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DESIGN OF AN ELEMENT USED TO ABSORB THE IMPACT ENERGY FOR A RAILWAY TRANSPORT VEHICLE

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Rezumat. *Lucrarea are ca obiectiv dimensionarea elementelor de absorbire a energiei de impact şi verificarea prin calcul a rezistenței mecanice a acestora, în cazul unui vehicul de transport pe şine. Calculul a constat în stabilirea optimă a diametrului interior, a grosimii peretelui şi a înălțimii cilindrului. În calculele efectuate, s-a avut în vedere standardul pentru țevi trase EN 10216:2002 astfel încât dimensiunile alese să corespundă cu cele posibil a fi livrate de industria de profil. Pe baza calculelor prezentate în lucrare, urmează să se elaboreze un model numeric ce va fi analizat cu ajutorul metodei elementelor finite şi validat experimental.*

Abstract. *The main objective of the paper is dimensioning of the impact energy absorption elements and verification of their mechanical strength in the case of a rail transport vehicle. The calculation consisted in optimal setting of the inside diameter, wall thickness and cylinder height. In the calculations, the standard for cold drawn pipes EN 10216: 2002 has been taken into account so that the chosen dimensions correspond to those delivered by the industrial suppliers. Based on the calculations presented in the paper, a numerical model will be developed, analyzed using the finite element method and validated experimentally.*

Keywords: Energy absorption, buffer, collision, railway, wagon.

1. Introduction

It is well known that structural design of a vehicle able to ensure the safety of passengers in all possible situations of accident and all combination of vehicle type, masses and velocities cannot be achieved with reasonable costs. That is why a satisfactory level of safety is desired by minimizing the risks of usual accidents. Passive structural safety is the ultimate means of collision protection when all other means failed.

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According to the European Standard EN 15227:2008+A1:2010 [1], protection of passengers, train conductors and dangerous goods in case of a railway accident is ensured by the following measures:

- Absorption of impact energy should be controlled;
- Values of deceleration (and implicitly of impact forces) should be limited;
- The surviving space for conductors and the areas for passengers should not be affected by a crash;
- Risk of wagon overlapping and risk of derail when hitting an obstacle should be reduced.

In the last 15-20 years, research on impact energy absorption through controlled plastic deformation has continuously grown due to the increase of railway traffic and speed.

Following numerical simulations and collision tests, Witteman [2] has drawn the following conclusions on impact energy absorption: a. The specific energy absorbed following an impact increases with the increase of wall thickness of a profile more than when the perimeter of the profile is increased; b. Circular, hexagonal or octagonal profiles absorb a bigger quantity of plastic energy than rectangular profiles; c. The maximum impact resistance force is decreased when a trigger (controlled decrease of the profile cross section) is used; d. A decrease with 10% of the thickness of the section offers a maximum absorption of impact energy. The final conclusion is that the hollow circular profile (pipe type) is preferred as it offers the best energy absorption, being also simple and economic.

Shakeri et al. [3] presented a very efficient procedure to absorb the impact energy. It uses two pipes, one made of a very rigid material compared to the other one. When compression due to impact takes place, the rigid pipe is introduced in the deformable pipe. Thus, the impact energy is dissipated in the plastic expansion of the deformable pipe and in the friction between the pipes.

Lučanin et al. [4] and Tanasković et al. [5] studied impact energy absorption based on the plastic deformation of a pipe, following the decrease of the diameter and friction when the pipe is inserted in a tapered ring. Both static and dynamic tests have been performed, showing that the force remains constant during motion of the pipe.

This paper describes the design of a new element for impact energy absorption, after studying the above-mentioned solutions proposed in the scientific literature and the features of railway traffic in Romania.

2. Equations of the central collision between two bodies

Collisions are mechanical processes of short duration. For two bodies of masses m_1 and m_2, travelling with velocities v_1 and v_2, conservation of the total momentum can be written as:

$$m_1\vec{v}_1 + m_2\vec{v}_2 = m_1\vec{v}_1' + m_2\vec{v}_2' \tag{1}$$

where v_1' and v_2' are the velocities after impact.

According to the conservation of the total energy:

$$\frac{m_1 v_1^2}{2} + \frac{m_2 v_2^2}{2} = \frac{m_1 v_1'^2}{2} + \frac{m_2 v_2'^2}{2} + Q \tag{2}$$

where Q is the energy dissipated during collision. A particular case of collision is the fully plastic one, when the two bodies travel together after collision as a single body with a mass $m = m_1 + m_2$ and speed v. In this case, equations (1) and (2) are written as follows:

$$m_1\vec{v}_1 + m_2\vec{v}_2 = m\vec{v} \tag{3}$$

$$\frac{m_1 v_1^2}{2} + \frac{m_2 v_2^2}{2} = \frac{m v^2}{2} + Q \tag{4}$$

Solving the system of equations (3) and (4), the energy absorbed during collision can be calculated as:

$$Q = \frac{m_1 m_2}{m_1 + m_2} \cdot \frac{(\vec{v}_1 - \vec{v}_2)^2}{2} \tag{5}$$

3. Masses and velocities of the railway vehicles in Romania

In order to correctly evaluate the kinetic energy and, consequently, the energy dissipated in the collision process, a survey on the masses and velocities of railway stock in Romania is necessary. All data can be found in [6]. The masses and velocities for the most representative railway vehicles in Romania are listed in Table 1.

Theoretical calculations in this research were made for a speed of 10 m/s or 36 km/h, which largely covers the maneuvering speeds. According to the statistics of the Romanian Railway Investigation Agency - AGIFER, over 98% of the Romanian railway accidents with frontal collisions took place at velocities that do not exceed 36 km/h, on rails starting from stations, depots and railyards.

As reference mass for calculations, values of 130 t (for heavy locomotives), 80 t (for freight wagons), 70 t (for light locomotives, diesel and electric railcars) and 50 t (for passenger cars, metro or trams) were considered.

Table 1 Masses and velocities for railway vehicles

Type of vehicle	*Maximum mass on axle [t]*	*Maximum total mass [t]*	*Maximum speed [km/h]*	
			Maneuver	*Journey*
Electric locomotive	21.5	129	40	160
Diesel-electric locomotive	21.5	129	40	160
Diesel-hydraulic locomotive	17.5	70	40	100
Electric railcar (*)	17	68	40	120
Diesel railcar (*)	18	72	40	120
Freight wagon (**)	20	80	40	120
Passenger car	12.5	50	40	160

(*) The vehicle includes the engine side.

(**) The cases of freight wagons with a maximum mass of 22.5t on axle and or with six axles with a total mass of 120t were not considered here.

4. Scenarios of collisions for railway vehicles

Crash technology is a method for controlled transfer and absorption of a very high impact energy under elevated speed collision conditions. It was developed in order to increase the level of passive safety in railway transport.

The European standard EN 15227:2008+A1:2010 [1] defines the safety requirements for collision in the case of railway vehicles and can be applied to the following: locomotives and passenger trains (category C-I), underground trains (C-II), light trains (C-III) and trams (C-IV).

The most usual collision scenarios from which victims can occur are:

A. Frontal impact between two identical railway vehicles from all four above-mentioned categories;
B. Frontal impact between two different railway vehicles (excluding C-II and C-IV);
C. Frontal impact between a railway vehicle traveling at a speed of no more than 110 km/h and a heavy road vehicle (15t) on a level crossing (excluding C-II and C-IV);
D. Frontal impact between a railway vehicle from C-IV and a road vehicle of smaller mass (3t) (excluding C-I, C-II and C-III)

All scenarios involve equipping railway vehicles with crash buffers that can absorb an impact energy of at least 400 kJ / buffer.

The combinations of collisions according to the scenarios described above and taking into account the masses and speeds of railway vehicles from Romania are listed in Table 2.

Table 2 Scenarios of collisions for Romanian railway vehicles

Mass of fixed obstacle	Masses and speeds of vehicles			
	130 t	80 t	70 t	50 t
130 t	10 m/s -15 m/s*	10 m/s -15 m/s*	10 m/s -15 m/s*	10 m/s -15 m/s*
80 t	10 m/s -15 m/s*	10 m/s -15 m/s*	10 m/s -15 m/s*	10 m/s -15 m/s*
70 t	10 m/s -15 m/s*	10 m/s -15 m/s*	10 m/s -15 m/s*	10 m/s -15 m/s*
50 t	10 m/s -15 m/s*	10 m/s -15 m/s*	10 m/s -15 m/s*	10 m/s -15 m/s*
15 t	10 m/s -15 m/s*	10 m/s -15 m/s*	10 m/s -15 m/s*	10 m/s -15 m/s*
	30 m/s	-	30 m/s	30 m/s
3 t	-	-	-	10 m/s-15 m/s*
	30 m/s	-	30 m/s	30 m/s

(*) The speed of 15 m/s = 54 km/h was taken into account following new studies within the SAFETRAIN research program, made by SNCF (the French rail administration) and the Canadian industrial group Bombardier Transportation, led by UIC (International Union of Railways) and ERRI (European Institute of Railway Research). These studies demonstrate statistically the increase of the average speed of collisions following the international development of railway transport sector.

5. Estimation of the impact energy absorbed during collision of two railway vehicles

If, before collision, one of the vehicles is at rest (v_2 = 0), the speed after a plastic collision and the energy dissipated in the impact can be inferred from equations (3) and (5) respectively:

$$v = \frac{m_1 v_1}{m} = \frac{m_1 v_1}{m_1 + m_2} \tag{6}$$

$$Q = \frac{m_1 m_2}{m_1 + m_2} \cdot \frac{v_1^2}{2} \tag{7}$$

The energy dissipated on each end of the rail vehicle using two high-performance crash buffers is of maximum 2 × 0.5MJ = 1MJ from the total impact energy Q. Thus, it remains to dissipate Q'= Q – 2MJ (for two vehicles) or, for one vehicle Q'' = Q – 1MJ.

Taking into account the data from Table 2, and applying equations (6) and (7), one can calculate the speeds after collision and the energy dissipated during collision. The results are listed in Table 3. Analysis of these results reveal that if another 1.1 MJ of energy is absorbed through the vehicle end (0.55 MJ per buffer), most of the collisions considered in Table 2 would be covered, except those marked in red. In Table 3, the cases already covered by the use of the current crash buffers are listed in green while those that could be covered by using supplementary elements for energy absorption are listed in blue.

It can be noticed that an increase of the impact speed v_1 with 50% increases the impact energy Q 2.25 times.

Table 3 Speeds after collision and energy dissipated in collision

Mass m_2 of the obstacle		Masses and speeds of railway vehicles							
		m_1=130t		m_1=80t		m_1=70t		m_1=50t	
v_2=0	v_1	10m/s	15m/s	10m/s	15m/s	10m/s	15m/s	10m/s	15m/s
130t	v	5m/s	7.5m/s	3.8m/s	5.7m/s	3.5m/s	5.3m/s	2.8m/s	4.2m/s
	Q	3.25MJ	7.31MJ	2.48MJ	5.58MJ	2.28MJ	5.13MJ	1.81MJ	4.07MJ
80t	v	6.2m/s	9.3m/s	5m/s	7.5m/s	4.7m/s	7m/s	3.8m/s	5.8m/s
	Q	2.48MJ	5.58MJ	2MJ	4.5MJ	1.87MJ	4.21MJ	1.54MJ	3.47MJ
70t	v	6.5m/s	9.8m/s	5.3m/s	8m/s	5m/s	7.5m/s	4.2m/s	6.3m/s
	Q	2.28MJ	5.13MJ	1.87MJ	4.21MJ	1.75MJ	3.94MJ	1.46MJ	3.29MJ
50t	v	7.2m/s	10.8m/s	6.2m/s	9.2m/s	5.8m/s	8.8m/s	5m/s	7.5m/s
	Q	1.81MJ	4.07MJ	1.54MJ	3.47MJ	1,46MJ	3.29MJ	1.25MJ	2.81MJ
3t	v	-	-	-	-	-	-	9.4m/s	14.2m/s
	Q	-	-	-	-	-	-	0.14MJ	0.32MJ
15t	v	9m/s	13.4m/s	8.4m/s	12.6m/s	8.2m/s	12.4m/s	7.7m/s	11.5m/s
	Q	0.67MJ	1.51MJ	0.63MJ	1.42MJ	0.62MJ	1.40MJ	0.58MJ	1.31MJ
	v_1	30m/s		-		30m/s		30m/s	
	v	26.9m/s		-		24.7m/s		23.1m/s	
	Q	6.05MJ		-		5.56MJ		5.19MJ	
3t	v	29.3m/s		-		28.8m/s		28.3m/s	
	Q	1.32MJ		-		1.29MJ		1.27MJ	

After absorbing a part of the impact energy, it would remain to dissipate using crash buffers and supplementary elements another part equal to: $Q' = Q - 4.2$ MJ (for two vehicles) and $Q'' = Q - 2.1$ MJ (for one vehicle).

The maximum force that can be applied to the wagon end should not exceed 2.5 MN, i.e. 1.25 MN/buffer (the value of 2 MN cannot be exceeded with more than 25% due to strength considerations: the safety coefficients used to design such vehicles have a minimum value of 1.3 and decelerations should be less than 5*g* in order to protect passengers). To achieve the dissipated energy of 1.1 MJ (equal to the mechanical work of the 2.5 MN plastic deformation force) it must be quasi-constant over the plastic deformation length of the absorbent element: L = 1.1 MJ / 2.5 MN = 0.44 m.

6. Dimensioning of the absorption elements

Since the impact energy has elevated values, a material that is able to absorb it is needed. A non-alloyed steel with good plasticity and resilience is preferred. Although it's specific weight is higher compared to the one of composite materials, it has many advantages: a. Well-known and widespread manufacturing and processing technology with relatively low prices per unit of mass; b. Long service life with minimal maintenance costs comparable to rail vehicle structures (over 30-40 years); c. Low cost and relatively easy implementation on old structures already in operation and d. Easy recycling at the end of the life of the rail vehicle.

A cylindrical structural element was chosen since it has good strength, given by the symmetry with respect to the longitudinal axis and to all planes that contain this axis. An area for energy absorption is created using a matrix of such elements located on the frontal cross-beam of the wagons. The strength calculation has as main tasks to find the optimal inner diameter and wall thickness for compression loading. The structure should remain elastic and keep it's stability for forces up to 2.5 MN. Plastic deformation for forces higher than 2.5 MN should occur only by the increase of the diameter and thinning of the wall till failure. This will be achieved by inserting a very rigid, high-strength steel tapered element in the pipe.

The elements lying in front of the frontal cross-beam should not deform plastically simultaneously at compression forces smaller than 2 MN, condition imposed by the standards EN 12663-1/2010 [7] and EN 12663-2/2010 [8]. In the same time, these elements should deform plastically in a controlled manner for compression loadings higher with 25% than the limit of 2 MN (i.e. 2.5 MN), in order to protect the strength structure of the chassis of the wagon.

According to the previous hypotheses taken into account, the force at which the protection elements in front of the frontal cross-beam should deform plastically is 2.5 MN. Taking into account the overall size restrictions of the wagon, the maximum length of the assembly of protection elements is 550 mm. From the condition that the absorbed energy of 1.1 MJ is equal to the mechanical work of the force, it yields: $E = 2.5\ \text{MN} \times l = 1.1\ \text{MJ}$, leading to a length of the structural element $l = 0.44$m.

Since a number of 10 identical elements, symmetrically placed on the frontal cross-beam will be considered, then the compression force applied on each element and which yields plastic deformation through expansion is 250 kN.

Further, only one element will be analyzed. The location of these elements is shown in Figures 1 and 2 only for half of the frontal cross-beam, due to symmetry.

The outer diameter of the pipe was chosen with the value of 89 mm with a wall thickness of 4mm. Consequently, the cross section area is A = 1068 mm^2. Taking into account the value of the compression force F = 250 kN, a compression stress results as $\sigma = F/A = 234$ MPa.

The pipe has a fixed end and the other is deformable using the tapered penetrator. The whole assembly of energy absorption elements is fixed to the central cross-beam using four prestressed, high strength screws of diameter 24 mm, which can ensure a fastening force of more than 400 kN.

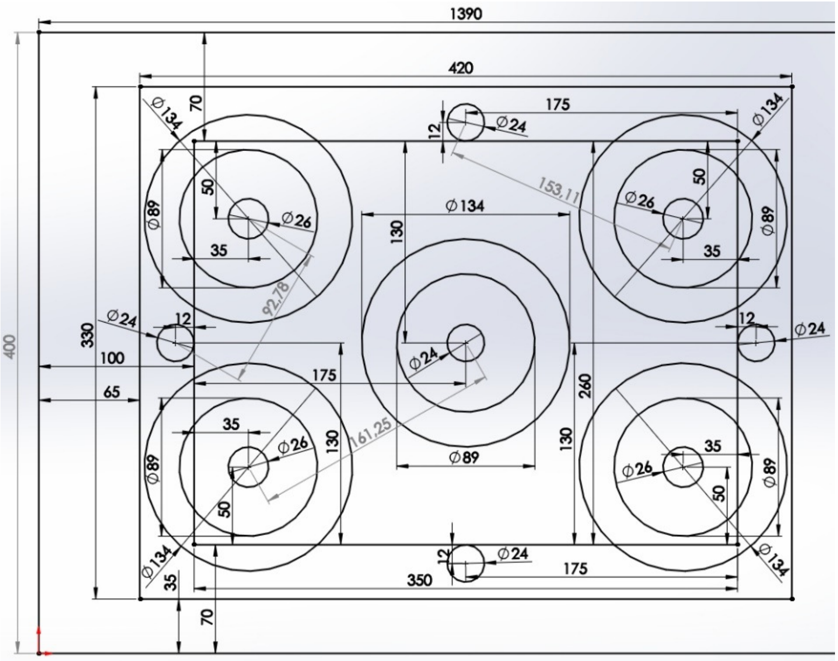

Fig. 1. Location of the energy absorption elements for half of frontal cross-beam

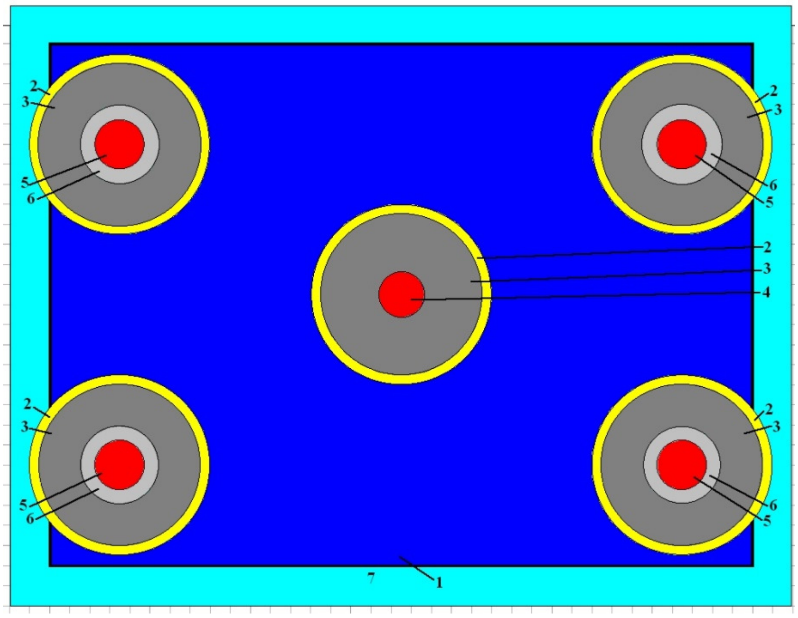

Fig. 2. Location of the energy absorption elements under the buffer pad: 1. Buffer pad; 2. Deformable pipe; 3. Guiding flange; 4. Fastening screw for the central guiding flange; 5. Fastening screw for the buffer; 6. Elastic washer for fixing the guiding flange; 7. Fixing plate

The main characteristics of the materials considered for the energy absorption elements are listed in Table 4.

Table 4 Characteristics of the used materials

Characteristics of the material	*Element*	
	Tapered penetrator	*Cylindrical pipe*
Type of steel	45Cr2	S232JR
Young’s modulus [MPa]	$1.9 \cdot 10^5$	$2.1 \cdot 10^5$
Poisson’s ratio	0.28	0.28
Mass density [kg/m^3]	7800	7800
Ultimate tensile strength [MPa]	780	360
Yield limit [MPa]	540	235
Elongation at break [%]	Minimum 12	Minimum 24
Thermal dilatation coefficient [K^{-1}]	$1.1 \cdot 10^{-5}$	$1.1 \cdot 10^{-5}$
Thermal conductivity [W/(m·K)]	14	14
Specific heat [J/(kg·K)]	440	440
Resilience [J]	Minimum 35	Minimum 27

A longitudinal cross section of the designed assembly is shown in Figure 3.

Calculations of the pipe were made taking into account the condition that the variation of the internal diameter during deformation should be less than 24% so that failure does not occur when the maximum diameter of the penetrator is reached.

The penetrator (Figure 4) is made of two cylinders with different radii, superimposed over a truncated cone in the center, with the center angle of 60° and the length of the generatrix of 20 mm. The connection was made using a 5 mm radius between both cylindrical and tapered surfaces. The lower cylindrical portion, 40 mm radius and 5 mm height, serves to guide the tapered penetrator on the inside of the pipe. The tapered penetrator was designed with a center angle of 60°. It’s behavior at compression with a force of 250 kN was studied, to ensure that only small elastic deformations occur, due to the chosen material which is a high strength alloy steel 45Cr2.

The support flange for cutting (Figure 5) has a radius of 50 mm and a height of 10 mm and it contains eight cutting tools, evenly distributed across the cross section. Each cutting tool has a width of 3 mm and exceeds the maximum diameter of the penetrator by 3 mm. The cutting tools produce eight channels in the expanded pipe material along eight generatrices positioned symmetrically with respect to the longitudinal axis of the pipe in order to obtain a controlled reduction of the cross-section of the pipe.

The design of the bending flange is depicted in Figure 6. A longitudinal section of the assembly after the beginning of the plastic deformation of the pipe is presented in Fig. 7.

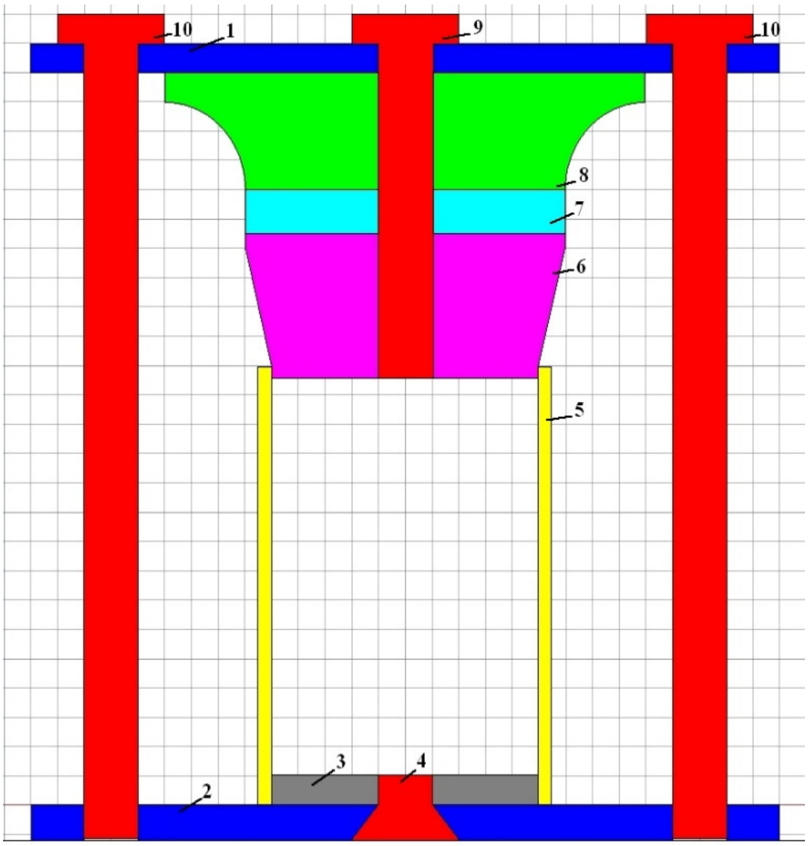

Fig. 3. Longitudinal cross section through the energy absorption assembly: 1. Frontal cross-beam; 2. Fixing plate for the penetrating assembly and the buffer pad; 3. Cylindrical guiding flange for the deformable pipe; 4. Fastening screw for the guiding flange; 5. Deformable pipe; 6. Tapered penetrator; 7. Support flange for cutting; 8. Bending flange; 9. Attachment screw for the penetrating assembly; 10. Prestressed fastening screws.

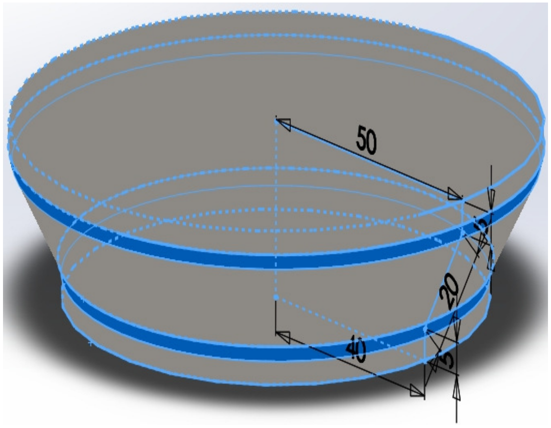

Fig. 4. The tapered penetrator

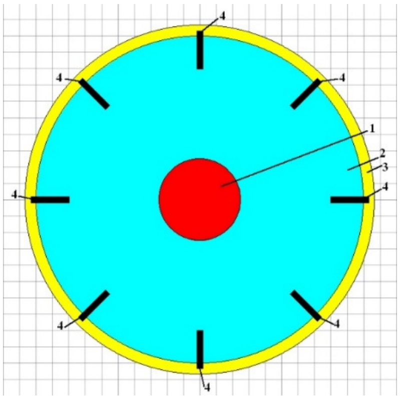

Fig. 5. The support flange for cutting: 1. Attachment screw for the penetrating assembly; 2. Support flange for cutting; 3. Deformable pipe; 4. Cutting tools

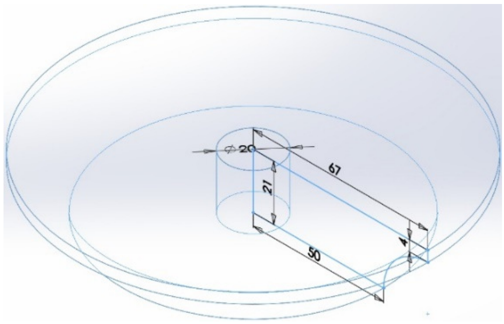

Fig. 6. The bending flange

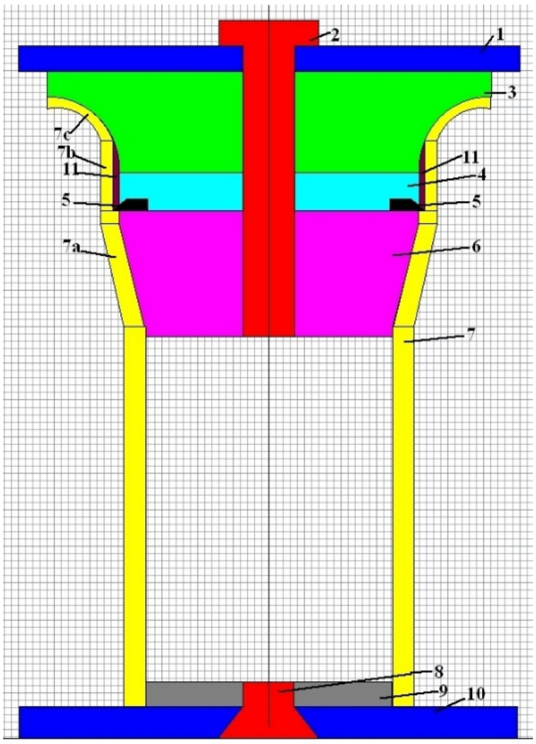

Fig. 7. A longitudinal section of the assembly after the beginning of the plastic deformation: 1. Frontal cross-beam; 2. Fixing screw for the penetrating assembly ; 3. Bending flange; 4.Support flange for cutting; 5. Cutting tools; 6. Tapered penetrator; 7. Undeformed pipe; 7.a. Expanded pipe; 7.b. Pipe with internal channel manufactured by cutting; 7.c. Pipe bent at 90°; 8. Fastening screw for guiding flange; 9. Cylindrical guiding flange for deformable pipe; 10. Fixing plate for the penetrating assembly and the buffer pad; 11. Channels machined by cutting

7. Stability calculations

The protection element was designed as a cylindrical pipe having a length L = 480 mm, with an internal diameter d = 81 mm and external diameter D = 89 mm. The stability of the pipe element under compression load was checked to ensure that only plastic deformation will occur without buckling. For this, the slenderness ratio was λ determined as:

$$\lambda = l_b \cdot \sqrt{\frac{A}{I_{\min}}} \tag{8}$$

where A = 1068 mm^2 is the cross section area of the pipe and $I_{\min}$ is the minimum moment of inertia which is

$$I_{\min} = \frac{\pi\left(D^4 - d^4\right)}{64} = \frac{\pi\left(89^4 - 81^4\right)}{64} = 966700\text{mm}^4$$

The parameter l_b is the buckling length, which, in this case is calculated for a bar fixed at both ends and is $l_b = 0.5L = 240$ mm.

Using equation (8), a slenderness ratio $\lambda = 8$ is obtained. For the chosen material, the following slenderness ratios are known: $\lambda_0 = 105$ and $\lambda_1 = 60$. Elastic buckling occurs for $\lambda > \lambda_0$, plastic buckling appears if $\lambda \in (\lambda_1, \lambda_0]$ while for $\lambda \leq \lambda_1$ buckling does not occur and only plastic deformation due to yielding is expected, which is the case in the studied structure.

8. Evaluation of the energy absorption processes with the designed assembly

From the whole available length of 550 mm, about 100 mm are occupied by the assembly penetrator – guiding flange and the fastening screws. Consequently, only a length of 450 mm can be considered as active part of the designed element. On this length, it is expected to obtain absorption of maximum 1.1 MJ (meaning 110 kJ for each of the 10 elements) under the action of a force of maximum 2.5 MN. Calculation of the energy absorbed by different processes is shown further.

8.1. Absorption of energy by expanding the pipe with 24 %

Increasing the inside diameter by 24% also involves increasing with the same percentage the circumference of the pipe across the length of 450 mm:

$$\Delta l_c = 0.24 \times \pi \times 89 = 67.1\text{mm}$$

The area of the longitudinal section of the deformable pipe portion is:

$$A = 4 \times 450 = 1800 \text{ mm}^2$$

The required force at the ultimate tensile strength of the material will be:

$$F_m = A \times \sigma_u = 1800 \times 360 = 648 \text{ kN}$$

The mechanical work of deformation by expanding will be:

$$L_e = F_m \times \Delta l_c = 648 \times 67.1 = \mathbf{43.5\ kJ}.$$

8.2. Absorption of energy by friction between the surface of the penetrator against the inner surface of the pipe

Taking into account the angle of 30° between the generatrix of the tapered surface with the vertical direction, the force normal to the surface is:

$$F_n = F \times \sin 30° = 125 \text{ kN}$$

The area of the tapered surface is:

$$A_t = \pi \times G \times (R + r) = 5655 \text{ mm}^2$$

The value of the applied pressure is:

$$p_n = F_n / A_t = 22.1 \text{ MPa}$$

This pressure should produce in the wall of the pipe a pressure at least equal to the yield limit from Table 4, [9]. The obtained value is:

$$\sigma = p_n \cdot (D + s)/2s = 235 \text{ MPa}$$

In the friction process, according to the molecular-mechanic theory, the force on the movement direction has two components: one necessary to overcome the molecular adhesion and the other that produce deformations. In the case of total plastic contact, the friction coefficient has two components and is defined as [10]:

$$\mu = (\tau_0/\sigma_y + \beta) + k_p \cdot (p_n/\sigma_y)^{1/2} = 0.89$$

where τ_0 = 110 MPa is the molecular specific strength of the material, σ_y = 235 MPa is the yield limit, β = 0.25 is the piezocoefficient characteristic for friction and k_p = 0.55 is the roughness coefficient [10]. Consequently, the friction force has the value:

$$F_f = \mu \times F_n = 111.2 \text{ kN},$$

and the mechanical work of the friction force is:

$$L_f = F_f \times L = \mathbf{50.1\ kJ}$$

8.3. Absorption of energy by cutting the pipe

In the case of cutting, shear of the material occurs in the area of action of the cutting tools. The area of the cross section for the eight machined grooves is:

$$A_f = 8 \times 3 \times 3 = 72 \text{ mm}^2.$$

For ductile steels, the ultimate shear strength can be approximated as $\tau_f = 0.8\sigma_u$, and so, the force required for cutting has the value:

$$F_c = A_f \times \tau_f = 20.7\text{kN}$$

Taking into account the length of 420 mm for the machined zone (by decreasing the length of the penetrator from the 450 mm of the active portion), the mechanical work for cutting is:

$$L_c = F_c \times 0.42 = \mathbf{8.7\ kJ}.$$

8.4. Absorption of energy by bending the pipe

Plastic bending involves loading the material beyond its yield limit over the entire trajectory of the plastic deformation. The area of the eight 24% thinned sectorial pipe sections resulting from cutting is:

$$A_b = 1068 \text{ mm}^2 - A_f = 996 \text{ mm}^2$$

For each section, the area is:

$$A_s = b \times h = 39 \times 3.2 = 125 \text{ mm}^2.$$

It can be considered that the is bar supported at both ends and has a bending length equal to the length of an arc of circle with the radius equal to 17mm, i.e. $l_b = \pi r/2 = 26.7$ mm and a bending stress σ_b equal to the ultimate strength. The force applied for one section is:

$$F_b = 2b \times \text{h}^2 \times \sigma_b / 3l = 3.6 \text{ kN}$$

The mechanical work in bending for one portion of section is:

$$L = F_b \times r = 61.2 \text{ J}$$

In the 410 mm available there are 15.4 such areas multiplied by 8 times. Finally, the total mechanical work in bending is:

$$L_b = L \times 8 \times 15.4 = \mathbf{7.6\ kJ}.$$

8.5. The total energy absorbed by an element through various methods of controlling the plastic deformation

By summing the mechanical work from each energy absorption process, the total energy absorbed by an element obtained after using the controlled plastic deformation method is:

$$E = L_e + L_f + L_c + L_b = 109.9 \text{ kJ}$$

This calculation proves that the value of 110 kJ of the energy required to be dissipated by a single element can be achieved.

One can notice that the dissipated bending and cutting energies are relatively comparable, but much lower than those for expansion and friction. Also, changing the number of cutting tools does not significantly affect the dissipated energy, as opposed to changing the penetrator angle and contact surface condition

Conclusions and future research

The paper presents a new assembly used to absorb energy during collision of railway vehicles. The main goal of the research is to evaluate the energy absorbed by the elements of the assembly and to assess if this energy is enough to limit the consequences of a collision between locomotives and/or wagons. The theoretical calculations have shown that the proposed solution for the elements of the impact energy absorbers could be achieved practically. The chosen solution could be effective even without the use of cutting and bending processes through controlled deformation to absorb impact energy. Implementing such a solution for new or existing railway vehicles would not raise any special technical or logistical problems. Also, it should be mentioned that the costs for implementing are relatively low compared to the benefits obtained in an unwanted case of collision.

Future research can validate experimentally the proposed solution. To achieve this goal, after the design process, the next steps are to develop a numerical model of the energy absorber element using CAD/CAE programs, to analyze the proposed mechanical structure using the finite element method and to validate the solution by testing it under laboratory conditions. Finally, the obtained results could be extrapolated for a wider range of similar solutions.

REFERENCES

[1] EN 15227:2008+A1:2010 - Railway applications – Crashworthiness requirements for railway vehicle bodies

[2] W.J. Witteman, *Improved Vehicle Crashworthiness Design by Control of the Energy Absorption for Different Collision Situations* - Ph.D. Thesis, Technische Universiteit Eindhoven, (1999).

[3] M. Shakeri, S. Salehghaffari, R. Mirzaeifar, Int J Crashworthiness **12**(5), 493-501 (2007).

[4] J.V. Lučanin, J.D. Tanasković, D.D. Milković, S. Golubović, FME Transactions, **35**(4), 201-204 (2007)

[5] J.D. Tanasković, D.D. Milković, J.V. Lučanin, - FME Transactions, **40**(1), 11-16 (2012).

[6] Regulation no. 006 from 26 October 2005 for towing and braking), Emitted by Ministry Of Transport; published in the MO no. 1043 bis in 24.11.2005 (in romanian)

[7] EN 12663-1:2010 - Railway applications - Structural requirements of railway vehicle bodies - Part 1: Locomotives and passenger rolling stock.

[8] EN 12663-2:2010 - Railway applications - Structural requirements of railway vehicle bodies - Part 2: Freight wagons

[9] R. T. Munteanu, Bulletin of AGIR **2**, 76-78 (2013) (in Romanian).

[10] A. Tudor, *Tribosisteme Integrate – Note de Curs (Integrated tribosystems – lecture notes)*, Faculty of Mechanical Engineering and Mechatronics, University Politehnica of Bucharest, http://www.omtr.pub.ro/didactic/tribosisteme.htm, accessed 31.05.2019.