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NORMATIVE ANALYTICAL CALCULATION OF STB TECHNOLOGY BOLTED JOINT FOR 661 SERIES RAILWAY VEHICLES

NORMATÍVNY ANALYTICKÝ VÝPOČET SKRUTKOVÉHO SPOJA TECHNOLÓGIE STB PRE ŽELEZNIČNÉ VOZIDLÁ RADU 661

Miroslav BLATNICKÝ, Denis MOLNÁR*), Ján DIŽO, Vadym ISHCHUK

1 INTRODUCTION

With increasing improvements, the design and technical characteristics of the vehicles along with their equipment change and advance as well as the train communication system [1]. The European Rail Traffic Management System (ERTMS) is the fundament of interoperability of railway transport in Slovakia. The purpose of mentioned system is to improve interoperability and performance of modern railways. The technically oriented basic project ETCS – the unified European Train Control System – is practically based on ERTMS. The ETCS aims to ensure the transmission of information from control systems in stations (or in sections between stations) to the driving vehicle [2-4]. Accordingly, the system must indicate the train running conditions to the train driver in a uniform manner. Thus, the system is logically made up of two parts. The first is the trackside part (i.e., located on the line), whose task is to collect data from station signalling equipment, lineside equipment or railroad crossing signal equipment. The result of the process is the train running authorisation accompanying the transformation into the prescribed structure of telegrams transmitted to the mobile part of the ETCS system. The second part, called the on-board equipment, is subsequently placed on the driving vehicle. Its task is to receive telegrams from the trackside part of the system. Supervision of the train running is carried out by

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evaluating the content of the received telegrams (data) along with comparing it with the information defining the real train motion (in other words, it checks compliance with the limit parameters of the train movement). Train protection system, in case of need, warns the train operator to react [5-7].

The subject of the paper is the safe attachment of the accessories of the train protection system MIREL VZ1, which is currently in operation on the lines in Slovakia, the Czech Republic, Hungary, and Poland (attachment of the module with bolts to the electric motor unit, the type drawing of electric motor unit can be seen *in **fig. 1***. ***fig. 2*** shows the STB module (functional gateway) and its location in the vehicle's technological cabinet) [8].

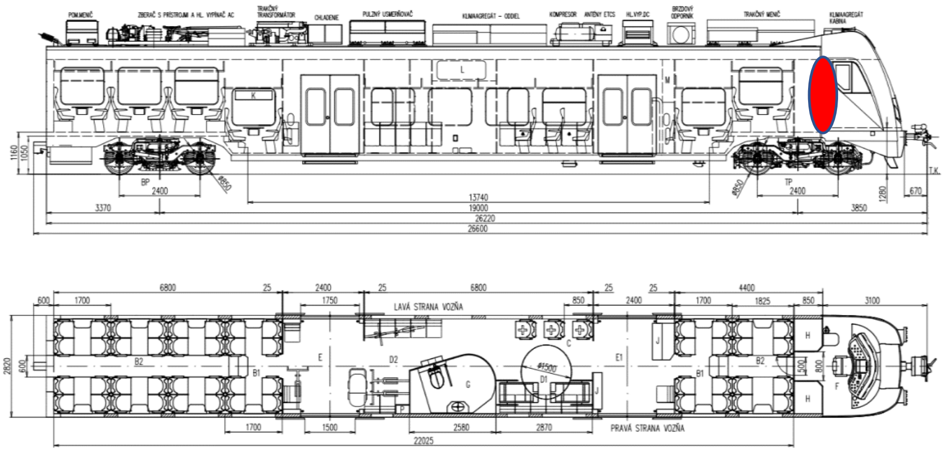
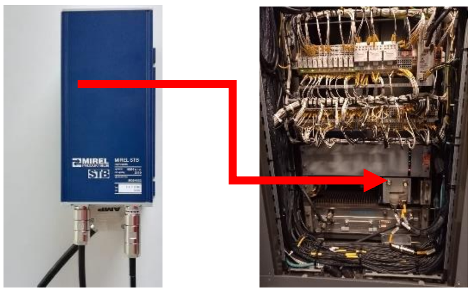


Fig. 1 *Type drawing of electric motor unit EJ class 660/661*

Obr. 1 *Typový výkres elektrickej jednotky EJ radu 660/661*



(a) (b)

Fig. 2 *Photographs of the MIREL STB gateway (a) and its location in the vehicle technology cabinet (b)*

Obr. 2 *Snímky brány MIREL STB (a) a jej umiestnenie v technologickej skrini vozidla (b)*

The extension module of the train protection device in question is the functional gateway MIREL STB. The basic function of mentioned gateway is to provide an interface between the basic unit of the MIREL VZ1 train protection system and the ETCS on-board equipment. These components can be classified as fixed equipment (train equipment). The safety development is closely related to the implementation of widely understood innovations. It is required to attach the fixed equipment, including the equipment inside the passenger compartments, to the structure of the passenger car body in order to prevent these fixed devices from coming loose, inasmuch as their release could pose a risk of injury to passengers or could lead to derailment [8,9]. For this purpose, the fastening of these devices is designed according to EN 12663-1:2010+A1:2014 standard [10]. Thus, the paper concentrates on a validation of the bolted joints of the devices to be fixed to the structure of the railway vehicle body (passenger car body) with regard to the mentioned standard.

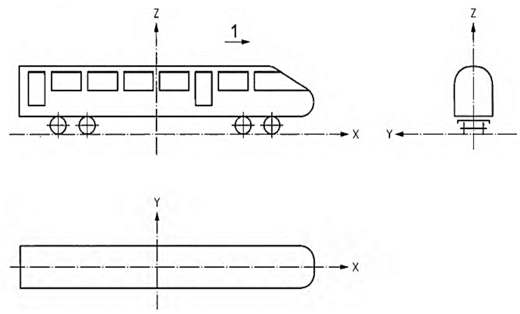


Fig. 3 *Coordinate system of the EJ electrical motor unit 660/661 class (HV5)*

Obr. 3 *Súradnicový systém elektrickej jednotky EJ radu 660/661 (HV5)*

EN 12663 standard specifies the values of accelerations acting on the components to be fixed to the vehicle body according to ***TAB. 1***. ***fig. 3*** demonstrates the considered vehicle coordinate system. The positive direction of the x-axis (coincident with the longitudinal axis of the vehicle body) is in the direction of travel. The positive direction of the z-axis (coincident with the vertical axis of the vehicle body) points upwards. The y-axis (coincident with the transverse axis of the vehicle body) lies in the horizontal plane and completes the right-handed coordinate system.

TABLE 1 *Considered accelerations resulting from EN 12663 standard [10]*

TAB.1 *Uvažované zrýchlenia vyplývajúce z normy EN 12663*

Acceleration in the axis direction	Value of the acceleration
x	± 3g
y	± 1g
z	(1 ± c*)·g
*c = 2 at the end of the vehicle, decreasing linearly to 0.5 in the middle of the vehicle	

In terms of a higher degree of safety (in comparison with all the above-mentioned normative requirements), all strength calculations of individual components will be carried out under the action of forces in all three directions (x, y, z – ***fig. 3***) and superimposed at the greatest possible load on the joints, i.e., combinations at different (most load-inducing) directions (+/-) of the applied forces.

2 INITIAL DATA FOR THE CALCULATION

Components (in the direction from top to bottom) JRU + FJ (FJ = EVC + NVC + Connection Box + FUN) + 2 terminals + VZ1 + STB (on the right side) + BTM ELBE5K, i.e., a total of 10 components, are installed in the technology cabinet on HV5 ***(fig. 4a***). Distribution of the vehicle technology cabinet in which the STB module with bracket is located along the length of the driving vehicle (**red mark in *fig. 1***).

The paper sets out to check the attachment of the STB component to the STB bracket. Moreover, the connection of STB assembly with bracket to the vehicle technology cabinet frame will be investigated. The input parameters of the calculation are:

– maximum mass of the STB specified by the manufacturer $m_{STB} = 1.7\ kg$,

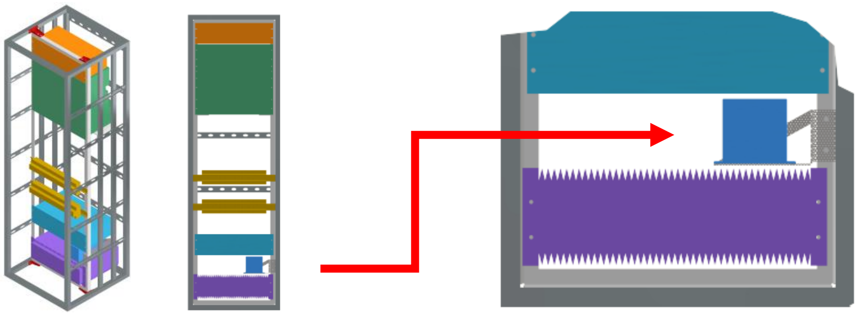
– maximum mass of the STB holder determined by weighing $m_D = 1.6\ kg$,

– considered vehicle length (bogie axle distance) $l_p = 19\ m$ (**fig. 1**),

– position of the mounted STB along the length of the driving vehicle from the outer wheelset ***0 m*** (***fig. 1***),

– strength grade of bolts M8x20 6HRH – ***8.8***, i.e., yield strength of the material $R_e = 640\ MPa$,

– considered value of the gravitational acceleration $g = 9.81\ m \cdot s^{-2}$.



(a) (b)

Fig. 4 *3D CAD model of the vehicle's technology cabinet (a) and detail of the STB attachment to the technology cabinet frame (b)*

Obr. 4 *3D CAD model technologickej skrine vozidla (a) a detail uchytenia STB k rámu technologickej skrine (b)*

Another input parameters are:

– the position of the centre of gravity of the STB is given by the manufacturer approximately in its centre. The considered position of the centre of gravity can be seen in ***figs. 6, 7***,

– the position of the centre of gravity of the bracket is determined by means of calculation and is shown in ***fig. 7***,

– minimum coefficient of friction between the contact surfaces of the components to be joined (steel to steel) $f = 0.1$ ***(-)***,

– the number of bolts used for the connection of the STB to the STB bracket $i_1 = 4$ ***(-)*** M6 (***fig. 5***), each with a core diameter $d_{j1} = 4.733$ ***mm*** within a tolerance of 6g,

– the number of bolts utilised to connect the STB bracket to the technological cabinet frame $i_2 = 2$ ***(-)*** M8 (***fig. 7***), each with a core diameter $d_{j2} = 6.466$ ***mm*** within a tolerance of 6g,

– acceleration in the x-axis direction with respect to EN 12663-1:2010+A1:2014 according to the specification stated in Appendix J-1 under index number 12, taking into account category P-II for passenger rolling stock $x = \pm 3g$,

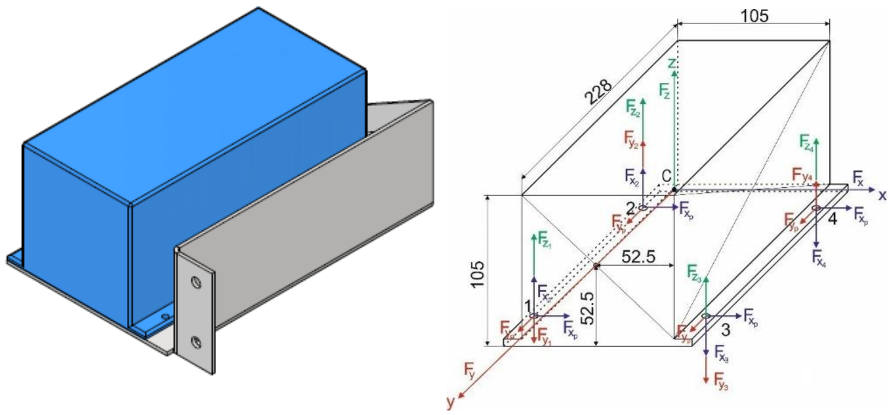
– acceleration in the y-axis direction with respect to EN 12663-1:2010+A1:2014 according to the specification stated in Appendix J-1 under index number 12, considering category P-II for passenger rolling stock $y = \pm 1g$,

– acceleration in the z-axis direction with respect to EN 12663-1:2010+A1:2014 according to the specification stated in Appendix J-1 under index number 12, taking into account category P-II for passenger rolling stock $z = (1 \pm c).g$,

– the coefficient *c* for the exact specification of the acceleration in the vertical direction (z-axis) was determined with respect to the distribution of the technological cabinet in which the STB along with the bracket are located along the length of the train with the value $c = 2$ ***(-)***, it being assumed that $c = 2$ ***(-)*** at the end of the vehicle and the value of this coefficient decreasing linearly up to the value $c = 0.5$ ***(-)*** as long as the component to be fixed is located in the centre of the vehicle.

– in terms of strength, the safety factor of the dynamically operating bolted joint is $n = 0.234$ ***(-)***,

– design stress is $\sigma_{design} = 150$ **MPa**.



(a) (b)

Fig. *5 3D CAD model of the STB module mounted on the STB bracket (bracket is made of 3 mm thick steel plate) (a) and the considered position of the centre of gravity C "STB" along with the geometry and with the marked force effects on the individual bolts 1 to 4 from the load induced by the accelerating effects (b)*

Obr. 5 *3D CAD model STB uloženého na držiaku STB (držiak vyrobený z oceľového plechu hrúbky 3 mm) (a) a uvažovaná poloha ťažiska Ť „STB“ spolu s geometriou a s vyznačenými silovými účinkami na jednotlivé skrutky 1 až 4 od zaťaženia vyvolaného zrýchľujúcimi účinkami (b)*

Calculation of the loading forces (from the STB), which emerges from fulfilling the requirement according to EN 12663-1:2010+A1:2014, i.e., ***x = ± 3.g, y = ± 1.g, z = (1 ± c).g***, where the coefficient *c* due to the distribution of the component fit along the length of the vehicle is considered with the value ***c = 2 (-)*:**

$$F_x = m_{STB} \cdot (\pm 3 \cdot g) = 1.7 \cdot (\pm 3 \cdot 9.81) \cong \pm 50 N, \tag{1}$$

$$F_y = m_{STB} \cdot (\pm 1 \cdot g) = 1.7 \cdot (\pm 1 \cdot 9.81) \cong \pm 17 N, \tag{2}$$

$$F_z = m_{STB} \cdot (1 + c) \cdot g = 1.7 \cdot (1 \pm 2) \cdot 9.81 \cong \pm 50 N \rightarrow \text{maximal.} \tag{3}$$

The determined forces can be seen in ***fig. 5b*** (F_x – purple, F_y – red, F_z – green). Calculation of the loading forces (from the bracket) resulting from fulfilling the requirement according to EN 12663-1:2010+A1:2014 standard, i.e., ***x = ± 3.g, y = ± 1.g, z = (1 ± c).g***, where the coefficient *c* due to the distribution of the component fit along the length of the vehicle is considered with a value of ***c = 2 (-)*:**

$$F_{x_D} = m_D \cdot (\pm 3 \cdot g) = 1.6 \cdot (\pm 3 \cdot 9.81) \cong \pm 47 N, \tag{4}$$

$$F_{y_D} = m_D \cdot (\pm 1 \cdot g) = 1.6 \cdot (\pm 1 \cdot 9.81) \cong \pm 16 N, \tag{5}$$

$$F_{z_D} = m_D \cdot (1 + c) \cdot g = 1.6 \cdot (1 \pm 2) \cdot 9.81 \cong \pm 47 N \rightarrow \text{maximal.} \tag{6}$$

3 VALIDATION OF ATTACHMENT OF THE STB MODULE TO STB BRACKET

The axial force in the bolt depends directly on the tightening torque specified by the manufacturer and the geometry of the employed **M6x20** bolts. These parameters along with the material combination (bolt strength grade **8.8 => R_e = 640 MPa**) and the safety factor (**0.1 ÷ 0.3** for dynamically loaded bolted connections, defines the design stress in the bolt core from the tightening torque, i.e., $\boldsymbol{\sigma_{design}} \approx$ **150 MPa**. The tensile strength condition according to solid-phase mechanics is represented by formula (7):

$$\sigma_{design1} = \frac{F_{o1}}{S_{j1}} \leq \sigma_{dov} \tag{7}$$

From where equation (8) holds for the permissible axial force acting in the bolt:

$$F_{o1} \leq S_{j1} \cdot \sigma_{design1} \tag{8}$$

The core area of the bolt is (9):

$$S_{j1} = \frac{\pi \cdot d_{j1}^{\ 2}}{4} = \frac{\pi \cdot 4.773^2}{4} = 17.6\,mm^2 \tag{9}$$

Thus, the maximum axial force in the bolt within the selected safety is (10):

$$F_{o1} = \sigma_{design} \cdot S_{j1} = 150 \cdot 17.6 = 2640\,N \tag{10}$$

The total normal force of the utilised bolts is (11):

$$F_{N1} = F_{o1} \cdot i_1 = 2640 \cdot 4 = 10560\,N \tag{11}$$

Accordingly, the maximum force transmitted by friction is determined by means of formula (12):

$$F_{T1} = F_{N1} \cdot f = 10560 \cdot 0.1 = 1056\,N \tag{12}$$

The force F_x (***Fig. 5b***) stresses the bolts to shear (in the x-axis) by its sliding action and a load (13) is applied on each bolt:

$$F_{xp} = \frac{F_x}{i_1} = \frac{50}{4} = 12.5\,N \tag{13}$$

Nevertheless, the force F_x has a rotational effect on the bolts with the torque value (based on ***Fig. 5b***) (14):

$$M_x = F_x \cdot \frac{105}{2} = 50 \cdot 52.5 = 2.625\,N \cdot m \tag{14}$$

The effect of the M_x torque is manifested in the bolts by the generation of axial forces (Fx_1 to Fx_4) as is depicted in ***fig. 5b***. This phenomenon will cause a change in the axial force in the bolt (tension, compression) with an order of magnitude lower value compared to the prestress (normal force of bolt). Due to the geometry, this axial force is the same for all four bolts and equally with a value of 12.5 N as the sliding effect of the bolts. Because of these rotational effects cause barely minimal changes in the axial forces of the bolts as well as do not put shear stress on the bolts, they can be neglected in the following calculation. This condition is applied to tightened bolts as well as on loose bolts. In terms of loose bolts, the resultant axial force will be significantly lower in contrast to the original prestress from tightening the bolt. The force F_y (***fig. 5b***) will stress the bolts through its sliding effect on shear (in y-axis), with a load (15) on each bolt:

$$F_{yp} = \frac{F_y}{i_1} = \frac{17}{4} \cong 5\,N \tag{15}$$

Nonetheless, the force F_y has a rotational effect on the bolts with the torque value (based on ***Fig. 5b***) (16):

$$M_y = F_y \cdot \frac{105}{2} = 17 \cdot 52.5 \cong 0.9\,N \cdot m \tag{16}$$

As in the previous case of the rotational effect of the force F_x, the effect of the torque M_y in the bolts is manifested by the creation of axial forces (***fig. 5b***). This phenomenon will cause a change in the axial force in the bolt (tension, compression) with an order of magnitude lower value in contrast to the prestress (normal force of bolt). Inasmuch as these rotational effects cause barely minimal changes in the axial forces of the bolts and do not put shear stress on the bolts, they can be discounted in the following calculation. Thus, this assumption is applied to both tightened bolts and loose bolts. In case of loose bolts, the resultant axial force will be significantly lower compared to the original prestress from tightening the bolt. The force F_z will exclusively stress the bolts in tension, with a load (17) on each bolt:

$$F_{z1\div 4} = \frac{F_z}{i_1} = \frac{50}{4} = 12.5\,N \tag{17}$$

Because of the low value of this effect and for the reason that it causes tensile (not shear) stress on the bolts, it can be neglected in the calculation. The total shear load on the bolts will be represented by the vector sum of the translational effects of the forces F_{xp} a F_{yp} (18):

$$F_{s_{1\max}} = \sqrt{F_{xp}^{\ 2} + F_{yp}^{\ 2}} = \sqrt{12.5^2 + 5^2} \cong \boldsymbol{14\ N < F_{t1}/i_1 = 264\ N} \tag{18}$$

14 N < 264 N, calculated per one bolt.

Therefore, if the bolts are tightened, their utilization is safe, as the above calculations demonstrated. The numbers of bolts and their diameters are designed with sufficient safety against all forces acting on the attachment of the STB module to the bracket. The bolted joints are correctly designed and respect the EN 12663-1:2010+A1:2014 standard. The determined value of the maximum shear force corresponds to the maximum shear stress (19) in the event of cessation of the friction force (loosening of the nuts):

$$\tau_{s_{1\max}} = \frac{F_{s_{1\max}}}{S_{j1}} = \frac{14}{17.6} \cong 1MPa\ , \tag{19}$$

which is a negligible value of the shear stress of the material of the used bolts. Therefore, even in the case of loosening of the bolts, their utilization is safe as demonstrated by the above calculations. The numbers of bolts and their diameters are designed with sufficient safety against the shear forces acting on the attachment of the STB module to the bracket. The bolted joints, even in case of loosening, are properly designed and respect EN 12663-1:2010+A1:2014 standard.

4 CALCULATION OF THE ASSEMBLY "STB MODULE - STB RACKET" TO THE FRAME OF THE TECHNOLOGICAL CABINET

Similarly, in the case of the connection of the STB bracket to the frame of the vehicle's technology cabinet, the axial force in the bolt directly depends on the tightening torque specified by the manufacturer and the geometry of the **M8** bolts employed. This phenomenon along with the material combination (bolt strength grade **8.8 =>** $\boldsymbol{R_e}$ **= 640 MPa**) and the safety factor (**0.1 ÷ 0.3**) for dynamically loaded bolted connections, defines the design stress in the bolt core from the tightening torque, i.e., $\boldsymbol{\sigma_{design}}$ **≈ 150 MPa**. The tensile strength condition according to solid-phase mechanics is given by formula (20):

$$\sigma_{design2} = \frac{F_{o2}}{S_{j2}} \le \sigma_{dov} \tag{20}$$

From where, for the permissible axial force acting in the bolt, the following formula is applied (21):

$$F_{o2} \le S_{j2} \cdot \sigma_{design2} \tag{21}$$

The core area of the bolt is (22):

$$S_{j2} = \frac{\pi \cdot d_{j2}^{\ 2}}{4} = \frac{\pi \cdot 6.466^2}{4} = 32.83\,mm^2 \tag{22}$$

The maximum axial force in the bolt within the selected safety is subsequently given by means of formula (23):

$$F_{o2} = \sigma_{design2} \cdot S_{j2} = 150 \cdot 32.83 \cong 4925 N \tag{23}$$

Thus, the total normal force of the bolts utilised is (24):

$$F_{N2} = F_{o2} \cdot i_2 = 4925 \cdot 2 = 9850 N \tag{24}$$

Further, the maximum force transmitted by friction is (25):

$$F_{T2} = F_{N2} \cdot f = 9850 \cdot 0.1 = 985 N \tag{25}$$

The force F_x acting in the x-axis direction (***Fig. 6*** – green axis) will stress the bolts to shear (in the x-axis) by its sliding effect, with a load (26) on each bolt:

$$F_{xSTB} = \frac{F_x}{i_2} = \frac{50}{2} = 25 N \tag{26}$$

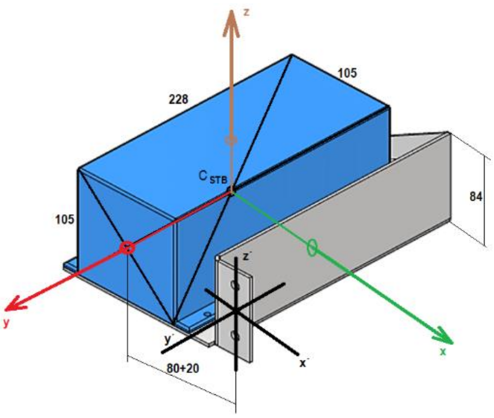


Fig. 6 *Considered position of the centre of gravity C "STB" and the position relative to the coordinate system x´, y´, z´ - axes of symmetry of the bolts 1 and 2*

Obr. 6 *Uvažovaná poloha ťažiska Ť „STB“ a poloha voči súradnému systému x´, y´, z´ - osí symetrie uloženia skrutiek 1 a 2*

The impact of the torque $M_{xz'}$ is manifested in the bolts by generation of axial forces. This phenomenon induces a change in the axial force in the bolt (tension, pressure) with an order of magnitude lower value compared to the prestress (normal force of the bolt). Inasmuch as these rotational effects cause barely minimal changes in the axial forces of the bolts and do not put shear stress on the bolts, they can be neglected in the following calculation. Thus, this assumption is applied to tightened bolts as well as to loose bolts. In case of loose bolts, the resultant axial force will be significantly lower in contrast to the original prestress from tightening the bolt. Nonetheless, the force F_x has a rotational effect (about the y'-axis) on the bolts with a torque value (based on ***fig. 6***) (28) and causes a shear load on the bolts in the x-axis direction (29).

$$M_{xy'} = F_x \cdot \left(\frac{105}{2} - \frac{84}{2} \right) = 50 \cdot 10.5 = 0.525 N \cdot m \tag{28}$$

$$M_{xy'} = 2 \cdot F_{xy'} \cdot 25 \Rightarrow F_{xy'} = \frac{M_{xy'}}{2 \cdot 25} = 10.5 N \tag{29}$$

The F_y force has no sliding effect on the bolts, exclusively tensile/compressive forces are generated, with the load (30) on each bolt:

$$F_{y1} = \frac{F_y}{i_2} = \frac{17}{2} = 8.5 N \tag{30}$$

In addition, this insignificant force effect does not produce shear load on the bolts and will be neglected in the calculation. Nevertheless, the force F_y also has a rotational effect on the bolts (around the z´-axis) with the torque value (with regard to ***Fig. 6***) (31):

$$M_{yz'} = F_y \cdot (80 + 20) = 17 \cdot 100 \cong 1.7 N \cdot m \tag{31}$$

The effect of the $M_{yz'}$ torque is signified in the bolts by the development of axial forces, as shown in ***fig. 6***. This phenomenon produces a change in the axial force in the bolt (tension, compression, approximately 40 N). Thus, with an order of magnitude lower value compared to the prestress (bolt normal force of 4925 N). Since these rotational effects cause insignificant changes in the axial forces of the bolts as well as do not put shear stress on the bolts, they can be discounted in the following calculation. This condition applies to both tightened bolts and loose bolts. From the point of view of loose bolts, the resultant axial force will be significantly lower in comparison with the original prestress from tightening the bolt. Nonetheless, the force F_y has the rotational effect (about the x'-axis) on the bolts with a torque value (according to ***fig. 6***) (32):

$$M_{yx'} = F_y \cdot \left(\frac{105}{2} - \frac{84}{2} \right) = 17 \cdot 10.5 = 0.18 N \cdot m \ , \tag{32}$$

and creates an additional axial load on the bolts (approximately 5 N), which will be neglected in the following calculation. The force F_z acts on the bolts by its sliding effect (shear in the z-axis), with the load on each bolt (33):

$$F_{z1} = \frac{F_z}{i_2} = \frac{50}{2} = 25 N \tag{33}$$

Nevertheless, the force F_z has a rotational effect (about the x´-axis) on the bolts with the torque value (based on ***Fig. 6***) (34):

$$M_{zx'} = F_z \cdot \frac{228}{2} = 50 \cdot 114 = 5.7 N \cdot m \tag{34}$$

The impact of the torque $M_{zx'}$ is signified in the bolts by the development of axial forces (approximately 114 N). This phenomenon induces a change in the axial force in the bolt (tension, pressure) with an order of magnitude lower value compared to the prestress (normal force of the bolt with value of 4925 N). Because of these rotational actions generate minimal changes in the axial forces of the bolts and do not put shear stress on the bolts, they can be neglected in the following calculation. Thus, this assumption is applied to tightened bolts as well as to lose bolts. In terms of loose bolts, the resultant axial force will be significantly lower contrasted to the original prestress from tightening the bolt. However, the force F_z has a rotational effect (about the y'-axis) on the bolts with the torque value (based on ***fig. 6***) (35) and causes a shear load on the bolts in the x-axis direction (36).

$$M_{zy'} = F_z \cdot (80 + 20) = 50 \cdot 0.1 = 5 \ N \cdot m \tag{35}$$

$$M_{zy'} = 2 \cdot F_{zy'} \cdot 25 \Rightarrow F_{zy'} = \frac{M_{zy'}}{2 \cdot 25} = \frac{5}{0.05} = 100 N \tag{36}$$

Force F_{xD} stresses the bolts to shear (in the x-axis), with load on each bolt (***fig. 7***) (37):

$$F_{x_{D1}} = \frac{F_{x_D}}{i_2} = \frac{47}{2} = 23.5 N \tag{37}$$

A similar method of calculation (with regard to ***fig. 7***), considering the neglect of rotational effects in the calculations for both tightened and loosened bolts and the facts stated in the preceding paragraphs, is employed in order to calculate the remaining torques and forces required, the values of which are given in ***TAB. 2***.

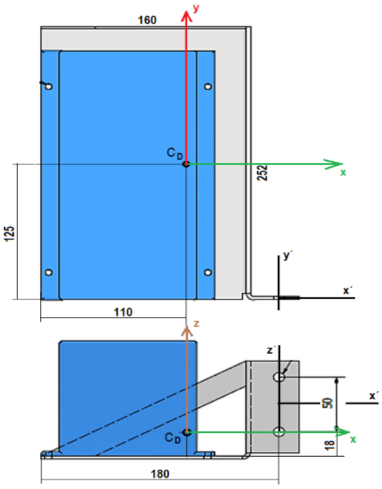


Fig. 7 *The considered position of the centre of gravity of the C_D "bracket" and the position relative to the x´, y´, z´ coordinate system*

Obr. 7 *Uvažovaná poloha ťažiska $Ť_D$ „držiaka" a poloha voči súradnému systému x´, y´, z´*

TABLE 2 *Calculated required torques and forces*

TAB. 2 *Vypočítané požadované momenty a sily*

Notation of torque/force	Value	Unit
$M_{xD_{z'}}$	6	N·m
$M_{xD_{y'}}$	1.2	N·m
$M_{yD_{z'}}$	1.2	N·m
$M_{yD_{x'}}$	0.4	N·m
$M_{zD_{x'}}$	6	N·m
$M_{zD_{y'}}$	3.3	N·m
$F_{xD_{y'}}$	24	N
$F_{zD_{y'}}$	66	N

Thus, the total force causing the shear load in the x-direction is given by the formula (38):

$$F_{S_x} = F_{x_{STB}} + F_{x_{y'}} + F_{z_{y'}} + F_{x_{D1}} + F_{xD_{y'}} + F_{zD_{y'}} = 25 + 10.5 + 100 + 23.5 + 24 + 66 \qquad (38)$$

$$F_{S_x} = 249\,N$$

The total force producing the shear load in the z-direction is represented by means of the equation (39):

$$F_{S_z} = F_{z_1} + F_{z_{D1}} = 25 + 23.5 = 48.5\,N \qquad (39)$$

Resultant shear load is (40):

$$F_{S_{2\max}} = \sqrt{F_{S_x}{}^2 + F_{S_z}{}^2} = \sqrt{249^2 + 48.5^2} \cong \boldsymbol{254\ N < F_{T2}/i_2 = 492.5\ N}, \qquad (40)$$

254 N < 492.5 N calculated per bolt.

Therefore, if the bolts are tightened, their utilization is safe, as demonstrated by the above calculations. The numbers of bolts and their diameters are designed with sufficient safety against all the forces acting on the attachment of the bracket to the frame. The bolted joints are correctly designed and respect the EN 12663-1:2010+A1:2014 standard. The calculated value of the maximum shear force corresponds to the maximum shear stress (41) in the event of the frictional force ceasing (by loosening the nuts):

$$\tau_{S_{2\max}} = \frac{F_{S_{2\max}}}{S_{j_2}} = \frac{254}{32.83} \cong 8\,MPa\,, \qquad (41)$$

which is a negligible value of the shear stress of the material of the bolts employed. Therefore, even in the case of loosening of the bolts, their utilisation is safe as shown by the above calculations. The numbers of bolts and their diameters are designed with adequate safety for the shear forces acting on the attachment of the bracket to the frame. The bolted joints, even in case of loosening, are designed accurately and respect EN 12663-1:2010+A1:2014.

5 CONCLUSION

The paper in question dealt with functional and strength calculations of bolted joints for mounting ETCS components in the HV5 vehicle, namely the electric motor unit class 660/661, according to the requirements of the normative document EN 12663-1:2010+A1:2014 for P-II passenger rolling stock. On the basis of the analysis of the achieved results, it is proved that:

– in case of tightened bolts, where transfer of forces is provided by friction, their utilisation is safe for each analysed bolted joint. The numbers of bolts and their diameters are designed with sufficient safety for all the forces acting on the attachment of the components to the vehicle body,

– in case of loose bolts, where forces are transmitted through the bolt shank, they are safe to use for each bolted joint scrutinised. The numbers of bolts and their diameters are designed with acceptable safety against the shear and tensile (compressive) forces acting on the attachment of the components to the vehicle body. In other words, all bolted joints presented in this work are designed correctly and respect the EN 12663-1:2010+A1:2014 standard, taking into account the P-II category for passenger rolling stock.

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Summary

The paper deals with the check calculations of the bolted joints of the fastening device STB (Set-top box) to the structure of the gross vehicle structure (vehicle body) according to the provisions of Commission Regulation (EU) No 1302/2014. Fixed devices, including devices inside passenger compartments, shall be fastened to the structure of the wagon body in order to prevent the loosening of these fixed devices, inasmuch as their loosening could pose a risk of injury to passengers as well as could lead to derailment. For this purpose, the fixing of these devices shall be designed according to the specification in question, considering category P-II for passenger rolling stock. The standard defines that for the sake of calculation of the forces in the attachments of the equipment in vehicle operation, the masses of the components must be multiplied by the defined accelerations, with regard to the condition that the load cases must be applied individually. As a minimum additional requirement, the loads resulting from the accelerations must be considered in combination with the maximum loads that the devices can generate themselves.

Resumé

Článok sa zaoberá kontrolnými výpočtami skrutkových spojov upevňovaného zariadenia STB (Set-top box) ku konštrukcii hrubej stavby vozidla (vozidlovej skrine) podľa ustanovenia nariadenia komisie (EÚ) č. 1302/2014. Pevné zariadenia vrátane zariadení vo vnútri priestorov pre cestujúcich sa upevňujú na konštrukciu vozňovej skrine tak, aby sa zabránilo uvoľneniu týchto pevných zariadení, keďže ich uvoľnenie by mohlo predstavovať riziko zranenia pre cestujúcich alebo by mohlo viesť k vykoľajeniu. Na tento účel sa upevnenie týchto zariadení projektuje podľa predmetnej špecifikácie, pričom sa zohľadňuje kategória P-II pre osobné železničné koľajové vozidlá. Norma definuje, že na výpočet síl v pripevneniach zariadení v prevádzke vozidla, musia byť hmotnosti komponentov násobené definovanými zrýchleniami s podmienkou, že prípady zaťaženia musia byť aplikované individuálne. Ako minimálna doplnková požiadavka musia byť zaťaženia vyplývajúce zo zrýchlení uvažované v kombinácii s maximálnymi zaťaženiami, ktoré zariadenia môžu generovať samotné.