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Analysis of damage origin of bevel gear wheels

Silvia Maláková[a], Samuel Sivák[a], Gabriel Fedorko[b,*], Vieroslav Molnár[c], Miroslav Tomek[d]

[a]*Faculty of Mechanical Engineering, Technical University of Košice, Letná 9, 042 00 Košice, Slovak Republic*

[b]*Faculty of Mining, Ecology, Process Control and Geotechnologies, Technical University of Košice, Letná 9, 042 00 Košice, Slovak Republic*

[c]*Faculty of Manufacturing Technologies, Technical University of Košice with a seat in Prešov, Bayerova 1, 080 01 Prešov, Slovak Republic*

[d]*Faculty of Logistics and Crisis Management, Tomas Bata University in Zlín, nam. T. G. Masaryka 5555, 760 01 Zlín, Czech Republic*

[*]Corresponding author. *E-mail address:* gabriel.fedorko@tuke.sk (G. Fedorko).

ABSTRACT

The safety and smooth operation of manufacturing facilities also depends on ensuring the failure-free operation of technical equipment. Power transmission failures in particular are usually the cause of shutdown of the entire manufacturing facility. The characterization of gear failures according to their origin plays an important role in determining safe operating conditions. The presented paper is dedicated to the problem of damage of specific gears, defining the extent of damage, determining the possible causes of damage. The problem is solved for a single-stage helical bevel gearbox with, which is a part of the mechanism of a small hydroelectric power plant. The aim is to determine the conditions for proper operation, based on the evaluation of the extent of damage to this gearbox, so that the gearbox is no longer damaged to such a large extent.

Keywords: Mechanical engineering, bevel gearbox, steel, mechanical testing, wear, maintenance error, preventive maintenance

1. Introduction

The classification of gear failures according to cause is of particular importance because it makes it possible to determine the operating conditions that led to the damage. The American standard lists up to 22 different types of gear damage, but in our environment there is not yet a uniform classification.

Gear failures are very diverse. Important for gear dimensioning are failures that are fatigue and grinding at higher speeds or at high slip speeds [**1**].

Good lubrication in the fluid friction area is a prerequisite for smooth running of the gearing. If these conditions for the formation of a lubrication layer are not fulfilled, metal to metal contact occurs, which can lead to unacceptable wear by abrasion and later result to jamming.

Rapid wear will also occur in the presence of impurities in the lubricant that result in an abrasive effect. Protection is provided by filtering the lubricating oil or changing the oil frequently enough, especially after run-in. Also, loss of lubricating properties of the oil due to exceeded temperature limits leads to additional wear of the gearing, which is common for high-speed gearboxes where high local temperatures occur at tooth contact.

When gears with high contact pressures are meshed, the local temperature can exceed the permissible limit at inappropriate slip ratios or at high speeds and jamming occurs. This is explained as a result of the lubricating layer breakdown. At high temperature, tooth contact in the area of the frictional limit tends to lead to micro-welds creation and consequently to material separation due to the relative motion. Fine and coarse grooves are formed on the sides of the teeth in the direction of meshing according to the intensity of jamming, the surface condition, and the magnitude of the load.

Research was conducted by Korka et al. [**2**] to evaluate gear pitting using vibration signal analysis. In this work, a single helical gearbox integrated in an open energy test rig was used to analyze four pinions with various pitting grades. The findings improved knowledge of the effects of pitting development in gear systems and offered data for determining the stage of gear fault. It was determined that as the gears' pitting grade grew, and so did the vibration amplitudes at the meshing frequencies and their harmonics.

Research on emerging pitting and what contributes to its formation was conducted by Kopiláková et. al. [**3**]. Pitting damages were described, with an emphasis on C-C gearing. The Niemann's stand was used for three levels of working load testing. The major area of interest was the impact of pitting damage on the kind of gear. The experiment's findings were evaluated with those of involute gearing.

The mesh stiffness changes of a pair of external spur gears with tooth pitting were examined in study Lei et. al. [**4**]. A unique probability distribution-based model for tooth pitting has been proposed. In-depth analyses were conducted on the effects of tooth pitting on time-varying mesh stiffness, and the suggested model was confirmed by comparison with a finite element model.

Research on the vibrational properties of a single-stage spur gearbox with pitting was conducted by Sonawane et al. [**5**]. It was determined that root mean square (RMS) acceleration increased as the extent of pitting rose from modest to moderate or severe. It was determined that it was possible to anticipate the gear health state by determining its RMS acceleration.

In order to examine the vibrational properties of a spur gear system with a pitting failure, Deng et al. [**6**] presented a dynamics model based on three-dimensional line contact elastohydrodynamic lubrication (EHL). Through an iterative process, the 3D-EHL model and the transverse torsional dynamics model with gear pitting fault were connected in this model. The EHL has an impact on the vibration properties of gear systems with pitting failures, according to the results.

Li et al. [**7**] used the multibody dynamic program $ADAMS$ to simulate the crack propagation pattern of the broken gear system. The findings showed that as torques and fracture lengths rose, so did the amplitude of the harmonics of the meshing frequency and the sideband frequencies. Additionally, it was claimed that because to the tooth fracture fault's significant energy effects and quick increase in vibration amplitude, immediate energy was involved.

Bohme et al. [**8**] evaluated the suggested methodology's accuracy by contrasting it with load-controlled bevel gear tests at various hardening layer thicknesses. This study is a continuation of a previous study done by authors, where the model for carburized CrNiMo steel was used and assessed. Wheel-initiated tooth flank fracture was the main type of failure. Correlating the computed material utilizations resulted in the presentation of a repeating life factor. Under well specified test settings, the improved approach has demonstrated its ability to distinguish between and reliably predict pitting and subsurface fatigue.

A numerical model was developed by Liu et al. [**9**] to assess the contact fatigue damage progression of a megawatt-level wind turbine carburized gear pair. Using the elastohydrodynamic lubrication theory, the gear contact pressure was calculated. To estimate the fatigue, the Basquin equation and the Dang Van multiaxial fatigue criteria were used. Using the Palmgren-Miner rule, the gear damage accumulation under a range of loads was assessed. It was mentioned that the load spectrum should be taken into account while predicting fatigue life rather than only relying on a constant amplitude loading situation.

Selected issues with an investigation of damage to wind turbine planetary gear were presented by Bejger et al. [**10**]. The wear and failure of planetary gear has been examined by the authors. They claimed that although stray currents may not always result in electrical failures, they can significantly speed up the wear process in wind turbine gears. Additionally, oxygen depolarization, the root cause of corrosion "foci," may take place on the surface of meshed gears in a wind turbine.

The study of Liu et al. [**11**] provided a model including the wear process of a wind turbine gear pair's interface characteristics, mechanical properties, and residual stress gradients. Results showed that throughout the wear process, the critical damage location develops from near-surface to subsurface. This demonstrated that surface initiated contact failure gives way progressively to subsurface initiated contact failure as the most likely failure mechanism.

Castelani et al. [**12**] condition monitored a gear-based system in non-stationary operation located in wind turbines. New method for monitoring was proposed where the idea was that the vibrations were measured at the tower and not at the gearbox. Such a method would mean that there would not be any disruption to the wind turbine operation. Measurements were conducted on five wind turbines, where 3 were heathy ones giving the reference measurements, fourth one was freshly after planetary bearing fault and the fifth one was undergoing a high speed shaft bearing fault. It was concluded after procession of multivariate Novelty Detection algorithm and inspection of data by Principal Component Analysis that the proposed method yielded positive results, meaning that the monitoring at a base is viable.

Evaluation of measurement accuracy for three measurement systems was performed by Wrzochal [**13**]. Author evaluated the systems which differ from each other in key design features. A laser Doppler vibrometer was used as a reference point, which as the research shown, made possible to decouple vibration measurement indications from changes in dynamic state. The proposed system composed of two parts. First part was used for gaining a vibration spectra, second stage was used for comparison of data from reference sensor. This resulted in identification of the most accurate system and pinpoint undesired factors for measurement accuracy.

Grzeszkowski et al. [**14**] focused on integrated planetary gearbox in aero engines, where new method of monitoring was proposed. The monitoring was performed on gearbox during operation with the aim to obtain vibration measurement, sensor-dependent feature extraction and support-vector machine (SVM)-based classification. With the use of accelerometers, acoustic emission sensor and visual pitting surface, the two different SVM approaches were proposed. First one used Two-Class SVM and the second used One-Class SVM. Both methods were effective in pitting monitoring.

The presented paper is dedicated to the problem of damage of specific gears, defining the extent of damage, determining the possible causes of damage. The problem is solved for a single-stage helical bevel gearbox with, which is a part of the mechanism of a small hydroelectric power plant. The aim is to determine the conditions for proper operation, based on the evaluation of the extent of damage to this gearbox, so that the gearbox is no longer damaged to such a large extent.

2. Material and methods

The problem is solved for a single-stage bevel gearbox with curved teeth, which is part of the mechanism of a small hydroelectric power plant. Due to the frequent failure rate of the bevel gears, as well as to determine the damage, vibrations were measured on this assembly.

2.1. General description of a small hydropower plant

The subject of the damage investigation is a single-stage bevel gearbox, which is part of a small hydroelectric power plant on the Hornád River. Water from the Hornád River flows through the intake structure with the grid and cleaning machine installed and follows through the intake into the pressure feeder of the small hydropower plant building. The incoming water from the pressure feeder flows into the spiral chamber to the distributor wheel and further to the turbine impeller. The water is discharged back into the Hornad riverbed via a knee suction pipe. The building of the small hydropower plant is designed as a reinforced concrete structure with a flat removable roof part, and in the lower part with the installation of knee suction pipes. Steel beams are placed on the floor of the engine room for the purpose of positioning the gearbox of the small hydropower plant into the prescribed position and to ensure the required accuracy of the position of the gearbox.

The machinery of a small hydropower plant consists of two sets of the same, almost identical, separate units, designated $TG1$ and $TG2$ (**Fig. 1**), which each contain:

- Kaplan turbine with a diameter 1300 mm,
- the single-stage bevel gearbox with gear ratio $i\ = 3.08$,
- asynchronous generator type $1VF$ 600A - 86E,
- couplings between turbine and gearbox, then gearbox and generator,
- a hydraulic aggregate including machinery for the blades tilting of the turbine,
- machinery for pressure forced lubrication of the gearbox.

Water is taken from the intake building at a flow rate of 6-18 $m^3.s^{-1}$. The extracted water is fed through a pressure feeder to the blades of Kaplan vertical turbines (2 pcs), which transmit power through a shaft to the generator. The shared device for $TG1$ and $TG\ 2$ is:

- electrical device consisting of electrical switchboard $RM1$ and $RG1$,

- control system, for regulation and measurement of the small hydropower plant with the possibility of automatic operation of the small hydropower plant as a whole, or manual control of the operation separately.

Fig. 1. Hydrogenerator of small hydropower.

The technical design of the small hydroelectric power plant is implemented in such a way that it does not require permanent operation. It is a dedicated technical device and a dedicated electrical device that must satisfy a number of standards and regulations.

The parameters of the small hydropower plant are:

- the installed power capacity is 2 × 315 kW at the clamps of the generators in the low-voltage system,
- the equipment supplier gives a lifetime of 50 years for the rotating parts of the turbine,
- noise in the engine room 60 dB,
- water flow rate 6-8 $m^3.s^{-1}$.

The machinery of a small hydroelectric power plant, consisting of two sets $TG1$ and $TG2$ (turbine, gearbox, generator), operates symmetrically throughout the year and is in operation for almost the same time. It is a three-shift continuous operation throughout the year and also in operation all the time except for emergencies. The operating time is an average of 10 months of continuous three-shift operation per year.

After 5 years of continuous operation, only the “$TG2$" gearbox was damaged (the gearbox “$TG1$” were not damaged). The gearbox was adapted for the given application from the catalogue standard design to the specific conditions according to the requirements of the small hydroelectric power plant project. It is a KLN 400 type single-stage gearbox with helical bevel gears. The gear ratio is $i = 3.083$, the gearbox transmitted power $P = 330$ kW, at input speed $n_1 = 758$ rpm and output speed $n_2 = 246$ rpm. The number of teeth of the bevel pinion is $z_1 = 12$ and the number of teeth of the meshing bevel gear is $z_2 = 37$. The oil change is prescribed after 10 000 h of operation. Oil with viscosity $v = 44$ mm^2.s^{-1} at 50 °C was used. The material of the gear case is cast iron. The bevel pinion is an integral part of the input shaft, the output shaft is hollow. The total weight of this single-stage bevel gearbox is 2300 kg.

The gear transmission transfers power from the water Kaplan turbine (drive side) through a coupling on the hollow wheel shaft to a meshed pinion through a shaft and spring coupling to the generator shaft, where the electric current is generated.

2.3. Vibration measurement

Due to the increased noise level of the TG 2 of the small hydropower plant, the vibrations on this plant were measured. A schematic of the measurements with the marking of the measurement points is shown in **Fig. 2**.

Total vibration, FFT spectra and Time record were measured. Measurement - data collection was carried out for each of the seven measurement points in three mutually perpendicular directions (H - horizontal, V - vertical, A - axial).

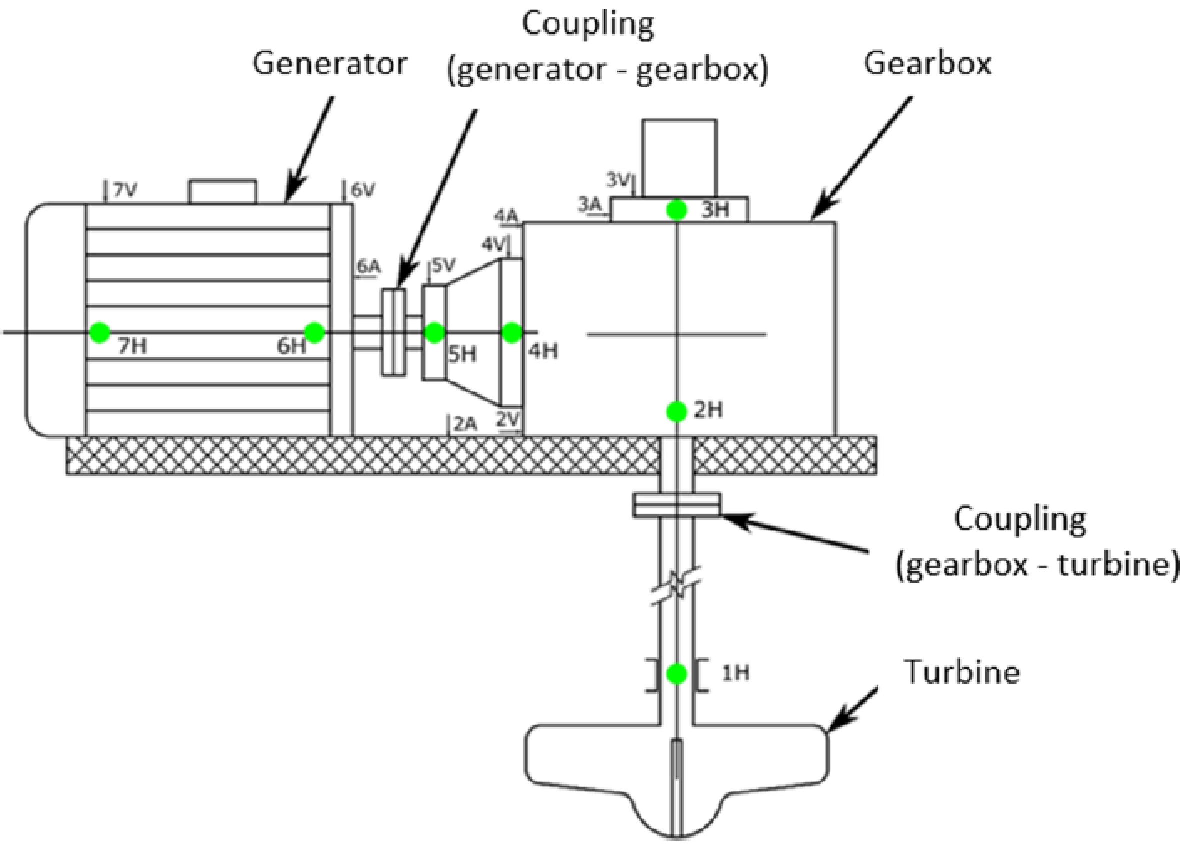

Fig. 2. Measurement diagram with marking of the measurement points on the $TG2$ turbogenerator (1-plain bearing, 2 - bearing of the vertical transmission shaft at the output, 3 - bearing of the vertical transmission shaft, 4 - bearing of horizontal transmission shaft near to the bevel pinion, 5 - bearing of the horizontal transmission shaft near to the coupling, 6 - the place on the generator closer to the coupling, 7 - the place on the left on the generator, measured directions: H - horizontal, V - vertical, A - axial).

The following measurement methods were used:

MFV effective vibration velocity value (10-10 000 Hz, unit mm.s^{-1}). This method can be used to detect dynamic unbalance of rotating masses, misalignment of shaft connection by couplings, loose or insufficiently rigid base, bent shaft and resonance phenomena. Each rotating component produces a vibration at a specific frequency, so the specific frequency points to the component. Increasing magnitude of machine vibration indicates deteriorating machine condition. A significant frequency and a rise in amplitude at that frequency indicates a component failure.

$EN3$ vibration acceleration envelope (500-10 000 Hz, unit E.g.). This method can be used to evaluate the operating condition or damage of rolling bearings and gears. Acceleration enveloping is a two stage process. The first step is to apply a band pass filter to the mix of low and high frequencies of a defective bearing's unfiltered waveform. This isolates only the frequencies in which the signal of interest is hiding. The filtered output will identify repeating, high frequency signals. On paper, this process would be represented as a series of spiking energy bursts energy (see the graph below); these are the impacts from the rolling elements hitting the defect of the rotating bearing. The second step in the process is to pass the filtered output through an enveloper, which rectifies (or demodulates) the waveform, by inverting the negative part to positive, and extracts the repetition rate of the energy bursts. This 'envelope' is now used as a true vibration signal - helping it to stand out from the noise. The envelope helps to contain regularly spaced signals, such as a single defect on a raceway, but other causes of noise, such as shaft rub, are random - so will not produce evenly spaced peaks.

Time Acc time record of acceleration, used to evaluate the correct meshing and damage to the gearing. **Table 1** shows the recommended alarm levels of the main measurement methods for the $ALARM1$ (warning) and $ALARM\ 2$ (danger) levels, which are specified by the manufacturers of the individual parts of the turbogenerator.

Considering the practical requirement to find out the causes of damage to the bevel gearbox of a small hydroelectric power plant, the paper will be devoted to the issue of damage to this bevel gearbox.

3. Theory/calculation

Gear teeth are deformed under the influence of load. Defining individual types of gear tooth damage plays an important role in defining incorrect operating conditions.

3.1. Failures of gears

Gear failures are very diverse. Depending on the cause, the failures are divided into two groups, namely damage to gearing surfaces and damage to gears by tooth failure.

Damage to gears can be caused by improper lubrication conditions of gears [**15,16**], impact stresses, incorrect meshing conditions [**17**], material defects, as well as design and technological defects. Damage to the gears can possibly be caused during design, manufacture, assembly and operation. Damage to gears is also manifested by an increase in the noise level of the transmission mechanisms.

Damage to the gearing surface can be divided into:

- wear (slip, abrasion, interference, corrosion, erosion or heating),

- galling,
- plastic deformation caused by rolling, knocking or the action of foreign bodies,
- pitting, pitting corrosion (running, progressive, micro pitting) - a manifestation of surface fatigue of the tooth flank (most commonly with unhardened surfaces),
- spalling, spalling of the surface layer in scales, larger areas or crumbling - a manifestation of surface fatigue of surface-hardened gearing.

In addition to the aforementioned failures, there are technological failures (abrasive cracks, cupping cracks, cracks from forging or casting).

The classification of gear failures according to cause is of particular importance because it makes it possible to determine the operating conditions that led to the damage. The American standard lists up to 22 different types of gear damage.

When calculating the strength of the gearing, the bending and contact load capacity is generally taken into account. For gears with hardened gearing flanks, the limit state is usually fatigue fracture, especially for cemented and hardened gearing.

Table 1 Recommended alarm levels.

Measurement method	Turbine and gearbox Measurement points 1, 2, 3, 4, 5		Generator Measurement points 6, 7	
	ALARM 1 warning	ALARM 2 danger	ALARM 1 warning	ALARM 2 danger
MFV [$mm.s^{-1}$]	4.5	11.2	4.5	11.2
EN3 [gE]	4.5	11.2	4.5	11.2
Time Acc [g]	4.5	11.2		

For gears with hardened and soft gearing, the limit state is fatigue wear of the surface layer (pitting). Due to the severity of the fracture failure of gear teeth, a higher reliability is often required due to the level of safety against fatigue fracture and therefore the load capacity criteria cannot be determined in a completely definitive way.

3.2. Tooth fracture

If the load is exceeded by a large amount, for example by an unexpected large impact, tooth fracture can occur due to overloading. The fracture surface is granular because it is most often a static fracture [**18**]. These extraordinary, unexpected effects in working operation cannot be fully taken into account in the calculation, as this would lead to oversizing of the gear components.

Fatigue fracture can occur after a certain period of operation of the gearing [**19**]. The fatigue fracture surface is characterized by ray-like and contour lines from the point of fatigue fracture [**20**].

In **Fig. 3**a the fatigue fracture is at the tooth base. This fracture is typical in cemented gears under small overload. It starts at the region of maximum stress concentration. A smoother fatigue fracture surface is visible at the fracture face in the direction of the applied load force, continuing with a coarse fracture surface.

The fatigue fracture in the area above the tooth base (**Fig. 3**b) occurs most often at the point of contact fracture on the sides of the teeth. It occurs in hardened gears with a small layer thickness (nitrided, nitride-cemented gear teeth). When a unilateral force is applied due to uneven loading in a light fit, fatigue fracture of a tooth part can occur (**Fig. 3**c).

The cause of an early fatigue fracture can be cracks of technological origin, which are formed during hardening or grinding of the sides of the teeth. Also in surface hardening, it is necessary to exclude a change of structure in the tooth base, which acts as an indentation effect. Grinding of gears which are made without protuberance also produces residual stresses in the tooth base which are detrimental to bending fatigue.

Also, the reduction of tooth thickness by wear, increases stresses in the tooth base, which can exceed a threshold value and lead to tooth fracture.

3.3. Damage to tooth surfaces

The tooth flanks are subjected to cyclic contact stresses, which induce typical fatigue wear - pitting - when the limit value is exceeded and a certain number of cycles is reached (**Fig. 4**a). Pitting is usually circular in shape and is usually located in the negative slip region below the rolling point [**21,22**]. The onset of tooth flank damage is signaled by an increase in noise and drive vibration, which gradually increases. Deep pitting occurs in refined gearing with higher strength. Pitting can be delayed by increasing the strength of the gearing material or by using one of the surface hardening methods.

In steel gears of formed and cast soft wheels (material ultimate strength limit Rm $<$ 500 MPa) and in gears of quality alloys (ductile cast iron), a large number of small pittings often occur in the area below the rolling point under load over a wide interval around the contact fatigue limit (**Fig. 4**b).

Progressive, advancing pitting, which gradually leads to the destruction of the lateral teeth to an intolerable degree, must be distinguished from so-called recessive pitting, which disappears after a certain time of gearing activity, and this means that we consider this degressive - recessive pitting to be acceptable [**23**].

On the sides of the gearing of nitrided and carbonitrided gears, roughening often occurs in the area of allowable stress (**Fig. 4**c), which is related to peeling of the surface coating. Pitting often occurs later in this roughened area.

For soft steel and ductile cast-iron wheels, there is often occurrence accompanied with minor pitting of plastic deformation (**Fig. 5**a). This is a sign of gearing overload, shock and oscillation [**24**]. Plastic pitting can be observed in the rolling point region, plastic elongation of the tooth head means unbalanced slip ratios and large friction forces in this region [**25,26**].

The gear needs a certain period of time to run-in, which under normal conditions results in the adjustment of the meshed tooth flanks and the levelling of surface irregularities [**27**]. This does not continue after the initial wear, the tooth flanks are smooth and shiny in the contact zone. The normal wear that can be observed in soft steel wheels progresses very slowly and turns into a defect on further operation [**28**].

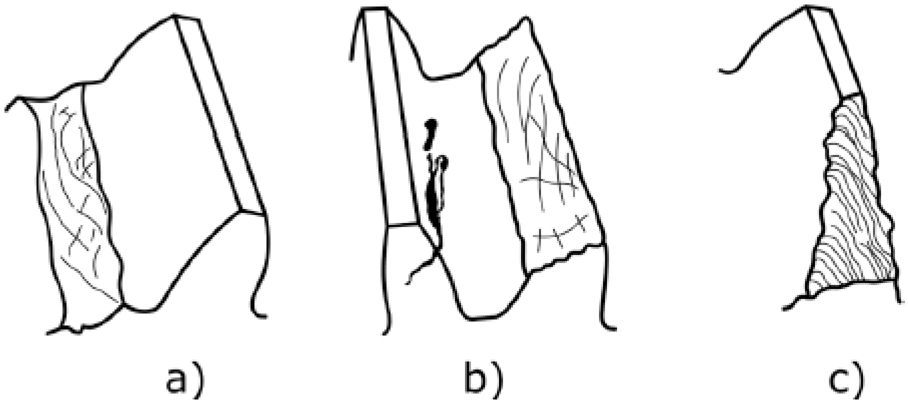

Fig. 3. Tooth fractures: a) fatigue fracture at the base of the tooth, b) fatigue fracture in the area above the base, c) fatigue fracture of part of the tooth.

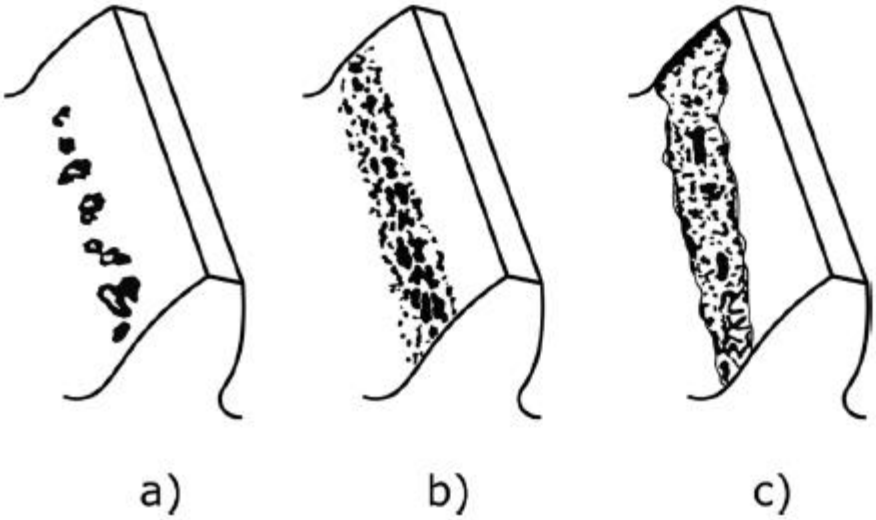

Fig. 4. Tooth pitting: a) Pitting - alloy steel ($R_m = 900$ MPa), b) Pitting - hardened cast iron, mild steel ($R_m < 500$ MPa), c) Pitting (nitrided gearing).

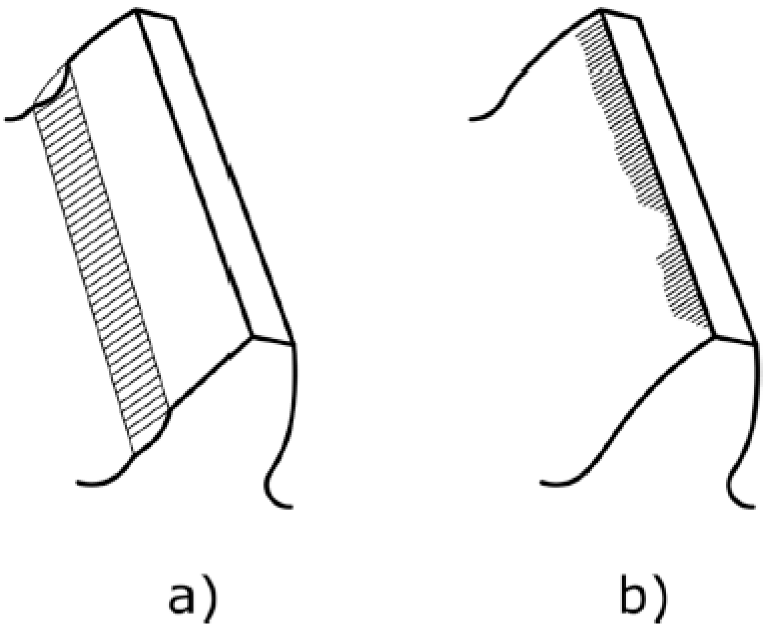

Fig. 5. a) Plastic deformation (ductile cast-iron, soft steel), b) Tooth flank jamming.

In contrast to fatigue contact damage to the tooth flanks, gear jamming often occurs during transmission run-in. Light jamming has a small range and is manifested by fine grooves in the slip direction. When the load is reduced and the run-in is gradual, it may disappear after some time. **Fig. 5**b shows the damage to the tooth flanks caused by jamming. The indentation marks on the heads of the teeth are indicative of misaligned ratios in this area and insufficient lubrication. The depth and character of the indentations depend on the hardness of the tooth flanks and on the level of loading.

Jamming that extends to the entire contact surface may result in the wheel being taken out of service. The working flanks of the teeth are often destroyed by coarse indentations and plastic creep.

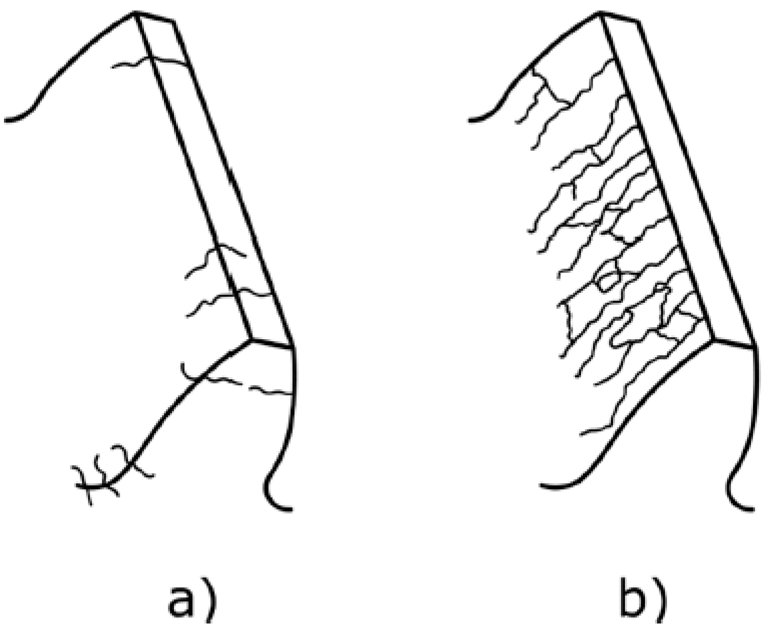

Fig. 6. a) Cracks after hardening, b) cracks after grinding.

We can limit the tendency of the teeth to jam by selecting the appropriate combination of material and lubricating oil, reducing the allowable pressure and designing a correction to compensate for the dimensional slips.

Cracks after hardening (**Fig. 6**a) most often originate from the edges of the teeth. It can be oriented in the direction of the meshing, but also in a direction perpendicular to the meshing. They occur in cemented and surface hardened gears [**29**].

In cemented gears with a supersaturated coating, grinding cracks occur on the flanks of the teeth (**Fig. 6**b).

4. Results and discussion

Based on the evaluation of the extent of damage to this assembly, it is necessary to establish the conditions of proper operation so that damage to such a large extent does not occur again. Material hardness tests on damaged gears were performed. The hardness met the requirements stated on the manufacturing drawings of gear wheels.

4.1. Transmission vibration measurement

The vibration measurements of the small hydropower plant of the $TG\ 2$ system were carried out at the maximum permissible load. The results of the measurements are given in **Table 2**, where the level of evaluation is verbal and thus the evaluation of the operating condition has the following four levels:

- good (no measurement has achieved this rating),
- satisfactory,
- unsatisfactory (requiring repair),
- emergency (immediate shutdown and immediate repair necessary).

The measurement results showed that the unsatisfactory to emergency condition of oscillation and vibration is on the pinion shaft. This means that it is a measurement in the horizontal direction at points $4H$ - bearing of horizontal transmission shaft near to the bevel pinion and $5H$ - bearing of the horizontal transmission shaft near to the coupling. The large oscillation of the pinion shaft in the bearings has resulted in oscillating motion of the gear orbit multiplied by the distance from the bearing which has largely deteriorated the gear meshing conditions.

The results of the measurement in the horizontal direction at points $2H$ - bearing of the vertical transmission shaft at the output, and $3H$ -bearing of the vertical transmission shaft, were unsatisfactory. The results of the measurement in the horizontal direction at points $1H$ - plain bearing, and $6H$ -the place on the generator closer to the coupling, and $7H$ - the place on the left on the generator, were satisfactory.

Fig. 7 shows the Time record of vibration measured at point $4H$ (see **Fig. 2**) - input horizontal shaft with bevel pinion.

Vibration measurements were also carried out on the gearbox in part $TG1$. The measurement results at all measurement points were satisfactory.

4.2. Damage to bevel pinion

On a spiral bevel gear, that is, on both the pinion and the wheel, the gearing has been damaged. As far as the bevel pinion is concerned, here it is obvious that several types of damage on the teeth occurred.

There are distinct, very large and prominent pits along the entire length of the tooth near the root of the tooth to a height of $^{1}/_{3}$ to $^{1}/_{2}$ of the tooth on each pinion tooth (**Fig. 8**). These are manifestations of surface fatigue. Damage is manifested by the removal of metal particles and the formation of cavities (pits). The pits are not due to wear, but to surface and subsurface stresses caused by the repeated application of forces. This damage occurs on contact as a result of rolling and sliding. This pitting corrosion occurs most commonly on heat-treated gearing. After breakage (chipping, peeling off) of the small metal particles of the active surfaces of the tooth, the surface is left dotted with distinctive pits. Commonly and most often this pitting is manifested in the area of the pitch circle. In this case, the pitting, the pronounced pitting, manifested itself near the root of the tooth, which means that, the meshing conditions were broken.

The spiral bevel pinion has distinct and deep grooves and indentations on the active sides of the teeth (applicable to all teeth of the pinion) in the part of the surface from $^1/_3$ of the height of the tooth up to the head of the tooth (**Fig. 9**). The formation of prominent grooves and indentations is due to plastic deformation and to wear of the tooth flanks.

This type of damage occurs on slow-speed gears where there is significant slip in the direction of the tooth contact line and generally in materials with low surface hardness. The pinion teeth are cemented and hardened according to the manufacturing drawing. The faces of all the teeth of the spiral bevel pinion are severely damaged by cold jamming. This type of damage is reversible as long as the mechanism of its origin is removed.

Table 2 Measured vibration values.

Measurement point	Level of evaluation	MFV [mm.s^{-1}]	EN3 [gE]	Time Acc [g]
Point 1H – plain bearing	Satisfactory	0.3	–	2.6
Point 2H – vertical transmission shaft	Unsatisfactory	1.2	3.9	8.6
Point 3H – vertical transmission shaft	Unsatisfactory	2.8	2.7	5.4
Point 4H – horizontal transmission shaft	Emergency	2.6	6.6	16.5
Point 5H – horizontal transmission shaft	Emergency	3.2	7.6	20.5
Point 6H – generator	Satisfactory	2.4	0.6.	–
Point 7H – generator	Satisfactory	2.8	0.9	–

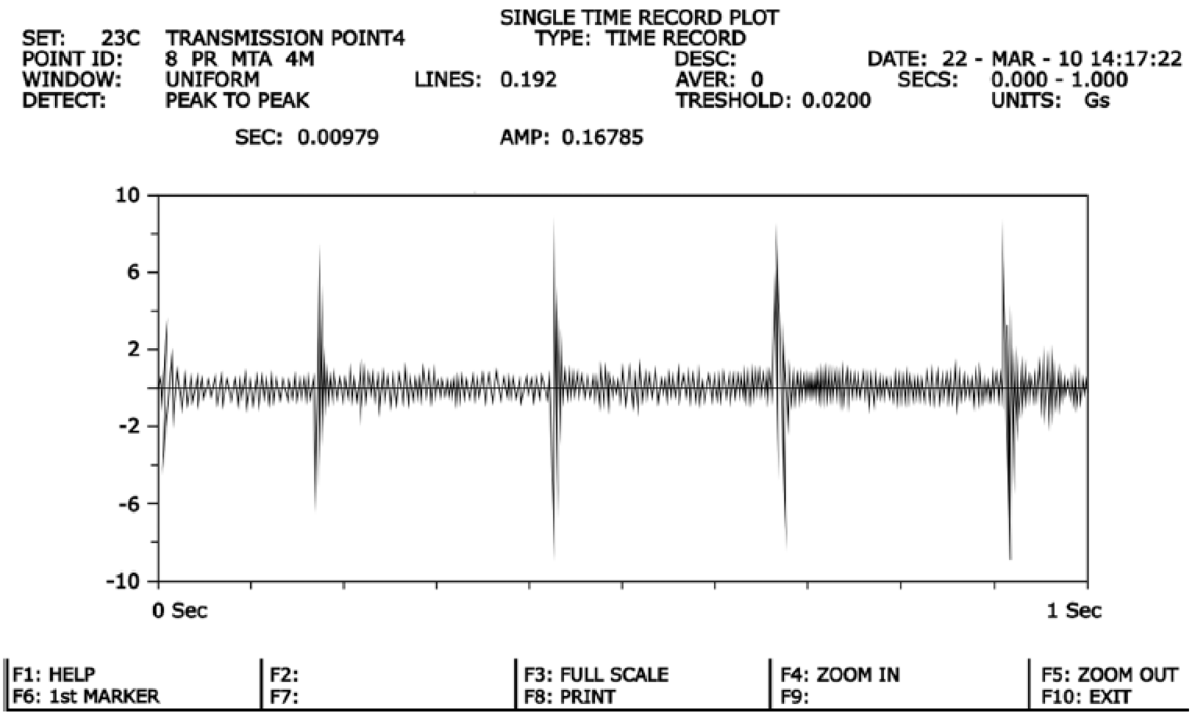

Fig. 7. Time vibration record - input shaft of bevel gearbox with bevel pinion.

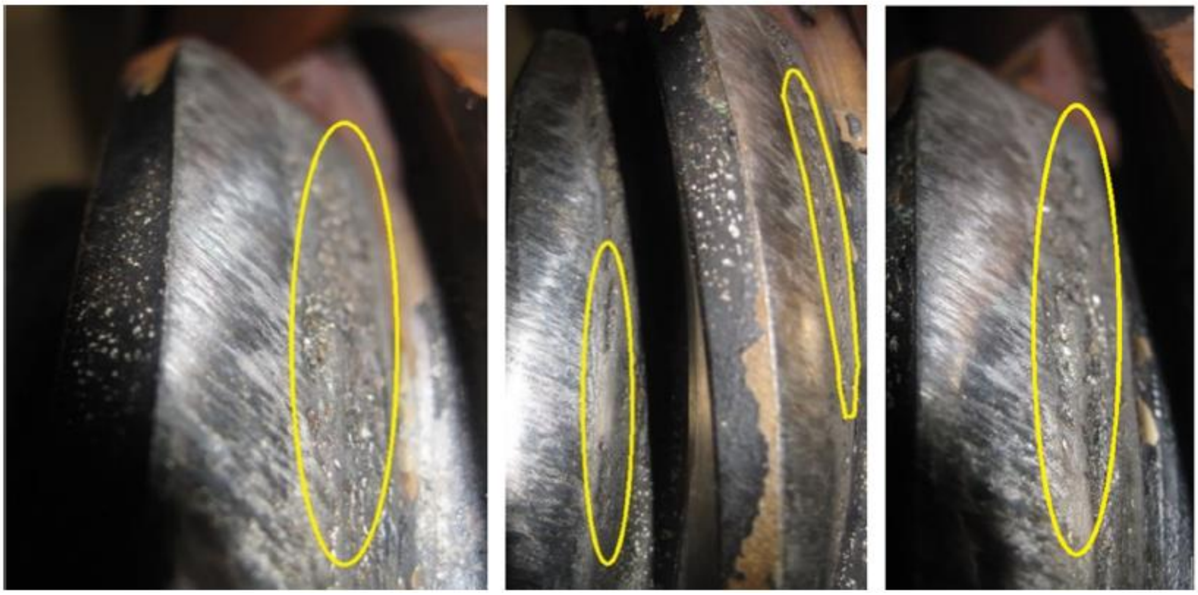

Fig. 8. Pitting on the gearing of a bevel pinion.

The removal of material is generally due to abrasion by substances contained in the lubricant (in this case a mixture of oil, paint and solid metal particles) and the mutual sliding of the two surfaces. This is not normal wear of the gearing due to a gradual change in the nature of the surface of the machined side of the tooth. If it were normal wear, the active tooth surface would take on a mirror-like shine with slight wear. At the same time, the temperature of the oil in the gearbox was reliably measured and monitored throughout the operation of the gearbox. Therefore, it is concluded that this damage could not have been caused by exceeding the permissible temperature.

Three teeth of the bevel pinion out of a total of 12 are damaged by breakage of an irregular shape part of the side of the tooth located at the head area of the tooth. It is a deep damage to the flank of the active tooth surface by breaking out a portion of the tooth to a thickness of up to $^{1}/_{3}$ of the tooth profile (**Fig. 10**). It is a sign of fatigue fracture.

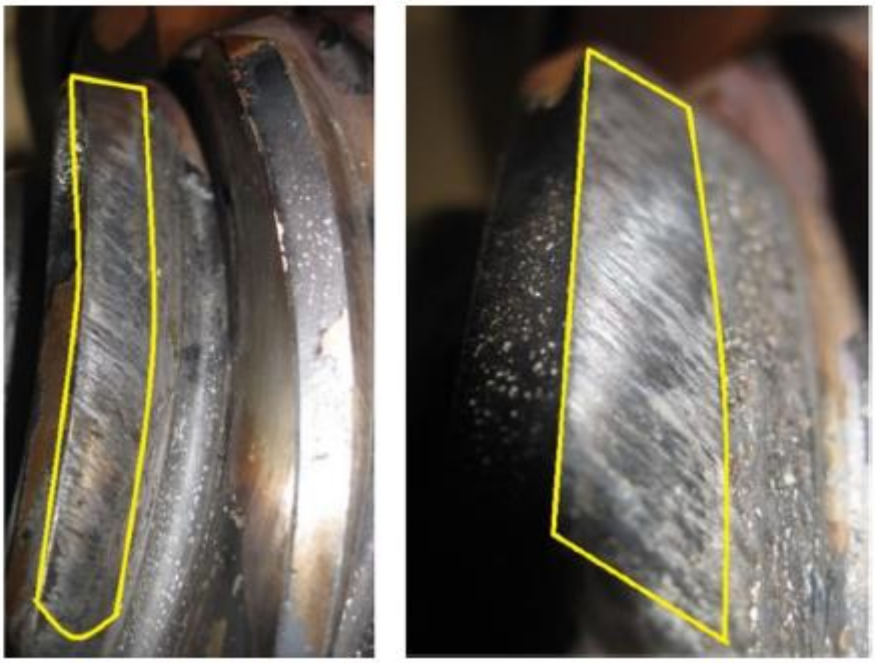

Fig. 9. Indentation - jamming of a tooth side.

4.3. Bevel gear wheel damage

There was extensive damage to the gearing on the spiral bevel gear, not a single tooth was left intact (**Fig. 11**). Some teeth are broken off across the width of the tooth. Lubrication nozzles used for lubrication of meshed teeth can be seen on Figure. More detailed damage to the gearing on the bevel gear is shown in **Fig. 12**.

The most destructive damage includes fracture. Each tooth is damaged by fracture, where it is possible to distinguish the following cases:

- Fracture of the tooth across the full width of the gearing (**Fig. 12**a). Three teeth out of the total number of teeth were damaged in this way. This is a fatigue fracture in the area above the root of the tooth, specifically in the area of the head of the tooth.
- On the gear, several teeth are broken off in the pitch area of the tooth length, from $^{1}/_{4}$ to $^{1}/_{2}$ of the tooth length (**Fig. 12**b). The cause of tooth fracture is fatigue damage to the material as a result of repeated bending stress on the tooth. The fracture starts on the tension-loaded side of the tooth by propagation of a small crack over a large number of loading oscillations. The fracture surface of the tooth shows two distinct zones, namely the initial and final fracture, with typical fatigue (fine-grained) and force (crystalline, rough) parts.

Fig. 10. Breakage of part of a bevel pinion tooth.

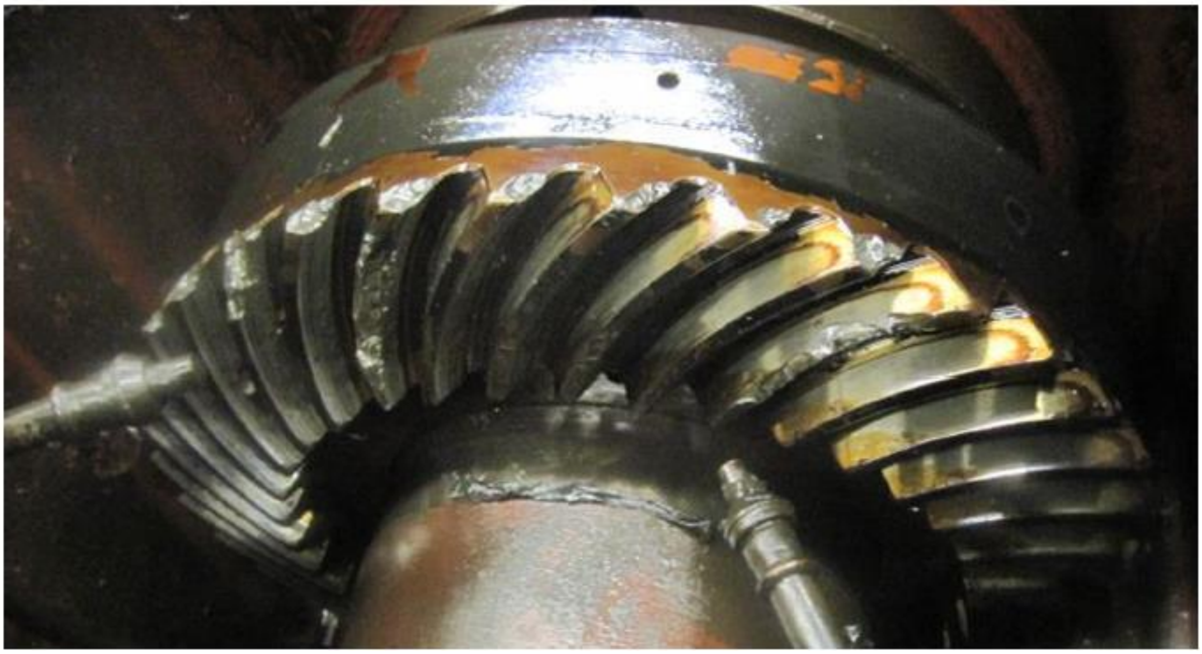

Fig. 11. Damage to the bevel gear gearing.

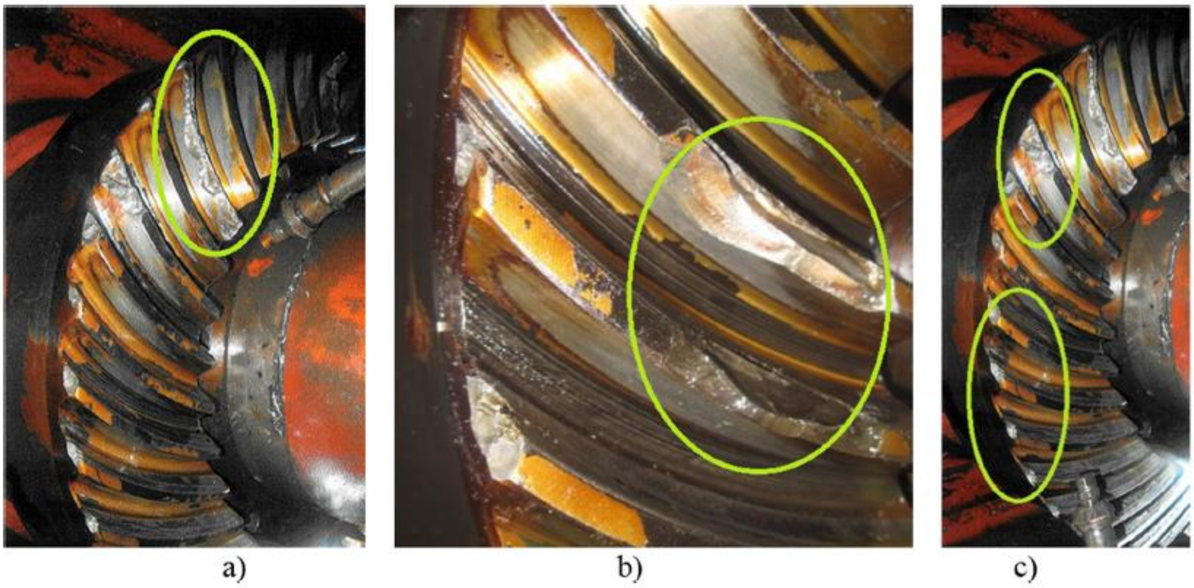

Fig. 12. Damage to bevel gear gearing a) fracture in the area of the tooth addendum across the width of the tooth, b) fracture in the middle of the width of the gearing, c) fracture of the tooth at the end of the width of the gearing.

In the fatigue zones, there are no obvious signs of plastic deformation. The surface of this part of the fracture is smooth, matte in appearance, with noticeable lines of a kind of crack propagation process.

- All gear teeth have broken off chunks of the tooth ends on the addendum side farthest from the centerline of the wheel-shaft (**Fig. 12**c). They are all bend stressed by variable loads at low cycle counts. When compared to pinion damage, there is no broken tooth and no chunk broken off at the head of a tooth. The pinion tooth ends are damaged and worn as are the teeth along their length. Fracture marks in the same type of fracture can vary widely from one location to another even over a small observed section. The fracture position of some teeth is predominantly granular, characterized by poorly visible plastic deformation in the part of the fracture that is granular. The part of the fracture surface near the center of the fracture surface is duller, less or not at all granular with an elevation. The origin of the wear is at the location of the surface hardened layer. The fracture can identify the shear point, the plastic deformation point and the vulnerable fracture surface. These are predominantly semi-fragile to ductile fractures.

The origin of the fracture is ultimately due to external stresses, which in this case have exceeded the strength or fatigue limit of the material. The specification of the influences that led to this is as follows:

- change in material properties over time (material fatigue),
- uneven loading of the tooth (only part of the tooth has been broken out) as a result of improper gear meshing,
- deformation of the tooth due to impurities (tooth fragments dispersed in the lubricant),
- impact of poor maintenance e.g. timeliness and suitability of oil changes; poorly suited operation (14' shutdown time without lubrication),
- improper heat treatment (increased brittleness),
- improper shaft alignment and thus built-up vibrations,
- manufacturing defects (tension caused by notches due to non-compliance with technology),
- inadequate design (inappropriate shape or dimensions; or inadequate elimination and capture of forces from rotating parts of the turbine).

Due to the fact that only a part of the tooth has been broken out, this is an uneven loading of the tooth. On the basis of investigation and examination, the tooth failure occurred alongside the path of the formation and development of cracks (as a result of fatigue). The base of the micro-mechanism of breakage is the gradual, time-staggered loss of bonding between atomic pairs. This statement can only be confirmed with certainty by metallographic analysis, which has not been carried out. Fractographic analysis of the fracture surfaces was used in the investigation. This is partly because multiple views of the same fracture at different magnifications must be chosen for a complete characterization of the fracture. Fractures at low magnification were investigated, where the cause of their formation, the toughness or brittleness of the material was examined.

Like the damage to the teeth of the bevel pinion, the teeth of the bevel gear also show signs of pitting damage as well as pitting damage to the sides of the active tooth surface. There are significant grooves and indentations on the active tooth flank. This type of damage is most common in slow-speed gears. A significant slippage in the direction of the gearing contact line has shown.

4.4. Damage to the gearbox body

The gearbox remained intact and undamaged. An external impact on the body is excluded. Inspection of the gearbox, already dismantled, revealed that:

- there was no lubricating oil in the gearbox,

- Inside the body, on the sides as well as on the bottom, there was a 2-3 mm thick layer of paste stuck to the sides, made of lubricant -oil, peeled paint, crumbs and metal filings from the wheels, as well as small and larger metal parts of the metal of the gears of the wheels (**Fig. 13**).

In this condition, these bits, metal particles and semi-solid mixture were getting into the inter-tooth space when the gearbox was operated and the teeth were pushing away from each other, which significantly worsened the meshing conditions of the gear. Such a considerable amount of metal filings, crumbs and peeled off paint was formed gradually, over a long period of time. It is safe to say that this is not a random and short-lived phenomenon.

5. Conclusions

Damage to gears is the result of a gradual long-term process of wear in the interaction of several factors.

One of the factors is insufficient lubrication, that is, insufficient formation of an oil film on the teeth. This was created when the equipment was shut down according to the operating book for 14 min. The pressure lubrication does not work during the run time (until the complete stop). At the same time, there were long periods of time between oil changes.

Another factor is the peeling of paint from the inner wall of the gearbox, which was getting into the inter-tooth space and deteriorating the lubrication (scales of hardened paint were pushing the oil out of the tooth contact area and disrupting the integrity of the oil film).

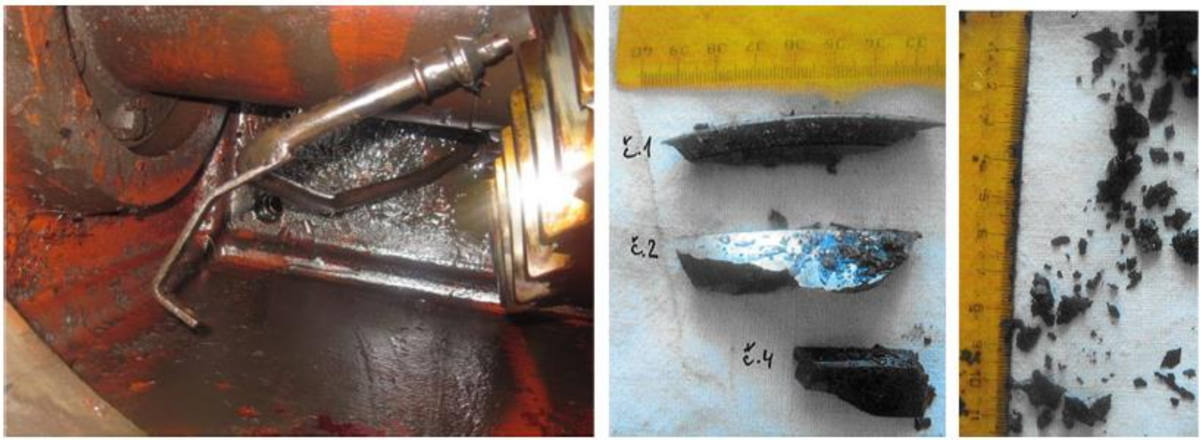

Fig. 13. Impurities and fragments from the gearbox.

Temporal changes in material properties, mainly on the tooth surface, as a manifestation of surface fatigue of heat-treated gearing. As a consequence, small parts of the teeth were separated, which, together with the oil, and scaly paint particles, formed an ’abrasive paste’. As a result, abrasive wear began to occur.

The formation of numerous and relatively perfect adhesion micro-joints as a result of imperfect lubrication, which led to the formation of loose abrasion particles. The condition for this wear is the slippage of the surfaces over each other. This occurred when the meshing ratios were changed, and the teeth slid over each other in addition to the rolling motion.

The occurrence of significant changes in the gearing meshing ratios, as well as the occurrence of additional stress on the teeth with the formation of impacts due to oscillations - vibrations. This caused additional stress on the gearing by the disappearing cyclic load. If damage due to overloading occurred in the gearing of the meshed gears, then fracture damage to the gearing would occur in both meshed gears, i.e. in the wheel and pinion. In the case of a pinion, only one tooth is damaged by partial fracture of a tooth. Therefore, a sudden and abrupt load change (load overrun) between the meshed gear teeth is dismissed as unlikely.

The gearbox failure was not sudden, and completely unexpected. Long before the failure that led to the gearbox being taken out of service, the equipment was showing signs of changes in its operating condition. These changes include, for example, increased mechanical vibration values, higher or slightly increasing temperatures, increased noise, impurities in the work fluid. The operator did not apply predictive maintenance methods at all.

The resulting failure and damage to the gearing of this magnitude could have been prevented or avoided on the basis of a continual up-to-date objective knowledge of the actual technical condition of the gearing. The inspection could have been carried out and the actual state of the gearing could have been established by visual inspection of the gearing through the inspection holes in the gearbox (the operating mechanic testified that he had not carried out the inspection and was not even aware that he should have carried out such an inspection) during the shutdown. By observing whether the equipment exhibits increased mechanical vibration and increased noise when running under different loads, as well as when changing the lubricating fluid filter, paying attention to whether there is an increased content of unwanted impurities and contaminants in the working fluid.

Proposal of measures to prevent damage to the gearbox during further operation:

- shorten the period of time for changing the oil in the gearbox (instead of a two-year cycle, change the oil after a year and a half),
- ensure the lubrication of the gears during the stop process, i.e. from the start of shut down until the gearbox comes to a complete stop,
- make a visual inspection of the gearing through the holes in the gearboxes designated for this once a week,
- exchange the lubricating fluid filter more often,
- use a high-quality coating of the internal walls of the gearbox so that it does not peel off,
- checking the geometry of the bevel gearing,
- regularly monitor vibrations in critical places.

Further research will be focused on the development of an experimental and software tool based on the results of vibration from diagnostics without a need of disassembly, simplifies troubleshooting and prediction of possible causes of gear train failures.

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