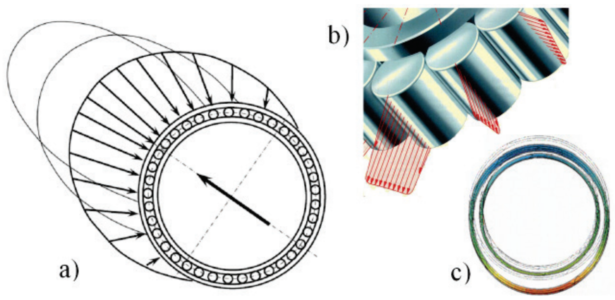


Figure 11 Effect of bearing set accuracy: a) inner bearing clearance, b) ring axes parallelism, c) ring oval shape and eccentricity

4 Effect of gear resonance
4.1 Gear drive resonance identification

Gear teeth meshing is a dynamical process which generates dynamical excitation forces, such as elastic variable forces and collision forces but also forces due to sliding and rolling of tooth flanks. Gears are specific for their characteristic to work in resonance, unlike the other mechanical structures whose amplitudes increase so much in resonance that could lead to failure in a short time. In the case of gear resonance, due to anti phase displacement of gears, teeth alternately collate by left and right flank, generating intensive alternate forces (Fig.12b and 6b). Dynamic factor K_v increase, gears can operate but produce very strong dynamic disturbance.

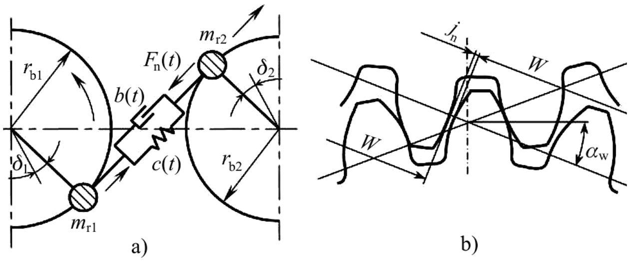


Figure 12 Dynamic model of gear pair in mesh (a) and flank clarence space for alternate colisions (b)

Basic parameters for calculating the resonance frequencies of gear pair in mesh is teeth stiffness and value of the reduced mass (Fig. 12a). According to DIN 3990 for helical gear pair at first stage of transmission, theoretical teeth stiffness is:

$$\frac{1}{c_{th}} = 0{,}04723 + \frac{0{,}15551}{z_{n1}} + \frac{0{,}25791}{z_{n2}} - 0{,}00635 \cdot x_1 - 0{,}11654 \cdot \frac{x_1}{z_{n1}} - \\ - 0{,}00193 \cdot x_2 - 0.24188 \cdot \frac{x_2}{z_{n2}} + 0{,}00529 \cdot x_1^2 + 0{,}00182 \cdot x_2^2. \quad (1)$$

Equivalent numbers of teeth and offset factor of these gears are z_{n1} = 56, z_{n2} = 63,4, $x_1 = x_2$= –0,198. For these values, theoretical teeth stiffness is c_{th} = 17,49 N/μm, mm which presents the force which can produce teeth deformation of one μm per one mm of gear width. The influence of elastic displacements in teeth contact is included multiplying by 0,8, and the influence of helical angle ($\beta_{1\text{-}2}$) multiplying by cos $\beta_{1\text{-}2}$, so real teeth stiffness is c' = 0,8·c_{th}·cos $\beta_{1\text{-}2}$ = 0,8·17,49·cos11° = 13,73 N/μm, mm. Average specific stiffness of mashed teeth is c_{th} = c'·(0,75·ε_α+0,25) for contact ratio ε_α = 1,88, c_γ = 22,79 N/μm, mm. Total average mashed teeth stiffness is $c=c_\gamma \cdot b$, for face width b = 150 mm, it is c = 3418 N/μm. Rotation mass of the first gear is the mass of the gear itself together with the shaft, rotation mass of input clutch with the rotor of electromotor, all of them with total moment of inertia J_I = 3,761 kg·m^2. The second shaft with gears 2 and 3 has the moment of inertia J_{II} = 5,147 kg·m^2. According to calculation of gear geometry, the radii of basic circle of gears 1 and 2 are r_{b1} = 177 mm and r_{b1} = 200 mm, so the reduced masses in direction of contact line (Fig.12a) and equivalent mass are $m_{r1} = J_I / r_{b1}^2$ = 3,761/0,177^2 = 120 kg, $m_{r2} = J_{II} / r_{b2}^2$ = 5,147/0,2^2 = 128 kg,

$$m_e = \frac{m_{r1} \cdot m_{r2}}{m_{r1} + m_{r2}} = \frac{120 \cdot 128}{120 + 128} = 62 \text{ kg}. \quad (2)$$

Natural frequency of the first pair of gears is

$$f_{rI} = \frac{1}{2\pi}\sqrt{\frac{c}{m_e}} = \frac{1}{2\pi}\sqrt{\frac{3418 \cdot 10^6}{62}} = 1181 \text{ Hz}. \quad (3)$$

Operation teeth mesh frequency of the first gear pair is $f_I = n_1 \cdot z_1/60$=1480·53/60=1307 Hz. The ratio between exciting and natural frequency f_I/f_{rI}=1307/1181=1,1 is in the range 0,85 ÷ 1,15, which indicates the full resonance of helical pair. For bevel pair (the second stage), applying the same procedure, natural frequency is:

$$f_{rII} = \frac{1}{2\pi}\sqrt{\frac{c}{m_e}} = \frac{1}{2\pi}\sqrt{\frac{1562 \cdot 10^6}{99}} = 632 \text{ Hz}. \quad (4)$$

and teeth mesh frequency is $f_{II} = n_3 \cdot z_3/60$ = 1305·23/60 = 500 Hz. The ratio between the exciting and the natural frequency f_{II}/f_{rII} = 500/632 = 0,792 is close to resonance, since it is close to the lower limit of 0,85 ÷ 1,15.

4.2 Gear drive design parameters harmonisation

Resonance of the gear pair 1-2 could be avoided if a significantly big difference were made between teeth mesh frequency and natural frequency of this gear pair. Natural frequency depends on teeth stiffness and magnitude of rotation masses. These magnitudes could not be significantly changed at the present design solution of this gearbox. The teeth mesh frequency depends on rotation speed and number of teeth. The rotation speed also could not be changed. There is only one possibility left, and that is to change the number of teeth (z) and gear module (m), so the gear center distance and transmission ratio remain unchanged. Increasing gear module from 7 mm to 12 mm and proportionally decreasing the numbers of teeth and helix angle from 11° to 8,1°, the center distance of 400 mm remains the same. For these changed parameters, following the same procedure, it is calculated natural frequency of f_{rI} = 1199 Hz and frequency of gear teeth meshing f_I = 749 Hz. The ratio between these frequencies is f_I/f_{rI} = 749/1199 = 0,62, which indicates that new parameters led helical pair out of resonance and led it into sub-critical service condition.

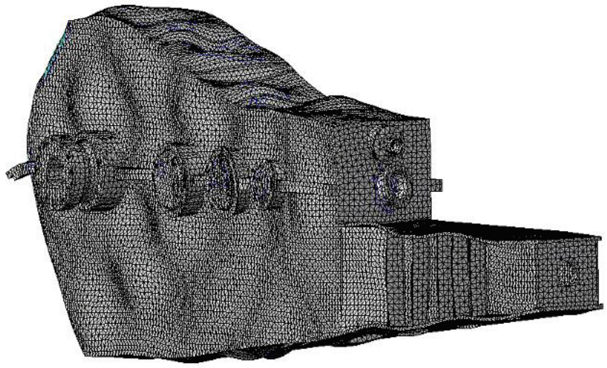


Figure 13 Modal shape of gear housing natural vibration with 1300 Hz

By this correction is avoided gear unit housing natural frequency of 1300 Hz (Fig. 13) which can produce and increase the level of gearbox noise.

5 Conclusion

For understanding of complex failure process and identification of its source, there has been conducted partial and mutual analysis of multiple influences which are the object of different research fields, concerning the current problem.

- Catastrophic failure of the bearing in gear drive unit of BWE SRs-1600 is the result of simultaneous effects of influences, such as missing of fixing bar horizontal freedom in joint with carrying structure, gear housing elasticity, local bearing set elasticity, bearing ring axes parallelism deviation and inner bearing clearance.
- A set of design solutions is suggested in order to prevent causes for bearing failure. The main solution is a redesign of reducer fixing bar joint. The second suggestion is stiffness increase of the reducer housing bearing area, and also increase of dimension and position accuracy of the bearing area.
- Resonance of gears at the first stage in gear drive unit of BWE SRs-1301 is identified as a cause of spherical bearings failure at the second shaft. The resonance generated alternated forces of high intensity, especially axial forces which led to rapid failure.
- By replacement of design parameters of helical gear pair at the first stage of transmission, resonance and dynamical loads generated by resonance were eliminated. Also, matching of frequency of teeth meshing with natural frequency of housing is eliminated and smooth work of gearbox is provided.

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