

Strength analysis of an electric locomotive transmission mechanism

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ABSTRACT

The transmission of traction electric motor torque to the electric locomotive wheelset is carried out through a wheelset reducer with cylindrical gears with straight or inclined teeth. The wheelset reducer is one of the most important units of a modern electric locomotive. It bears significant alternating loads when the locomotive starts, accelerates and stops. It is of paramount importance to choose the right drive option, which the locomotive movement and dynamics of its running depend on. At high running speed, the system must be reliable in terms of the oscillations of locomotive unstrung masses. The thirty-year operation of electric locomotives in Bulgaria has shown a number of shortcomings of wheel reducers that cause difficulties with operation and repair activities in locomotive depots. The paper presents a strength analysis of a transmission mechanism, that has shown low operational reliability leading to locomotive derailment. The results of analysis have indicated the areas, which need monitoring in operation to prevent the occurrence of incidents and accidents due to the transmission mechanism operation.

KEYWORDS

Railway, electric locomotive, transmission mechanism, strength analysis

1. INTRODUCTION

The mechanisms that transmit drive torque from the traction electric motor shaft or the output shaft of a mechanical, hydraulic or other transmission system to wheels, without violating the freedom of movement necessary for relative displacement of wheels in regard to the bogie frame and of the bogie in regard to the frame, are marked as transmission mechanisms [1]. Their structural elements are reducers, shafts, joints, supports, couplings, etc.

The locomotive technical and economic parameters significantly depend on transmission mechanisms. This is the reason for a number of requirements put to them. Summarising, the more significant ones are expressed in the following:

- the transmission mechanism operation should not be accompanied by appearance of high-value dynamic forces and should not limit the operating modes;
- the mechanism reliability should be high and meet the requirements for economic efficiency;
- it should provide high operating economy;
- it should not possibly reduce the space intended to install the traction motor;
- the design should be simplified and should allow the arrangement of traction motors in a way to be accessible for inspection and repair and protected from snow, moisture and dust.

The transmission mechanisms can be divided into two groups depending on whether they drive a separate wheelset or a group of wheelsets of mechanisms for individual drive and for group drive. The constructive implementation of transmission mechanism depends on whether it transmits torque from the shaft to a traction electric motor, or from the shaft at the output of a hydraulic or mechanical transmission system, therefore the transmission mechanisms are divided into two groups on this basis: transmission mechanisms for drive with a traction electric motor and transmission mechanisms with hydraulic or mechanical transmission systems.

The construction practice has shown that usually the transmission mechanisms for drive traction electric motors belong to the group of mechanisms for individual drive, and the transmission mechanisms in hydraulic or mechanical transmission systems – to mechanisms for group drive. The opposite is rare, but the group drive of wheels with an electric motor is perspective for development.

The kinematics scheme and design of transmission mechanisms depend on the method of mounting the traction motor on a bogie or, as it is accepted to call that, on the method of traction motor suspension. Due to the above mentioned, the method of suspension is taken as a classification feature and based on that, the transmission mechanisms for drive wheelsets by traction motors are divided into two groups: mechanisms with a support-axle suspended traction motor and mechanisms with a support-arm suspended traction motor.

The locomotive transmission mechanisms transmit torque from the locomotive engine to the wheelsets, driven by it (when the engine is in traction mode) and vice versa (when the engine is used to stop the locomotive).

The type and structure of power transmission is determined by the locomotive engine type, power and properties, the method of its suspension, its location in relation to the locomotive longitudinal axis and type, purpose and maximum speed of the locomotive itself [2].

2. CLASSIFICATION OF TRANSMISSION MECHANISMS

The locomotive power transmissions can be classified as follows [2 - 7]:

1. *According to the number of wheelsets, which are simultaneously driven*, they are power transmissions for individual drive of locomotive wheelsets and power transmissions for group drive of locomotive wheelsets.
2. *According to their regulating properties*, they are non-adjustable (i.e. constant gear ratio) power transmissions and adjustable (i.e. with variable gear ratio) power transmissions.

The non-adjustable gears are the lightest, simplest and cheapest locomotive power transmissions (in terms of manufacturing, operation and repair). They belong to the category of mechanical power transmissions, since they transmit torque through gear meshing, lever systems or power chains.

The adjustable power transmissions are relatively complex and expensive, but unlike non-adjustable ones, they allow the adjustment of locomotive engine features according to the changing locomotive operating conditions. They are: mechanical (created on the basis of a multi-speed gearbox with gear meshing); hydraulic and combined.

Usually, the locomotive reversion is also implemented through the adjustable power transmissions, while the direction of its engine rotation remains unchanged.

3. *According to the value of unstrung masses* there are three types of transmission mechanisms that can be distinguished:

- a first class transmission mechanism with a support-arm traction motor and a support-axle reducer;
- a second class transmission mechanism with a support-arm traction motor and a support-axle reducer;
- a third class transmission mechanism with a support-arm traction motor and a support-arm reducer.

In the first class transmission mechanisms, the traction motor is supported on one side by means of an elastic element on the bogie frame, and on the other side it lies on the wheelset by means of motor-axle bearings. The pinion is mounted on the traction motor shaft, thus forming a pair of gears with the large gear wheel mounted on the wheelset.

A variety of support-axle suspension is the support-center suspension, where the traction motor lies on a hollow shaft by means of sliding bearings, which in their turn are connected to the wheel by means of elastic elements.

Such is the suspension of locomotives of series 06, 07 and 51 operated in Bulgaria.

With the second class of transmission mechanisms, the traction motor transmits entire mass to the bogie frame, and the reduction gearbox transmits its mass through the axle of wheelset and thus it is the only one that belongs to the unsprung parts.

All electric mainline locomotives "Skoda" in BDZ have a second class of transmission mechanism, as well as series 46.

With the third class of transmission mechanisms, the traction motor, together with the wheel reducer, transmit their mass to the bogie frame thus joining to the sprung parts. In its turn, the third-class transmission mechanism is divided into two groups:

- First group – the movement is transmitted by the electric motor to the large gear wheel, which is mounted on a hollow shaft that transmits the drive torque to the wheelset using a clutch. This is called "Third class transmission mechanism" with one-sided transmission of torque, but it is where oscillations appear.
- Second group – the movement is transmitted by the electric motor to the large gear wheel, which is mounted on a hollow shaft that transmits the drive torque using a clutch on the both sides of wheelset. This is called "Third class

transmission mechanism" with two-sided transmission of torque. In this case, washers on the hollow shaft are out of phase by 90° , which minimizes the oscillations in the entire system.

Another structure of transmission mechanism with a hollow shaft is the "Jacque main" system where the torque transmission is one-sided (as in Siemens). The large gear wheel of the wheelset reducer is fixedly mounted on a shaped sleeve, rotating with it on the bearings of the reducer. Two pins are placed on the shaped sleeve, located to each other at 180° . Through them, the torque is transmitted to the driven ring, in turn connected to the hollow shaft by means of pins located at 90° to the above-mentioned two pins. On the other side, the shaft is also connected to the drive ring by the same pins, and the ring transmits the torque to the wheel by means of pins with rubber-metal hinges mounted on them.

In mechanisms equipped with an Alstom type clutch, the torque transmission is bilateral.

The driven pins are connected to the wheel by 4 bolts and to relieve the shear force, the pin is tightly mounted in the hole of wheel.

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First, the spacer ring is mounted on the pin, and then the rubber bushing is pressed into the head of the rein. After the rubber bushing fits snugly, a pressing washer is placed reliably fastened with a bolt and a washer. The rubber parts of bushings are relieved of the compressive forces.

The drive pins are pressed into the hollow shaft disk and are fixed with a special pin. The connection of drive reins with the drive pins and the floating frame is carried out in the same way as with the drive pins. The torque of traction motor through the drive pins is transmitted to the driven pins, through the reins and the floating frame suspended on them. This frame can be shifted with disturbances between the wheelset and hollow shaft.

The connection of the reins with pins and the floating frame is carried out by eight hinges, filled with rubber bushings, reinforced inside and outside with steel bushings. The inner bushing is pressed onto the pin, and the outer one is pressed to the rein lug or floating frame. The disturbances in coupling appear on the account of rubber deformation. However, a hinge of this type can provide simultaneous rotation and tilting in any direction in the area of small deformations. The great advantage of this hinge is that it does not need lubrication, since it has no friction parts, thanks to which it works silently.

The rubber deformation determines the angle of rotation of the reins relative to the spokes, the radial load and sagging of the locomotive sprung parts. With simultaneous displaced centers of the hollow shaft and wheelset up to 30 mm, the angle of rotation reaches 8° .

3. ADVANTAGES AND DISADVANTAGES OF TRANSMISSION MECHANISMS

3.1. First Class Transmission Mechanism

A significant disadvantage of the support-axle suspension is the increased weight of unsprung parts and the dynamic impact of railway track on the traction motor and transmission mechanism.

The strengthening of motor on wheelset limits the application of this type of suspension only to locomotives with average speeds of up to 100 km/h. For this reason, it is used only in locomotives intended for freight service.

3.2. Second Class Transmission Mechanism

The advantage of the support-arm suspension is that the traction motor is protected from vibrations by bogie spring suspension and the mass of unsprung parts on one wheelset is significantly reduced. However, it has a more complex design than the support-axle suspension.

3.3. Third Class Transmission Mechanism

The advantage of the third class transmission mechanism is that the traction motor, together with the wheelset reducer, transmits their mass to the bogie frame and refer to sprung parts of the system. With one-sided transmission of torque, vibrations and torsional moment appear in the mechanism, while with two-sided transmission of torque for one revolution of wheel, the floating frame center makes two revolutions. This is the main disadvantage of the Alstom type coupling.

4. DAMAGES TO THE TRANSMISSION MECHANISMS OF ELECTRIC LOCOMOTIVES IN BULGARIA

The electric locomotives of "Bulgarian State Railways" (BDZ), series 41, 42, 43, 43P are characterized by transmission of torque from the traction electric motor to wheelsets carried out by means of a single-stage wheelset reducer with straight-teeth cylindrical gears. The wheel reducer is suspended axially hinged to the bogie frame and is fixed to the wheelset. The torque from the traction motor to the reducer is transmitted by means of an original cardan joint of

"Hadji" type, manufactured by "Skoda" – Pilsen factories, where the torsion-cardan shaft acts as a damper of torsional vibrations. There is no fundamental difference in the design of wheelset reducers in the each locomotive series.

For the thirty-year period of operation of "Skoda" electric locomotives in Bulgaria, their wheelset gearboxes have shown a number of shortcomings that create difficulties in operation and repairs at locomotive depots. Some of the transmission mechanism elements characterized with unsatisfactory reliability were examined paper [8].

In 1979-1980, scientists from the Todor Kableshkov University of Transport and Technical University of Sofia developed various design options to readjust the connection of the pinion with the shaft. The developed suggestions covered practically all possible cases of cylindrical and conical press joints depending on the degree of using old parts. The main problem with the options was the welding of the splined shaft, made of chrome-nickel steel, which required a special technological mode and electrodes and mandatory heat treatment after that operation. One of the options provided extension of the spline joint durability by means of its bilateral centering in a manner similar to the design solution for the first delivery of electric locomotives of series 43. The most promising suggestion was to readjust the spline joint into a cylindrical-pressed one using shafts identical to those in locomotives of series 43 and $\text{и и } 43^p$, delivered as spare parts or manufactured in Bulgaria [8]. It was proposed to use the old pinions, including those scrapped because of damaged bearing journals or splines.

The transmission mechanism of electric locomotives of series 43/44 and 42/45 have low operational reliability. The damage occurs mainly in the cardan coupling, gears, gearbox lubrication, noise, vibrations, etc.

The damage in wheelset reducers is caused by the exceptional influence of vibrations [9, 10]. They have been studied experimentally, and it has been found that the obtained spectra are close in configuration and can be presented in one generalized case. The amplitudes located in the frequency range of about 4500 Hz belong to the reducer box. Analogous results have been obtained also for the traction gears of electric rolling stock. All main types of damage in gears lead to dynamic load on gears and bearings, and to vibrations in the reducer, respectively, as 45% of the rolling bearings are not repairable.

The structure of wheelset reducer of electric locomotive series 44 and 45 is the same for both series, but differs in the gear ratio, which is achieved by equipping them with different gears at a constant center distance. The largest amount of damage has been found in the parts of gearbox housing, square, bearings of driving wheel (pinion) and driven wheel (gear). These are the main transmission mechanism components that are subjected to both dynamic load and wear in operation.

Damage in gears and gear meshing leads to appearance of oscillations of different nature. These may be kinematic forces caused by deviations in teeth shape in the contact area, shock impacts determining the change of clearance in the gear meshing area, and friction forces between the contact surfaces.

The monitoring on locomotives owned by BDZ – Freight Transportation and BDZ – Passenger Transportation carried out in 2013-2025 showed the most common failures of transmission mechanisms, presented by percentage in Table 1.

Table 1: Damage to the elements of transmission mechanisms of the second and third classes

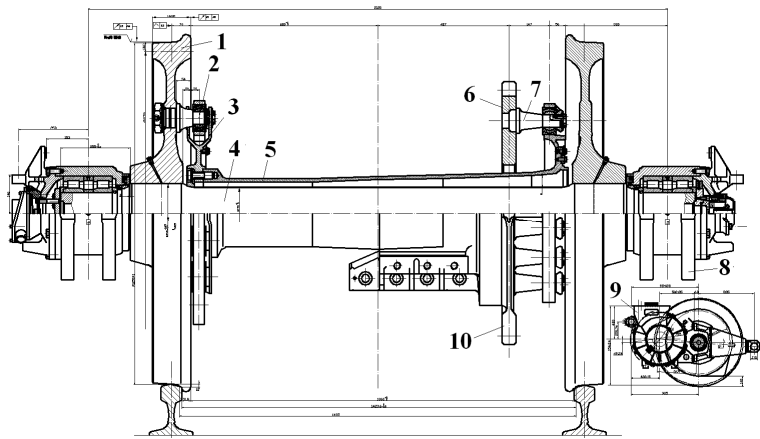
Elements	Electric locomotives with transition mechanisms		
	Of second class		Of third class with Alstom type clutch
	Series 43 and 43 ^p	Series 44, 45, 46	
Reducer housing	25,22%	27,37%	15%
Transmission square	29,98%	28,42%	-
Pinion bearings	16,74%	18,95%	18%
Gear bearings	1,56%	2,11%	5%
Pinion	4,02%	8,42%	8%
Gear	9,38%	8,42%	8%
Pinion flange	6,03%	0%	7%
Pinion shaft	1,34%	1,05%	2%
Traction motor	1,78%	0%	0%
Clutch	-	-	30%
Other	6,70%	4,21%	7%

From the observation done, the highest percentages of failures were found:

- for transmission mechanisms of the second class – the gearbox housing and the square;
- for transmission mechanisms of the third class – damage to the clutch.

5. SELECTION OF A TRANSMISSION MECHANISM TO BE STUDIED

The present study considers a third-class transmission mechanism with a support-arm suspended traction motor and a support-arm suspended reducer (Figure 1) and an Alstom type clutch (Figure 2), which has the most favourable dynamic indicators. One of the main features is that both the engine and the reducer belong to the sprung parts of the locomotive.



1- wheel; 2- “hollow shaft-wheel” connection; 3 – shift fork; 4 - axle; 5- hollow shaft; 6 - large gear wheel; 7- “gear wheel-hollow shaft” connection; 8 - bushing box; 9 - reducer; 10 - reducer box.

Figure 1: Third class transmission mechanism with ALSTOM type clutch.

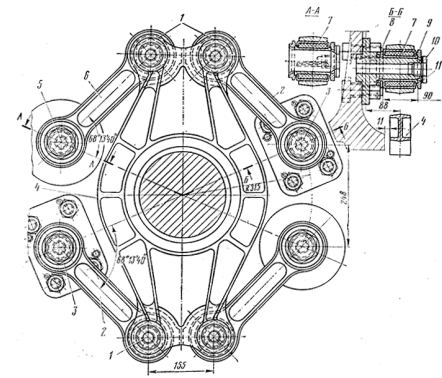


Figure 2: Alstom type clutch [3-7]

1- rubber-metal joints; 2 - guides; 3 - driven pins; 4 - floating washer; 5 –driving pins; 6 - connecting rods; 7 - rubber bushing; 8 - spacer ring; 9 - washer; 10 - bolt; 11 - securing plate.

To calculate the strength and deformation features of the transmission mechanism main elements, it is necessary to indicate the technical specifications of the locomotive type where they will be installed.

Basic technical data :Wheel formula –mm Bo - Bo; Power supply system – 25 kV/50 Hz; Track gauge – 1435 mm; Maximum speed – 175 km/h; Traction power (in the rolling circle of wheels) - 4 MW; Traction force at start-up – 260 kN; Traction force at 73 km/h – 196 kN; Braking force with electrodynamic (regenerative) brake – 118 kN; Total mass (traction mass) – 80 t; Minimum curve entry radius – 80 m; Mass of the locomotive superstructure (body) – 49 t; Mass of the bogies - 31 t; Length between buffers – 19380 mm; Distance between end axles – 12740 mm; Distance between center bolts – 9940 mm; Distance between axles in the bogie – 2800 mm; Wheel diameter – new wheels – 1250 mm; Wheel diameter – worn wheels – 1170 mm; Max. height with pantograph removed (above head rail) – 4590 mm; Maximum width – 3150 mm; Voltage/frequency for train heating – 1.5 kV /50 Hz; Gear ratio of axle gearboxes 22:106; Traction motors with air cooling 4 pcs. of 1020 kW each; Air compressor-screw /oil cooling 3560 l/min at 3000 l/min; Brake mass – brake mode – R 92 t; Brake mass – brake mode – P 63 t.

6. SELECTION OF A TRANSMISSION MECHANISM TO BE STUDIED

The strength and deformation analysis of a transmission mechanism will be done using SolidWorks Professional software. One of the modules applied to develop parts is Cosmos Works (in the new versions it is called SolidWorks Simulation).

The modeling of a third-class transmission mechanism with an Alstom type clutch is shown in Figure 3.

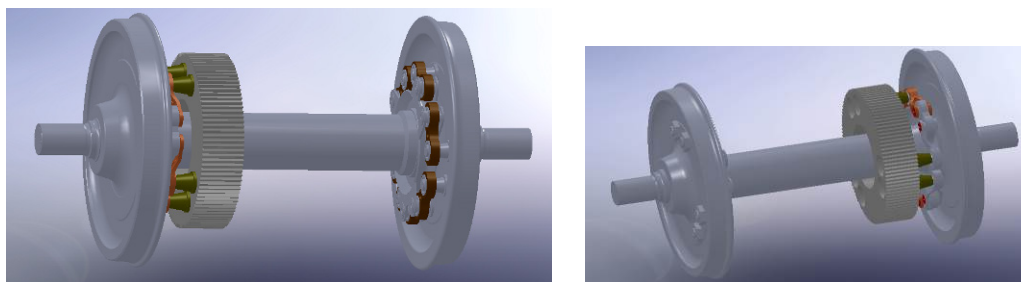


Figure 3: Model of the assembled mechanism

Its main elements, which have been studied in this paper, are the wheelset with a wheel, an axle, a hollow shaft; a hub; a large gearwheel, a shift fork and an eight-like part.

6.1. Strength-Deformation Analysis of a Wheel

The wheel model is shown in Figure 4.

The wheel is loaded by torque in the area of pressing with the axle and is stuck in the holes for attachment to the transmission mechanism. The part is covered with Solid Mesh (Figure 5).



Figure 4: Model of a transmission mechanism wheel

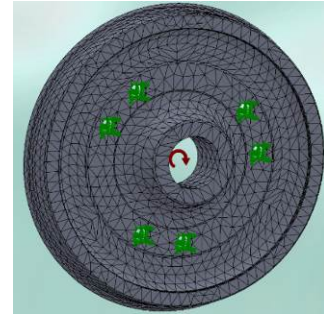


Figure 5: Solid Mesh

The results of study on the wheel are presented in Table 2.

Table 2: Results of the study on wheel

Name	Type	Max	Location
Stress1	VON: von Mises Stress	435.184 MPa Node: 8074	(-765.64 mm, -277.8 mm, -196.9 mm)
Displacement1	URES: Resultant Displacement	0.676519 mm Node: 22315	(-694.07 mm, -434.59 mm, 461.3 mm)
Strain1	ESTRN: Equivalent Strain	0.00146932 Element: 3572	(-749.68 mm, -14.07 mm, 404.49 mm)

Figure 6 shows the distribution of stresses on the wheel. The maximum value of loading is in the area of holes connecting the wheel to the drive mechanism, the maximum load value is 435.184 MPa.

Figure 7 shows the wheel deformation by loading, and Figure 8 presents the distribution of the relative elongation coefficient of the wheel due to loading.

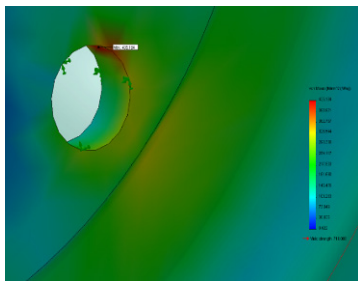


Figure 6: Stress distribution on the wheel

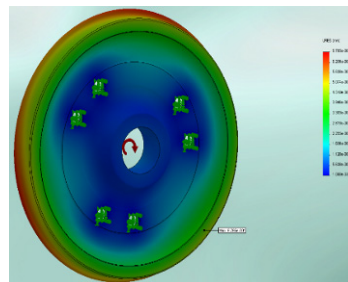


Figure 7: Deformation of the wheel by loading

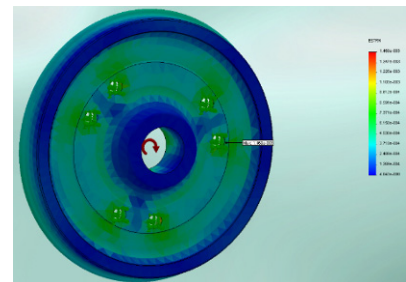


Figure 8: Distribution of the relative elongation coefficient of the wheel

6.2. Strength and Deformation Analysis of an Axle

The axle model is shown in Figure 9. The axle is loaded by torque in the central area and by vertical and horizontal forces in the axle journals, and locking occurs in the parts under hub.

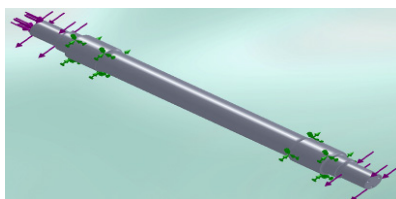


Figure 9: Axle model and load

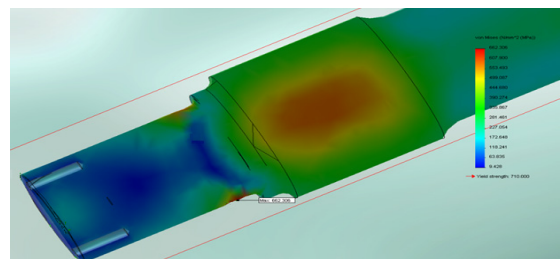


Figure 10: Areas of maximum stress

Figure 10 shows the areas of maximum stress along the axle.

It is observed that the concentration of stresses appears in the transition area between the journal of axle assembly and in the area of wheel pressing-fix, as well as in the area of pressing itself. The maximum load value is 662.306 MPa.

Figure 11 shows the axle deformation by the applied load, and Figure 12 presents the distribution of the relative elongation coefficient.

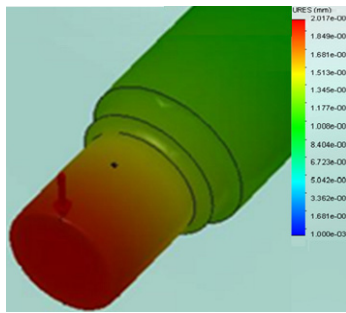


Figure 11: Deformation of the axle by load

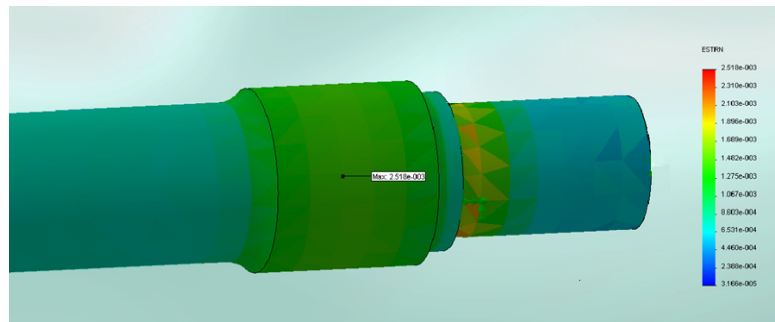


Figure 12: Distribution of the relative elongation coefficient

The results of the axle study are presented in Table 3.

Table 3: Results of the study on axle

Name	Type	Max	Location
Stress1	VON: von Mises Stress	662.306 MPa Node: 902	(-977.943 mm, -80 mm, 0 mm)
Displacement1	URES: Resultant Displacement	0.201699 mm Node: 18269	(767.464 mm, 80.84 mm, -50.07 mm)
Strain1	ESTRN: Equivalent Strain	0.00251756 Element: 9325	(827.235 mm, -0.438461 mm, 7.36 mm)

6.3. Strength and Deformation Analysis of a Hollow Shaft

The model of the hollow shaft is shown in Figure 13.

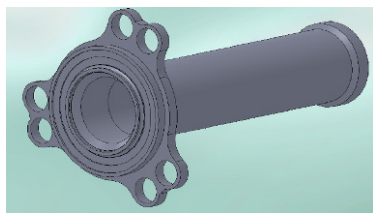


Figure 13: Model of hollow shaft

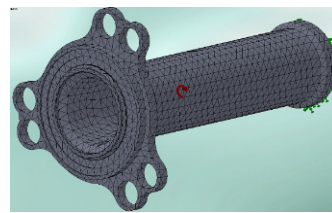


Figure 14: Loading and Solid Mesh

The loading on hollow shaft is carried out by torque concentrated in the hub area, and the locking occurs at the other shaft end. The part is covered with Solid Mesh (Figure 14).

The distribution of stresses along hollow shaft and the concentration of stresses are in the hub area between the lugs for the holes (Figure 15). The maximum load value is 597.821 MPa.

Figure 16 shows the deformation of hollow shaft due to the applied load, and Figure 17 shows the distribution of the relative elongation coefficient.

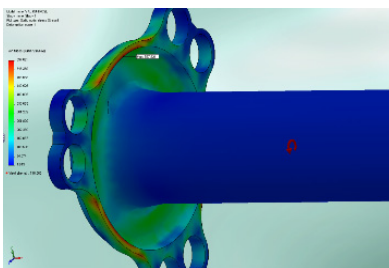


Figure 15: Areas of the maximum stress

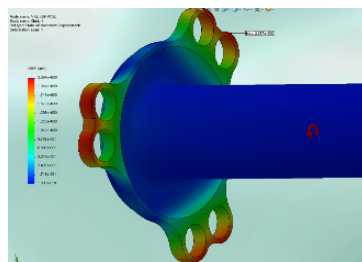


Figure 16: Deformation of the hollow shaft

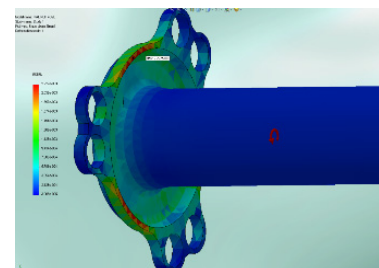


Figure 17: Distribution of the coefficient of relative elongation

The results of study on the hollow shaft are presented in Table 4.

Table 4: Results of study on the hollow shaft

Name	Type	Max	Location
Stress1	VON: von Mises Stress	597.821 MPa Node: 3245	(-1237.3 mm, 259.9 mm, 0.013 mm)
Displacement1	URES: Resultant Displacement	2.09714 mm Node: 19926	(-1239.4 mm, 153.5 mm, -369.2 mm)
Strain1	ESTRN: Equivalent Strain	0.002226 Element: 9885	(-1240 mm, 263.2 mm, 12.5 mm)

6.4. Strength and Deformation Analysis of a Hub

The hub model is shown in Figure 18. The hub is loaded by the torque concentrated in the hub central part, and the locking occurs in lug holes. The part is covered with Solid Mesh (Figure 19).

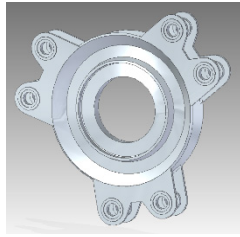


Figure 18: Hub model

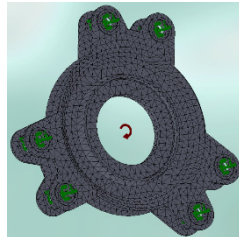


Figure 19: Hub loading and Solid Mesh

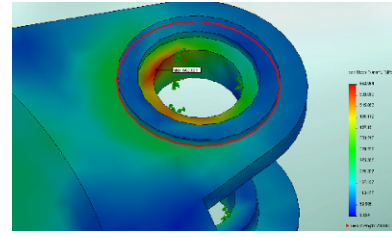


Figure 20: Areas of maximum stress in the hub

Figure 20 shows the distribution of stresses in the hub. The stress concentration is in the area of holes in the lugs. The maximum load value is 652.964 MPa. Figure 21 shows the hub deformation by loading, and Figure 22 shows the distribution of relative elongation coefficient.

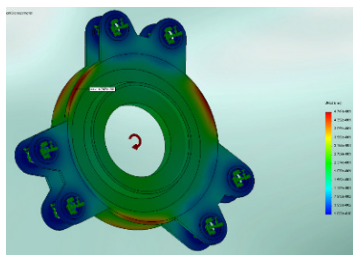


Figure 21: Deformed hub state

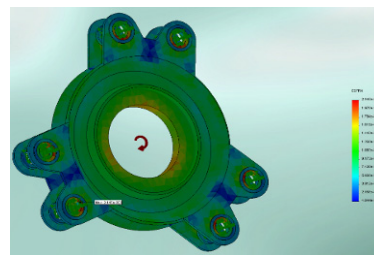


Figure 22: Distribution of the coefficient of relative elongation

The results of the hub study are presented in Table 5.

Table 5: Results of the hub study

Name	Type	Max	Location
Stress1	VON: von Mises Stress	652.964 MPa Node: 2285	(-21.672 mm, 47 mm, -330.51 mm)
Displacement1	URES: Resultant Displacement	0.474714 mm Node: 519	(-176.693 mm, -47.2506 mm, -192.6 mm)
Strain1	ESTRN: Equivalent Strain	0.00214 Element: 8301	(164.355 mm, -44.51 mm, -272.18 mm)

6.5. Strength and Deformation Analysis of a Gear Wheel

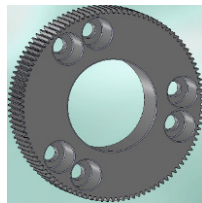


Figure 23: Gear wheel

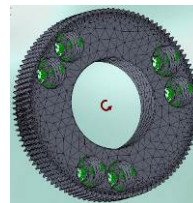


Figure 24: Load on the gear and Solid Mesh

The model of a large gear wheel is shown in Figure 23.

The gear wheel is loaded by the torque concentrated in the wheel central part, and locking occurs in the holes for gripping the mechanism. The part is covered with Solid Mesh (Figure 24).

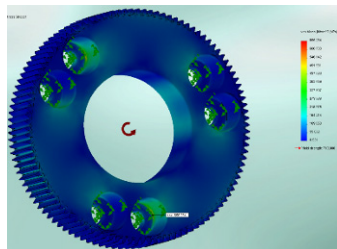


Figure 25: Areas of maximum stresses

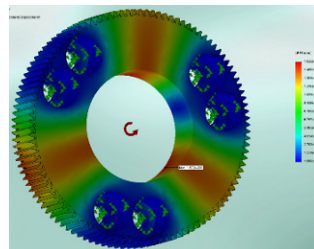


Figure 26: Areas of gear wheel deformation

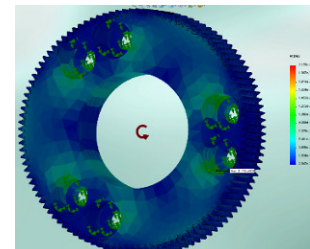


Figure 27: Distribution of the relative elongation coefficient

The distribution of stresses in gear wheel and their concentration are in the transitional part of holes for gripping the wheel (Figure 25). The maximum load value is 655.264 MPa.

Figure 26 shows the gear wheel deformation by the applied load, and Figure 27 shows the distribution of the coefficient of relative elongation.

The results of study on a large gear wheel are presented in Table 6.

Table 6: Results of the study on a large gear wheel

Name	Type	Max	Location
Stress1	VON: von Mises Stress	655.264 MPa Node: 63103	(53.0869 mm, 158.766 mm, 292.226 mm)
Displacement1	URES: Resultant Displacement	0.16216 mm Node: 65881	(-0.0049592 mm, -18.7 mm, 214.01 mm)
Strain1	ESTRN: Equivalent Strain	0.00217841 Element: 32280	(53.5429 mm, -201.8 mm, 227.4 mm)

6.6. Strength and Deformation Analysis of a Shift Fork

The shift fork model is shown in Figure 28. The load on the shift fork occurs with concentrated forces in the part lugs and the clamping occurs in the hole for gripping the gear wheel (Figure 29) and the part is covered with Solid Mesh (Figure 30).

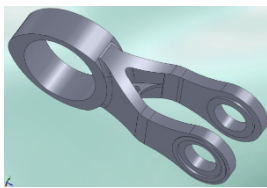


Figure 28: Model of a shift fork

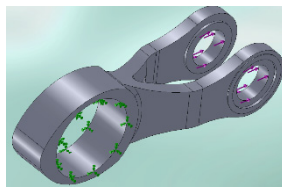


Figure 29: Load on the shift fork

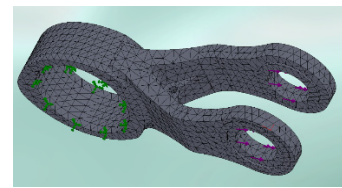


Figure 30: Solid Mesh

Figure 31 shows the distribution of stresses, which are concentrated in the reinforcement area between the two fork lugs. The maximum load value is 126.809 MPa.

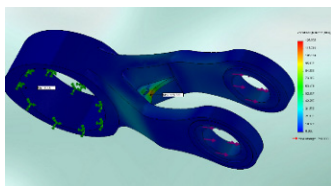


Figure 31: Areas of the maximum stresses

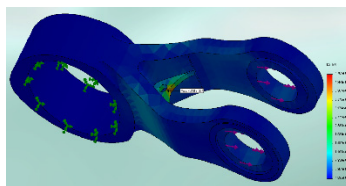


Figure 32: Distribution of the relative elongation coefficient

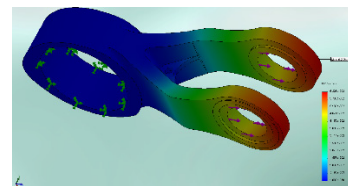


Figure 33: Deformed state of the part

The results of the study on a shift fork are presented in Table 7.

Table 7: Results of the study on a shift fork

Name	Type	Max	Location
Stress1	VON: von Mises Stress	126.8 MPa Node: 1596	(0.00029 mm, 0.728 mm, 98.13 mm)
Displacement1	URES: Resultant Displacement	0.0622768 mm Node: 105	(-0.000413 mm, 46.9 mm, 252.0 mm)
Strain1	ESTRN: Equivalent Strain	0.000476066 Element: 4889	(-2.500 mm, -0.169 mm, 98.73 mm)

6.7. Strength and Deformation Analysis of an Eight-Like Part

The model of an eight-like part is shown in Figure 34.

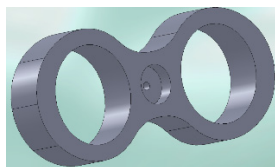


Figure 34: Model of an eight-like part

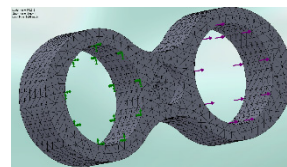


Figure 35: Loading and Solid Mesh

Loading on the eight-like part is caused by concentrated forces in one of the part holes, and the clamping occurs in the other hole (Figure 34). The part is covered with Solid Mesh (Figure 35).

Figure 36 shows the distribution of stresses that are concentrated in the centre between the two holes of eight-like part. The maximum load value is 4.360 MPa.

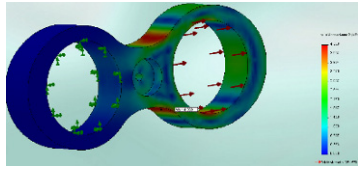


Figure 36: Distribution of the maximum stresses

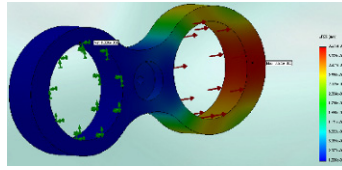


Figure 37: Deformation zone of the eight-like part

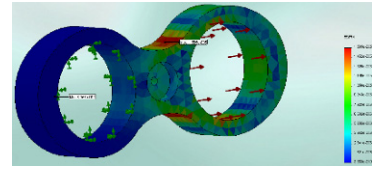


Figure 38: Distribution of the relative elongation coefficient

Figure 37 shows the eight-like part deformation by the applied load.

The distribution of the relative elongation coefficient is shown in Figure 38.

The results of study on the eight-like part are presented in Table 8.

Table 8: Results of the study on the eight-like part

Name	Type	Max	Location
Stress1	VON: von Mises Stress	4.360 MPa Node: 6509	(4.299 mm, -38.31 mm, -111.86 mm)
Displacement1	URES: Resultant Displacement	0.003573 mm Node: 11441	(21.5 mm, -3.5e-006 mm, -226.9 mm)
Strain1	ESTRN: Equivalent Strain	1.5845e-005 Element: 6908	(-6.6527 mm, 36.3 mm, -111.26 mm)

6.8. Assessment of the Strength Condition of Main Transmission Mechanism Elements Studied

Table 9 shows a sample of the maximum stresses obtained according to the von Mises Stress criterion and indicates the areas necessary to be monitored in operation and repair of the main elements of the studied transmission mechanism.

Table 9: Maximum stresses

Element	Max Stress	Areas to be monitored in operation and repair
Wheel	435.184 MPa	In the area of e holes for connecting the wheel to the drive
Axle	662.306 MPa	In the area of transition between the neck and hub part and in the pressed-fix wheel
Hollow shaft	597.821 MPa	In the area between lugs of the hollow shaft hub holes
Hub	652.964 MPa	In the area of holes in the lugs
Gear	655.264 MPa	In the area of holes for gripping the wheel
Shift fork	126.8 MPa	In the area of reinforcement between the two fork lugs
Eight-like part	4.360 MPa	In the centre between the two holes of the eight-like part

Table 9 shows that the most loaded main elements of transmission mechanism studied in this paper are the wheelset, hollow shaft and hub, especially the “hollow shaft-wheel” connection. The large gear wheel should be monitored in operation and repair as well as the teeth profile and parameters and the areas of connection holes.

The shift fork and eight-like part are slightly loaded elements.

In 2013, an electric locomotive with a third-class transmission mechanism and a clutch of ALSTOM type suffered a derailment because of damage caused to “hollow shaft-wheel” connection and the wheelset breakage (photos 1 and 2). That fact confirmed the results that had been obtained during the strength and deformation study on the selected transmission mechanism.



Photo 1



Photo 2

By calculating strength and deformation features of the main elements, using CAE software, it has become clear that the theoretically endangered sections coincide with the actually endangered ones.

7. CONCLUSION AND RECOMMENDATIONS

The presented paper highlights advantages and disadvantages of the types of transmission mechanisms indicated in the structure of electric locomotives operated by the Bulgarian railways companies.

The observations of their operation, carried out in 2013-2025, established the personages of damage caused to elements of transmission mechanisms of the second and the third classes.

This study put special accent to the third-class transmission mechanism with a clutch of Alstom type. The modeling of selected structures performed by Solid Works software determined the strength and deformation features of its main elements. The established areas of maximum stresses should be set under monitoring both in operation and repair.

REFERENCES

- [1] D. Petrovic, V. Aleksandrov, “Zeleznicka vozila. Osnove”, MF Kraljevo, ISBN 978-86-82631-66-8 p.223, (2013)
- [2] Т. Ружеков, Д. Стоянов, Г. Стайков, А. Стефанов, „Конструкция, Теория и Проектиране на локомотиви” С. ВМЕИ “Ленин”, (1987)
- [3] Drive concepts for standard vehicles, <https://railsystem.net/traction-systems/>
- [4] <https://tsa.at/drive-concepts/>
- [5] Drive systems, <https://evolution.skf.com/new-drive-systems-for-mass-transit/>
- [6] Railway Drive System Design: Traction Motor Drive System Basics, <https://theengineeringblog.com/railway-drive-system-design-traction-motor-suspension-tube-online-calculator/>
- [7] Traction drive system, https://sjsutst.polsl.pl/archives/2019/vol105/101_SJSUTST105_2019_Konowrocki.htm
- [8] Ж. Димитров, „Проблеми на ремонта и надеждността на предавателния механизъм при електрическите локомотиви” списание „Железопътен транспорт”, бр. 5, (1981)
- [9] Andreas Steimel, “Electric Traction – Motive Power and Energies Supply”, <http://dnb.ddb.de>, (2008)
- [10] R. Konowrocki, “Modelling of dynamic aspects of operation in railway vehicle traction drive system including the electromechanical coupling”. Scientific Journal of Silesian University of Technology. Series Transport. 2019, 105, 101-111. ISSN: 0209-3324. <https://doi.org/10.20858>, (2019)