

First Results of Experimental Analysis of Stress State and Motion of Tram Gearbox Hinge

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Abstrakt

Cílem práce je zmapovat stav napjatosti a pohyby závěsky nápravové převodovky částečně nízkopodlažní, šestinápravové tramvaje při reálných jízdních podmínkách. K tomuto účelu bylo použito měření pomocí odporových tenzometrů a také systém pro záznam a vyhodnocení pohybu Qualisys. V článku je popsána měřicí aparatura a její samotná instalace na zkoušenou tramvaj. Je zde také uveden postup měření včetně jednotlivých etap simulace reálného provozu, postup zpracování a vyhodnocení naměřených dat a zobrazení získaných výsledků.

Klíčová slova

Závěska tramvajové převodovky, experimentální analýza, tenzometrie, Qualisys

1. Introduction

During working of tram the chassis are dynamically loaded through acceleration, deceleration, passage curves and evidently by the roughness of the rail track. These added dynamic loads can significantly increase wearing of components of the chassis and drive or it can cause damage of some parts. Thus all the components have to be designed with respect to given conditions. In this case, problems with bearing mounting into gear box occurred after relatively little amount of driven kilometres in almost all gearboxes. One of the hypothesis, which can explain this problem, is too rigid suspension of the gearbox. The hinge doesn't allow sufficient movements of the gearbox and generated reaction forces are transferred to the bearings of the output shaft as an additional load. Therefore testing of the gearbox hinge was suggested. The aim of the measurement is to analyse behaviour of tram gearbox and its hinge during the real driving conditions.

1.1 Tram and its drive

Tested tram is partially low-storeyed tram. It has three tractive chassis and it can reach the maximum speed of 60 km/h. Each chassis (Fig. 1) has two tractive mounted wheels, which are driven by partially spring-mounted drive. That consists of asynchronous motor of power 90 kW and two-gear reduction gearbox connected with claw coupling. Big gear of the gearbox is pressed on the axle of the mounted wheels, which is imbedded into bearing units. Their connection with the frame provide four rubber-metal cone springs, that create the primary springing of the tram chassis.

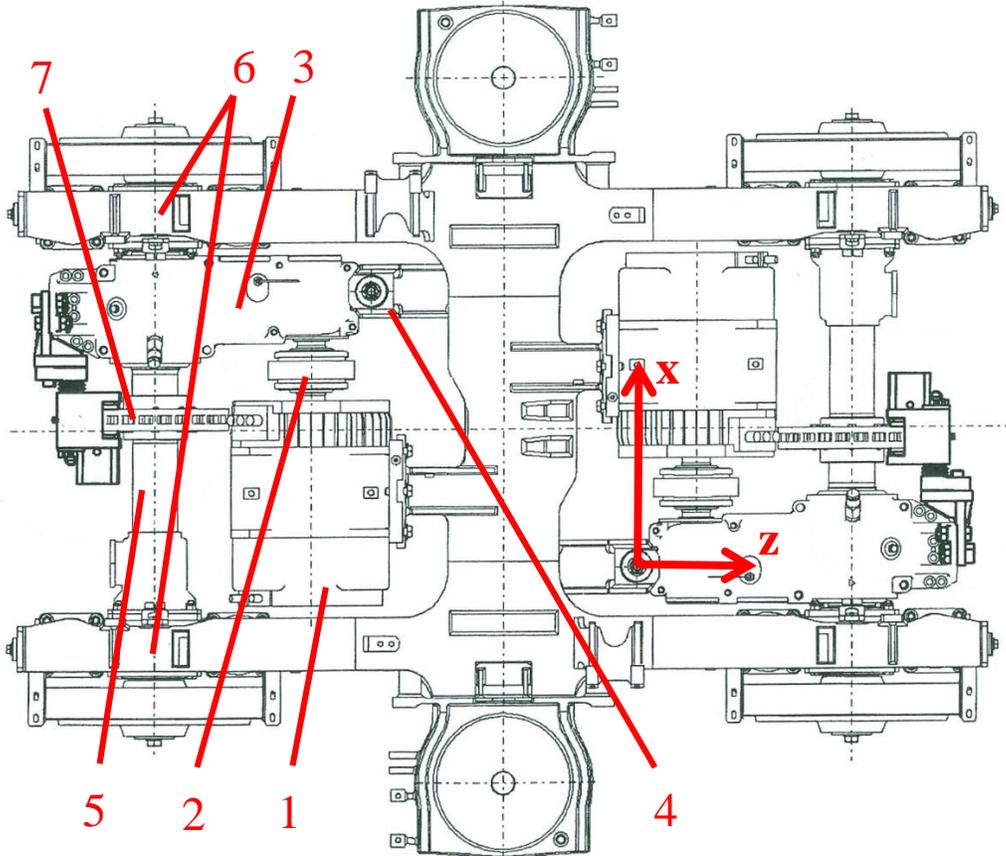


Fig. 1: Schema of the tram drive placed on the chassis – 1-Asynchronous motor, 2-Claw coupling, 3-Gearbox, 4-Gearbox hinge, 5-Mounted wheels, 6-Bearing units, 7-Disc brake

2. Hypothesis

The gearbox is furthermore connected to the chassis frame through the vertical gearbox hinge. The hinge catches reaction force generated by the rotation moment of the mounted wheels, but it has to allow some specific movements of the gearbox. The first one is swinging round the axis x and the second is swinging round the axis z . The maximal tensile force in the hinge for the two-gear gearbox for quasi-static (without respect to the inertial forces of gearbox) can be calculated as

$$S = \frac{M_M(i_c - 1)}{n} \quad (1)$$

where M_M is maximal torque of the asynchronous motor, i_c is gear ratio of the gearbox and n is distance between the output shaft and the hinge axis.

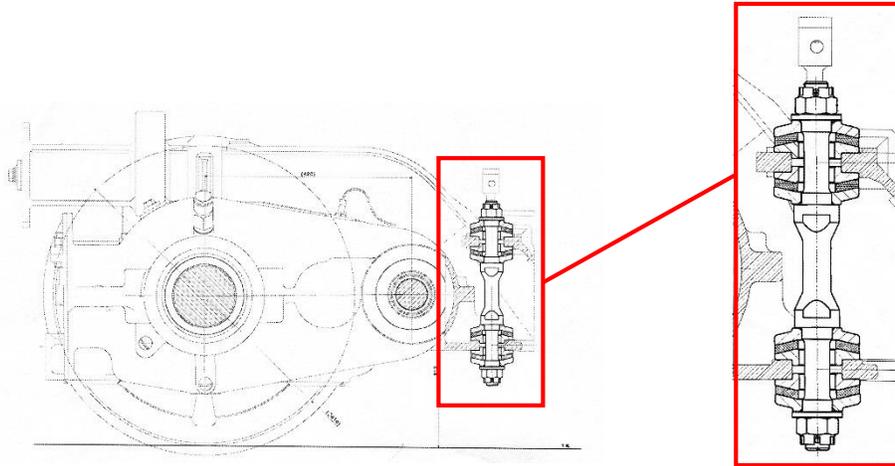


Fig. 2: Gearbox hinge design, [3]

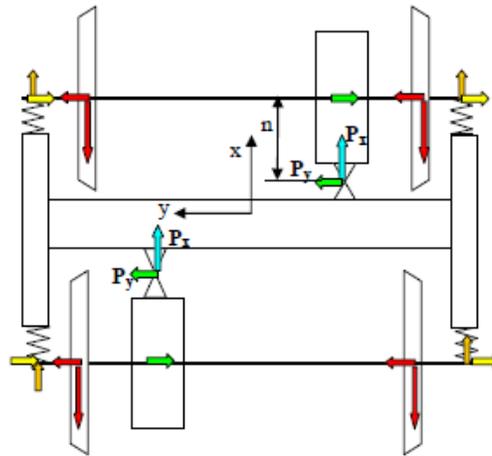


Fig. 3: Parasitic lateral forces, [3]

The hinge of tested tram is designed as a round bar of diameter 28 mm imbedded into rubber supports as it is shown in Fig. 2. These should behave nearly as an ideal spherical joints so the hinge can catch only tension or pressure force. Any bending stress in the hinge is parasitic because it is caused by lateral forces which produce undesirable additional radial and axial loading forces for bearings of the output gearbox shaft (Fig. 3).

3. Instrumentation

As was written above the hinge should transfer only tension or pressure forces. It is desirable to investigate its stress state. Specifically tension or pressure which are expected and bending stress which is undesirable but it can confirm the above mentioned hypothesis. For this purpose strain gauges are used. For the verification and better description of gearbox suspension behaviour the system Qualisys for capturing and evaluating motion is also used.

3.1 Strain gauges

Foil strain gauges are used to investigate stress state of the hinge. It is desirable to measure tension - pressure and bend simultaneously. Moreover the bend has to be find out in two perpendicular planes to obtain complete knowledge of the bend and the resultant including its orientation. The first plane is coincident with the tram drive direction (longitudinal plane) and the second is perpendicular and lies in the lateral plane of the tram. For measuring tension-pressure Wheatstone half-bridge arrangement has to be used due to bend compensation (i.e. two strain gauges). For measuring bend also Wheatstone half-bridge arrangement has to be used due to tension-pressure compensation (i.e. for measuring in two planes four strain gauges). To decrease quantity of strain gauges new arrangement is designed (Fig. 4.). Only four strain gauges are connected into four independent Wheatstone quarter-bridges with time synchronization.

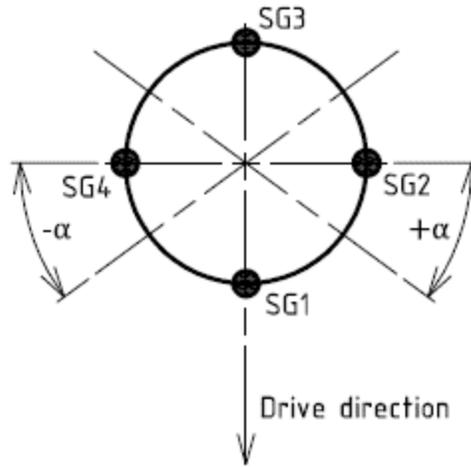


Fig. 4: Measuring arrangement of the strain gauges

This arrangement is very convenient because it is possible to measure tension-pressure like with full-bridge arrangement and simultaneously measure bending in two perpendicular planes like with half-bridge arrangements. Thanks to this improvement only four strain gauges has to be used, which spares not only the number of used strain gauges but also space and makes the connection of strain gauges easier. Acquired signals are obtained as strain from each strain gauge and they can be assembled after measurement like in corresponding Wheatstone bridges according to following formulas. Tension-pressure stress can be calculated from Wheatstone full-bridge formula.

$$\sigma_t = \frac{E}{4}(\varepsilon_1 + \varepsilon_2 + \varepsilon_3 + \varepsilon_4) \quad (2)$$

Bending stresses in two planes are a consequence of formula for half-bridge for bend.

$$\sigma_{b1} = \frac{E}{2}(\varepsilon_1 - \varepsilon_3) \quad (3)$$

$$\sigma_{b2} = \frac{E}{2}(\varepsilon_2 - \varepsilon_4) \quad (4)$$

Now the resultant and its orientation corresponding with Fig. 4 can be obtained.

$$\sigma_b = \sqrt{\sigma_{b1}^2 + \sigma_{b2}^2} \quad (5)$$

$$\alpha = \left[\text{atan} \left(\frac{\sigma_{b1}}{\sigma_{b2}} \right) \right] \cdot \left(\frac{180^\circ}{\pi} \right) \quad (6)$$

Foil strain gauges are installed in the middle of the cylinder surface of the new unused hinge precisely after 90° as it is shown in Fig. 5. For this application self-compensative strain gauges with nominal resistance 120Ω are used. Moreover there is installed semiconductor thermal sensor for monitoring changes of the temperature during test drives. As the hinge is located in the chassis without any cover it is necessary to protect installed strain gauges against outside actions, e.g. water, dust or oil. It has to be protected also during mounting to the chassis. For this purpose short part of plastic tube is used (See Fig. 6) together with flexible putty which creates big enough chamber for strain gauges between the hinge and the plastic tube and also wires can pass through it. Thus adapted hinge is prepared for mounting into chassis of tested tram.

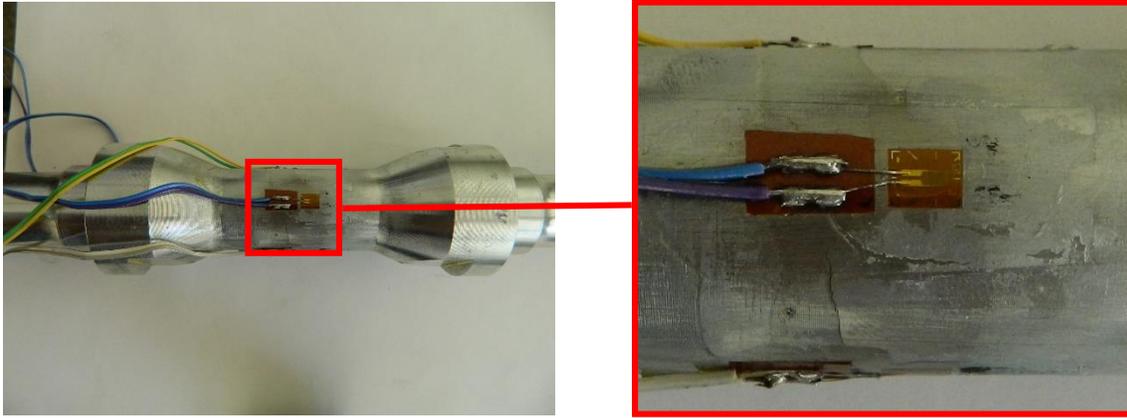


Fig. 5: Strain gauges installed on the hinge

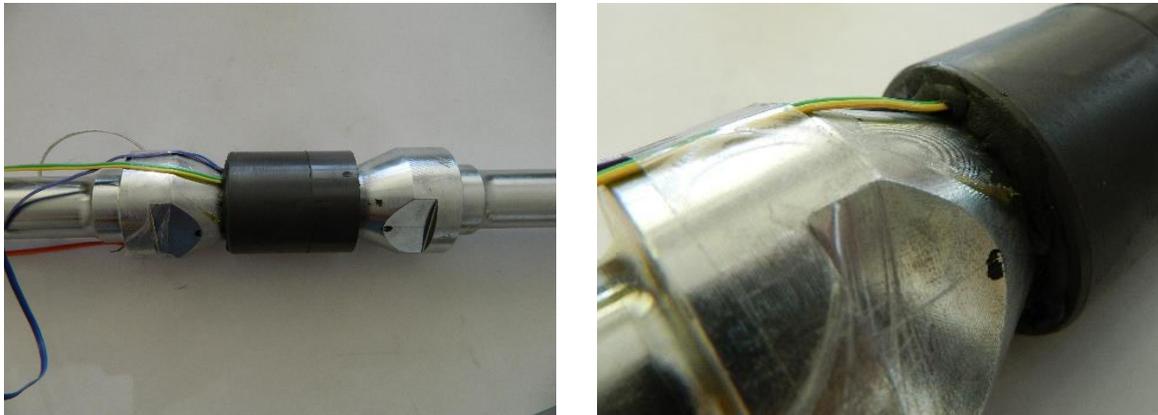


Fig. 6: Protective cover of the hinge with the strain gauges

3.2 Qualisys system

In addition to measuring stress state of the hinge it is convenient to monitor relative motions of the gearbox connected with the mounted wheels to the frame of the chassis and to the asynchronous motor. Then it is possible to evaluate relative shifts and rotations of the output shaft of the asynchronous electromotor to the input shaft of the gearbox and required range of motions of claw coupling. For this purpose optical system for continual capturing and evaluating motions Qualisys is used. For tracking motions of the object it is necessary to use three cameras and three markers creating the reference plane on monitoring object. In this case six spherical markers are used, three are placed on the gearbox and remaining three are placed on the asynchronous motor (see Fig 7). In each object (gearbox and asynchronous motor) is thus created local Cartesian coordinate system (see Fig. 8). Furthermore another marker is placed on the axle of the mounted wheels to sample revolutions so the covered distance and current speed can be calculated.



Fig. 7: Markers of the system Qualisys placed the gearbox and the asynchronous motor

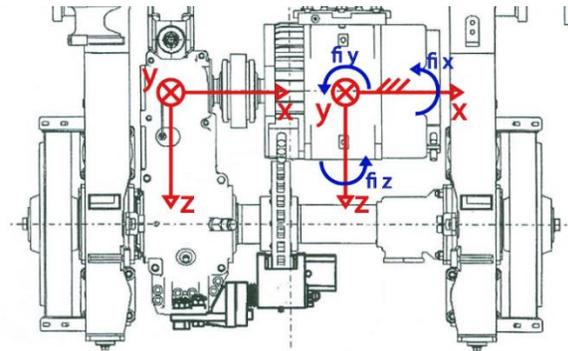


Fig. 8: Cartesian coordinate systems of the gearbox and the asynchronous motor

Altogether six cameras are used, they are divided into two subsystems – moving subsystem and stationary one. The first one is mounted on the special holder which is bolted to the tram frame so it moves with the tram. The second one is placed in the floor pit (See Fig. 9 and 10). The stationary system of cameras is there to verify accuracy of the moving system which could be influenced by the vibration of the tram frame. Both systems are calibrated together and time synchronized.



Fig. 9: Stationary and moving system of cameras



Fig. 10: Moving system of cameras mounted to the holder

4. Data acquisition

Measurement was carried out in the Motol Depot in Prague. The hinge with installed strain gauges and thermal sensor was mounted into rear mounted wheel of the rear chassis of tram tested tram (Fig. 11) and the wires were connected to the measuring central National Instruments placed inside the tram (Fig. 12). Sampling frequency was set for 2000 Hz. Both Qualisys cameras systems were installed as was described above (Fig. 9 and 10) and were connected to the Qualisys central also placed in the tram. Sampling frequency of Qualisys system was set for 500 Hz. Furthermore the torque of two asynchronous motors mounted to the chassis was recorded from the control unit of the tram to get the entire set of interesting values. Sampling frequency of the control unit is variable. All measuring centrals was manually time synchronized.



Fig. 11: Mounted and connected gearbox hinge

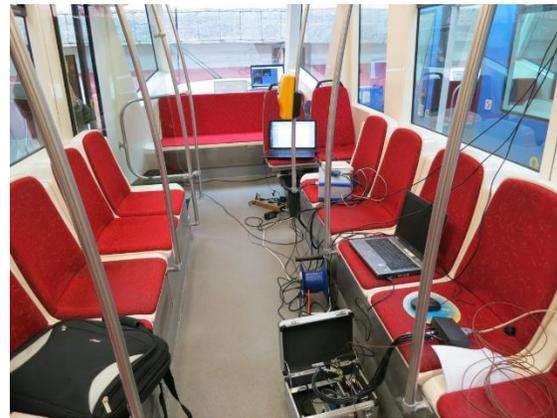


Fig. 12: Measuring central inside the tram

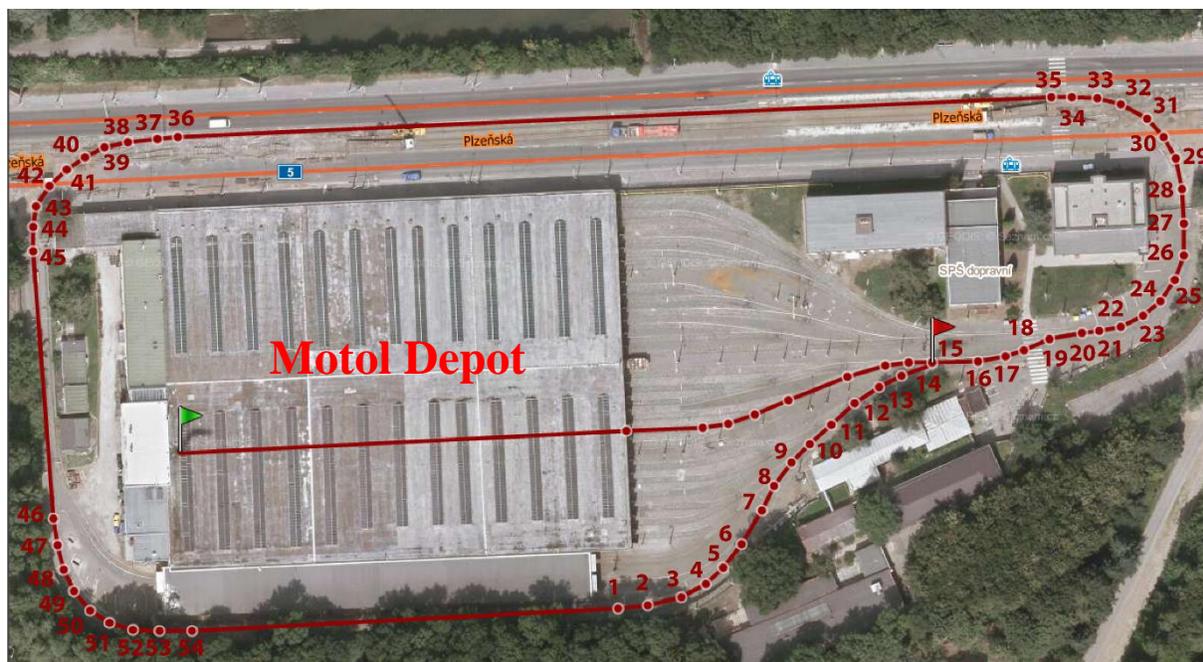


Fig. 13: Testing loop for common drive measurement

Test drive was divided into three regimes – maximal acceleration, maximal deceleration and common drive. Testing of maximal acceleration and deceleration was carried out inside the depot above the floor pit so the accuracy of moving cameras subsystem could be verified by the stationary subsystem. It has to be noticed that maximal deceleration was reached only by electrodynamic braking. Common drive was carried out on the loop inside and around the Motol Depot in total length of 721 m (Fig. 13). It is divided into smaller sections just for easier evaluation. These sections are described in Table 1. Measurement of the maximal acceleration and deceleration was accomplished three times and measurement of common drive twice. During all measurement was the tram empty except the driver and necessary measurement operators.

Table 1: Sections of common drive testing loop

Section	Type	Points	Time [s]	Distance [m]
1	Left curve	2 – 6	7 – 17	7 – 35
2	Left curve	46 – 54	273 – 309	564 – 616
3	Left curve / crossing	7 – 15	19 – 35	45 – 102
4	Left curve / crossing	21 – 27	45 – 58	144 – 180
5	Left curve / crossing	29 – 35	96 – 160	197 – 233
6	Left curve / crossing	37 – 45	192 – 198	452 – 498
7	Crossing	6 – 8	16 – 20	35 – 52
8	Ess curve	37 – 46	37 – 46	113 – 146
9	Straight segment	35 – 36	159 – 192	233 – 447
10	Straight segment	45 – 46	197 – 274	498 – 564
11	Straight segment	54 – 1	309 – 339	616 – 721

5. Data processing and evaluation

Lot of data was acquired from three measuring systems – strain gauges, system Qualisys and control unit of the tram. These data has to be time synchronized because only manual time synchronization was carried out. For synchronizing and evaluating data script written in the software Matlab is used.

Data from strain gauges were acquired with sampling frequency 2000 Hz and contained lot of noise. That's why lowpass linear filter with Hamming's window was applied to reduce it. Script calculates tension, bend in two perpendicular planes and resultant with its orientation according to the formulas (2) to (6) from data acquired from four independent Wheatstone quarter-bridges. Data captured by the Qualisys system were processed in user interface of Qualisys software. Cartesian coordinate system of asynchronous motor was placed to the centre of length of the asynchronous motor and axis x is coincident with to output shaft axis. As the gearbox is the object of interest, this coordinate system was selected as rigid. Cartesian coordinate system of the gearbox was placed to the plane which is coincident with the hinge axis and x axis is coincident with the input shaft axis. Initial mutual rotation of both coordinate systems is zero. The output values acquired from this software are relative shifts (U_x, U_y, U_z) and rotations (ϕ_x, ϕ_y, ϕ_z) of the gearbox to asynchronous motor (See Fig. 8). Next it was necessary to carry out the time synchronization of all acquired data. As the measurement was manually synchronized relevant data had to be shift just for a little time period. Processed data are further displayed onto plots by means of script written in program Matlab.

5.1 Acceleration and deceleration results

The main aim of this experiment is to detect presence of bend in the hinge during test drive regimes. Tensile stress can be used as a verification with analytical tensile stress in the hinge. Tensile force in the hinge is calculated according to the formula (1). For the diameter of the hinge 28 mm tensile stress can be calculated.

$$\sigma_t = \frac{S}{A} = \frac{4 \cdot M_M(i_c - 1)}{\pi \cdot d^2 \cdot n} = \frac{4 \cdot 4,38e5 \cdot (7,2 - 1)}{\pi \cdot 28^2 \cdot 580} = 7,6MPa \quad (7)$$

In the following plots there is shown comparison between acceleration and deceleration.

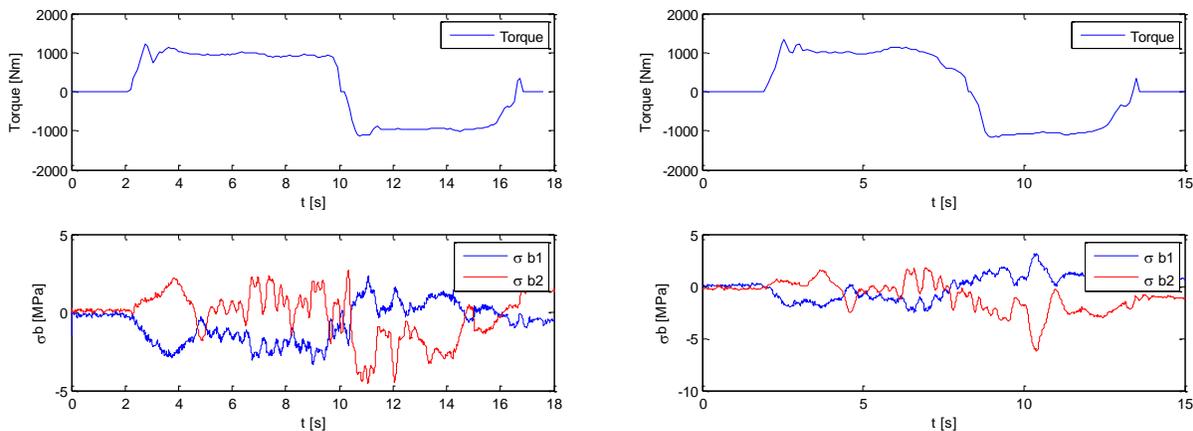


Fig. 14: Torque of two asynchronous motors and bending stress in two perpendicular planes for acceleration (left) and deceleration (right)

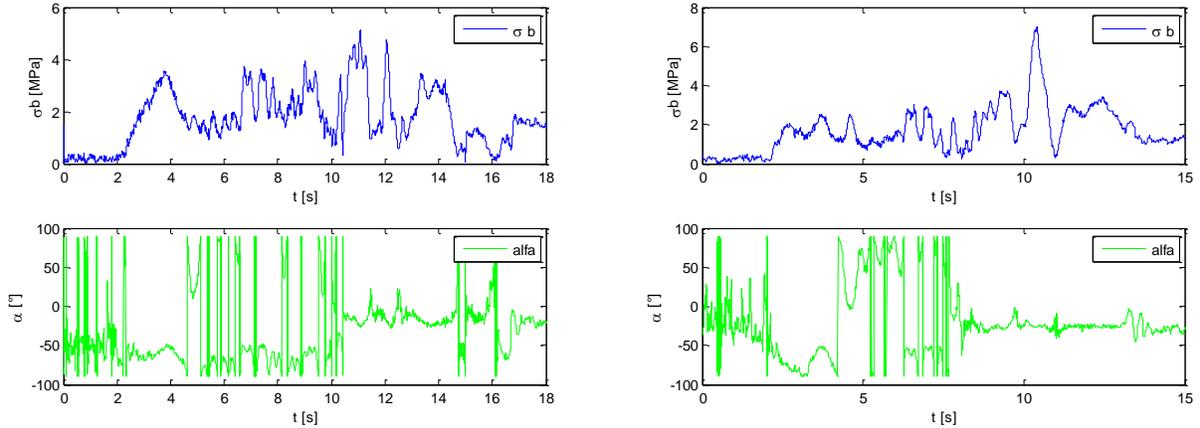


Fig. 15: Resultant of bending stress and its orientation for acceleration (left) and deceleration (right)

In above shown plots (Fig. 14 and 15) there is obvious that behaviour of monitored values is quite similar both for acceleration and for deceleration. During the acceleration regime the tram starts with the maximal acceleration to the maximum speed and decelerates gradually. During the deceleration regime the tram reach gradually the maximum speed and then the maximal electrodynamic braking is used. That's why the first half of the plots in case of acceleration regime and the second part in case of deceleration is more considerable. It is obvious that some amount of bending stress arises in the hinge during both acceleration and deceleration.

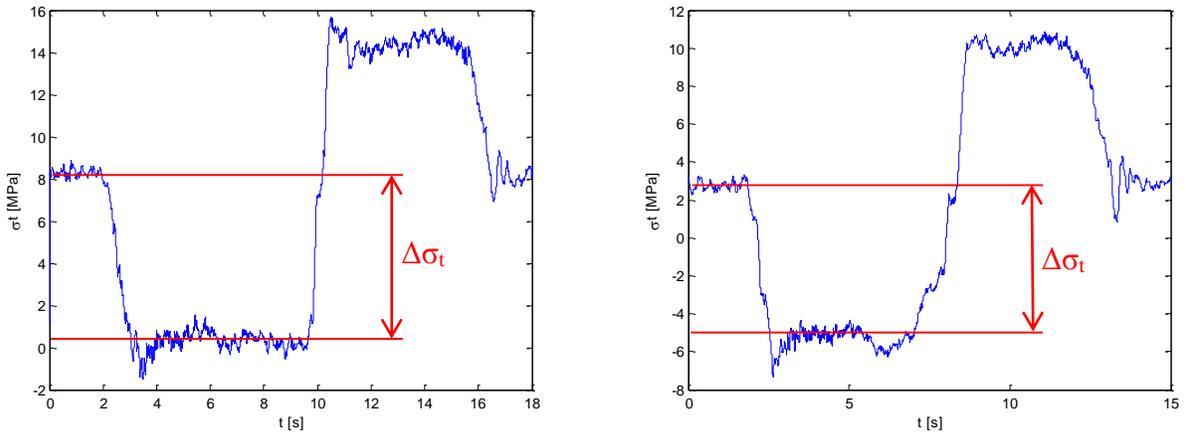


Fig. 16: Tensile stress in the hinge for acceleration (left) and deceleration (right)

In Fig. 16 there is behaviour of tensile stresses in the hinge. It can be seen that the tensile stress doesn't start at the zero stress. This indicates that some stress remains in the hinge even in unloaded state. These values aren't offset because they are changing during test drives. So it is necessary to evaluate only relative deviations of tensile stress $\Delta\sigma_t$, which in average reaches the value 7,8 MPa.

Due to the limited length of this paper it is necessary to reduce the number of plots of measured motions. So only some plots from measurement are shown for demonstration.

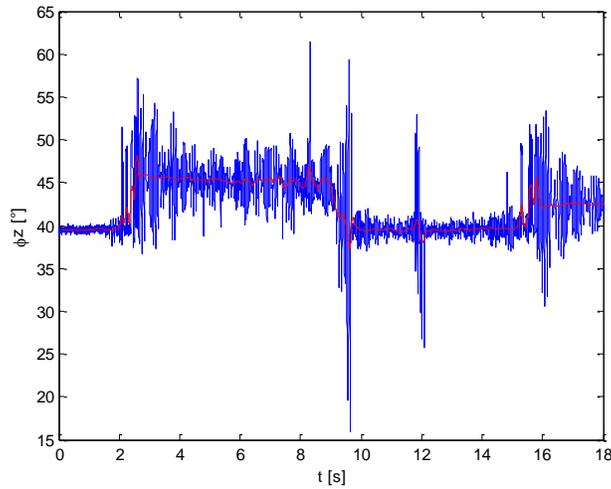


Fig. 17: Relative rotation of the gearbox to the asynchronous motor round the axis z for acceleration regime

In Fig. 17 there is shown rotation of the gearbox round the axis z during the acceleration regime. For better transparency red line is added. It is a representation of mean value during the time. The average deviation of the angle ϕ_z during the acceleration (time about 2 s – 9 s) is 5° .

5.2 Common drive

During the simulation of common drive altogether eleven different sections were passed through (Table 1). One representative section is chosen for the demonstration of the results. It is section 8 when the tram passed through the ess curve.

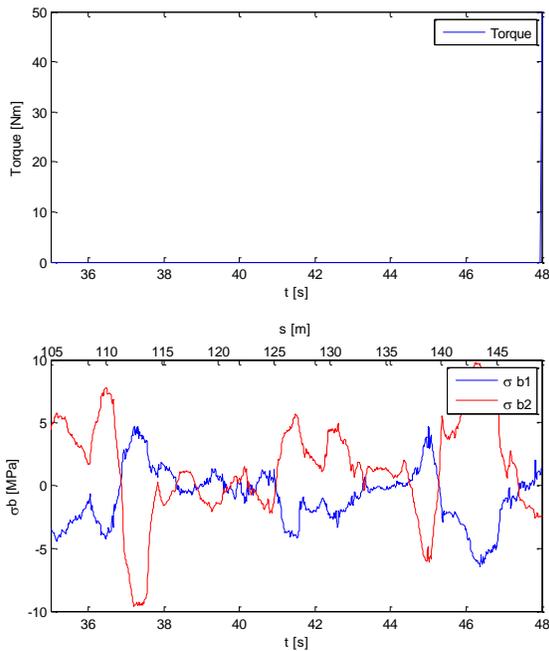


Fig. 18: Torque of two asynchronous motors and bending stress in two planes

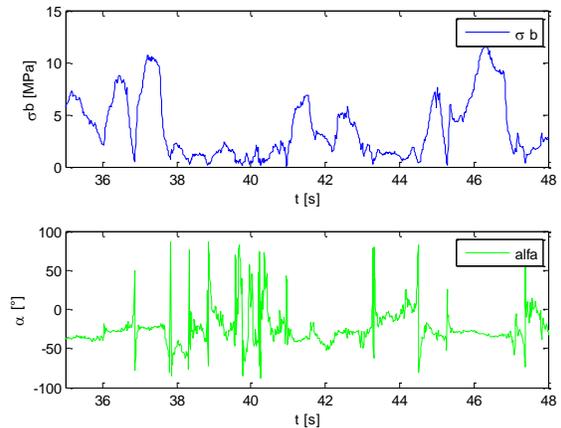


Fig. 19: Resultant of bending stress and its orientation

This section were passed through only by drifting (without bringing in the torque). In spite of this fact even in this regime bending stress occurs in the hinge (Fig. 18 and 19). In the Fig. 20 there is shown the instantaneous velocity of the tram calculated from the revolutions of the axle. These revolutions were determined from the motion of the marker applied on the axle.

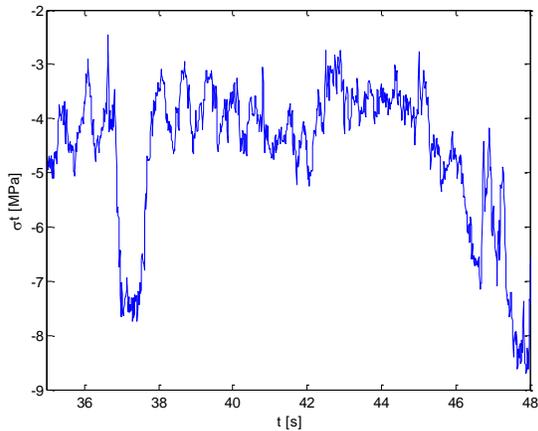


Fig. 20: Tensile stress in the hinge

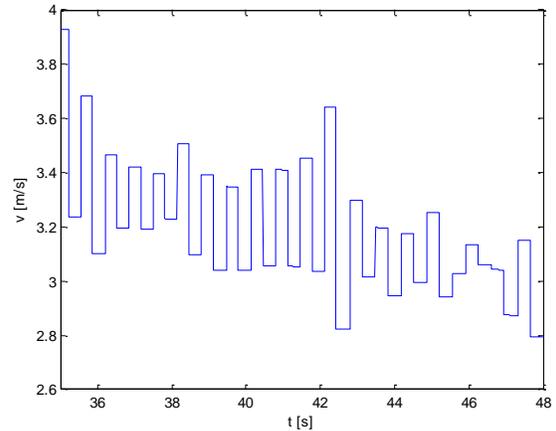


Fig. 21: Instantaneous velocity of the tram

In the Fig. 22 there is once again the demonstration of data acquired from the system Qualisys. In this case the average deviation of the angle ϕ_z during the passage through the ess curve is $6,5^\circ$.

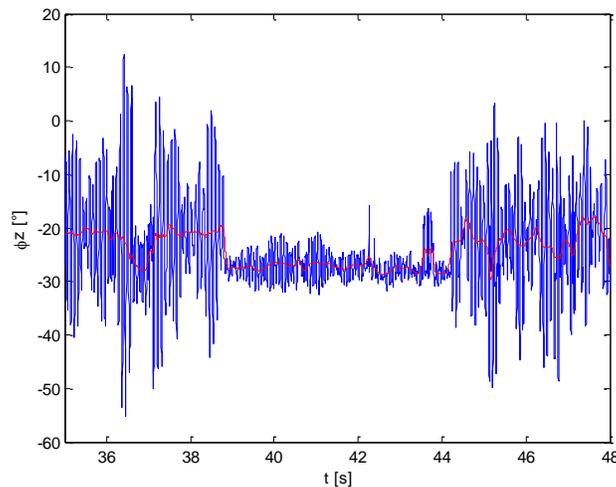


Fig. 22: Relative rotation of the gearbox to the asynchronous motor round the axis z for common drive regime, section with ess curve

6. Results and conclusion

The measurement proved the hypothesis that the gearbox hinge (including its bedding) doesn't catch only tensile and pressure forces. Vice versa the hinge is also bending stressed and thus limits the gearbox movements in directions of axes x and z . But the measured values of bending stress are quite low. Maximal values of bending stress are approximately 10 MPa. This can be caused for example by low dynamic effect. The tested tram was empty during test drives so it had low weight and the quality of the rail track in the Motol Depot was quite good. Another reason could be that

the gearbox hinge was mounted exactly with specified pretension so the rubber bedding isn't so rigid and the hinge allows bigger movements of the hinge. This hypothesis is proved by data measured by system Qualisys (Fig. 17 and 22). These plots show that the relative rotation ϕ_z of the gearbox to the asynchronous motor are higher than admissible values for the claw coupling. So it can cause damage of the coupling.

More detailed analysis of the measured data continues presently. Especially the analysis of measured motions by system Qualisys is carried out. On the ground of the reached results an appropriate structural and assembling modifications will be suggested. From above mentioned results it is obvious that the accomplished measurement proved the presence of the bending stress in the hinge but for better monitoring of tram drive behaviour will be convenient to carry out another measurement on the real rail track and with full loaded tram and different values of the hinge assembling pretension.

List of symbols

A	Area of hinge cross-section	(mm^2)
α	Angle of bending stress resultant	($^\circ$)
d	Diameter of the hinge	(mm)
$\Delta\sigma_t$	Deviation of tensile stress	(MPa)
E	Young's modulus of elasticity	(MPa)
ϕ_i	Relative rotations of the gearbox	(mm)
ε_i	Strain measured by strain gauges	(I)
i_c	Gear ratio	(I)
M_M	Asynchronous motor torque	(Nm)
S	Tensile force in the hinge	(N)
σ_b	Bending stress	(MPa)
σ_t	Tensile stress	(MPa)
U_i	Relative shifts of the gearbox	(mm)
v	Tram velocity	(ms^{-1})

References

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