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Research paper

Design of a storage mechanism designed for a railway vehicles producer

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Abstract

In this article the engineering design and strength analyses of a storage mechanism are presented. The handling device is intended to be placed in a warehouse and it will serve as a storage mechanism for longer metallurgical material in a railway vehicle manufacturing company. Stored material will be used for creation of some parts of railway wagons. It will allow to storage material in several levels in shelves. To meet requirements for manual control of the pull-out mechanism it will be necessary to design a pull-out mechanism with the sufficiently large transmission ratio in order to handle a shelf block weighing up to given load. Consequently, there are conducted the numeric strength analysis of steel construction which has to safely carry the total load of ten shelves.

Keywords: *Transport and handling machines, storage mechanism, numerical analysis, functional calculation.*

1. Introduction

Storage cannot be separated from material flows in all spheres of economy. The need for storage of material of all kinds arises due to the different pace of production and consumption, the different flow in all levels of a logistic chain. Moreover, the storage forms an inevitable part of the production technology. We can deal with warehouse issues in different ways, with reference to building solutions, the organization, technical equipment and many others. From the point of view of logistic objects it is particularly suitable to pay attention to three types of warehouses, namely the warehouse of bulk material, metallurgical material and pallets. Material characteristics, material amount, and storage technology requirements in connection with other processes are essential for the right choice of warehouses. In analysing the kind of metallurgical material in a certain engineering company it was found out that only 25 % of items are material of one item in bulk with a large turnover rate of revenues and costs.

The rest of material weighed less though their items formed 70 % up to 80 % of the whole items in that warehouse. That`s why it is necessary to provide with optimal conditions for storage and handling and at the same time it is simple to solve the technology of handled items offtake. Such piles of rod material are to be stocked in shelves (or racks) and we recommend to seize them by means of a grab bucket. Using this way of handling we do not need to bind material and in terms of controlling it is very suitable. The railway vehicles manufacturing company has to stock metallurgical material – rod material with the length of 5 meters intended for creation some particular parts of wagons, so for this reason it will be necessary to design the storage mechanism, in our case we call it the rack system with regard to customer`s requirements (Fig. 1).



Fig. 1: Metallurgical material warehouse within a company

2. Theoretical background on the steel construction design

Generally, handling equipment consists of bearing construction, driving mechanism and other parts (various additional equipment, e.g. a cab). Such a steel construction represents the main load-bearing part of any handling device. Mostly it means the bearing construction of all equipment working parts which transmits all external loads affecting the equipment. The shape of the steel construction is created in order to fulfil the aim of its application. Nowadays, execution of the construction is usually performed by means of welding individual parts. In the past, many handling machines have been assembled by riveted joints. Because of welding advantages, this technological process is considered also for our rack system production (Fig. 2).

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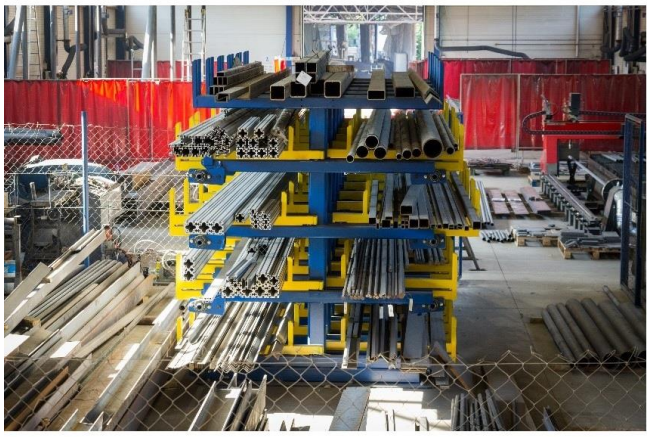


Fig. 2: The designed construction and its coming into operation

The safety of construction is one of the strictest demands of customers. Moreover, structures should to be light and strength enough in order to guarantee, that the production and use will be economic. Further, design of a construction structure have to be designed in such a way, that handling and access with all parts will be safety and convenient.

As for production of our steel structure welding is used, considered material (in our work we have used the structural steel) has to be weldable. Such steel contains less than 0.20 % of carbon. As the basic material of the structure and also for same more loaded parts of the structure we have used higher grades of steel, namely S355 [1]. The dimensioning of the whole structure resulting from specific conditions. For saving the occupied space, the structure allows to storage of material in several levels above each other. Further, in the structure designing we have considered the important fact. If there are storage various kinds of material into the rack, the manipulation with material (i. e. unloading and loading) would be time-consuming. Therefore, a general recommendation on the handling machines design is to construct them with one degree for freedom, which reduces the consumed time for operation and handling and simplifies manipulation significantly [2].

3. Construction design of rack system

The storage method ensures demanded movement possibilities, i.e. the rack movement. The material in weight of 3000 kilos will be stored on eight retractable racks. Each retractable rack is divided into six parts – webs mutually connected with shafts by means of shaft coupling and they are pulled out together by a crank handle.

Working with the sixths model means to save calculation because the construction is designed symmetrically from the view of geometry, fixing and loading.

Table 1: Construction parts

Structure	Dimensions of square tubing [mm]	Material
Central beam	160 × 160 × 10	S235
Horizontal beams	100 × 80 × 6	S355
Racks	80 × 60 × 4	S355

The construction is made from thin-walled square tubing which dimensions and materials are put in Table 1.

The construction design of steel construction consists of three elements (Fig. 3), i.e. bearing construction of central, vertical beams and horizontal beams, into which racks are pulled in and out, are welded to this construction.

Whereas the transmission of construction mechanism is driven by a crank handle, the racks will move forward at low speed, so the dynamic stress of construction is not significant. We count the maximum of allowed stress according to (1):

$$\sigma_a = \frac{R_e}{k}, \tag{1}$$

where R_e is the yield strength and k is the coefficient of safety. We consider for material yield strength of horizontal beams and racks R_e = 360 MPa and for coefficient of safety with regard to the static way of loading the construction k = 1.7 (-).

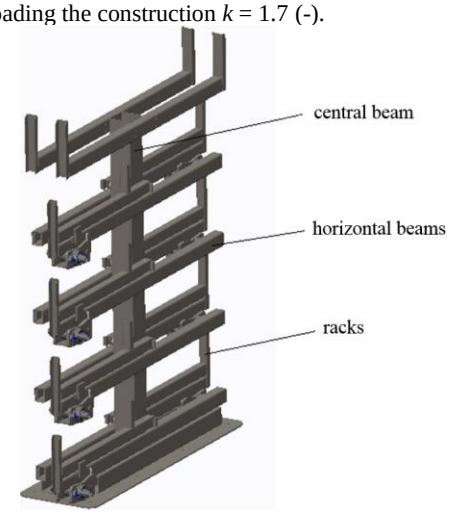


Fig. 3: Sixths model of rack system construction design

For material yield strength of central beams we consider R_e = 240 MPa and for coefficient of safety with regard to the static way of loading the construction k = 1.7 (-). Then for allowed stress in the construction by using material S355 steel we get the amount σ_a^{355} = 211 MPa and for material S235 steel σ_a^{235} = 141 MPa by means of the equation (1). In the place of weld of each construction parts it is necessary to decrease in the allowed stress of the amount which is experimentally practiced and decreases in the allowed stress of approximately 30 % in the place of weld. We include this fact in the calculation by using the coefficient of the influence of welding c (2):

$$\sigma_{aw} = \sigma_a \cdot c, \tag{2}$$

where σ_a is the allowed stress of basic material (calculated according to eq. 1). To reduce the allowed stress amount in the place of weld by recommended 30 % we choose the amount of welding coefficient c = 0.7 (-). By solving the equation (2) we get allowed amounts of stress in the place of welds for material S355 steel σ_{aw}^{355} = 148 MPa and for material S235 steel σ_{aw}^{235} = 98 MPa.

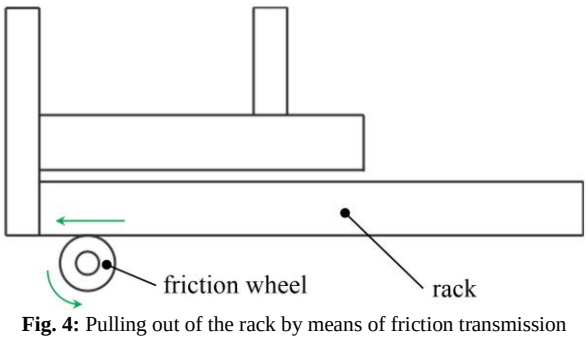


Fig. 4: Pulling out of the rack by means of friction transmission

The realization of the mechanism for webs movement will be carried out by means of a friction wheel (Fig 4) and we will expect low initial costs because its production technology is not really difficult.

Moreover, there is no need for much maintenance because it is not necessary to lubricate, even it is not allowed.

Another main advantage is a low frequency of failure rate. Hand drive is not suitable for all constructions because we should take into account person`s performance and comfort. We can use it only for a small load capacity, a low lift, and low working speeds during the occasional, not permanent use. When designing the hand drive, we have to remember that the permanent performance of an average person is about 75 W. And as well, we have to take

into account the ergonomic point so that a worker can use the equipment without any difficulty. The best way of transferring strength is by a handle at maximum circular speed of $1\ m \cdot s^{-1}$. During the permanent work of one worker (considering his performance) the force on the crank is set to the maximum of 100 N. It is also possible to increase the force up to 200 N during 10 minutes, of course in case of reducing the speed. Thus, the construction design of rack system follows (Fig. 5).

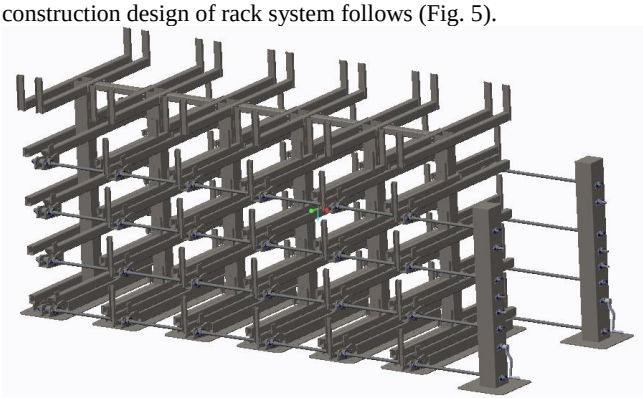


Fig. 5: 3D model of the designed rack system with pull-out webs

4. Functional calculation of the force on the hand crank

In Figure 6 we can see a scheme of a rack part with the pull-out friction mechanism. There is material put on the upper square tubing (1). This upper and as well lower square tubing (2) is pulled out by means of the friction wheel (3). The wheel (3) is firmly connected with the shaft (7) which is driven through the chain transmission (8) by means of the hand crank (9). The lower square tubing (2) is put into the larger square tubing (6). While pulling out the upper bearing (4) helps the movement and it is moving on the surface of the square tubing (6) without touching of the lower bearing (5) with the inner surface of the square tubing (6). With pulling in of the rack the lower bearing (5) helps, it moves on the inner surface of the square tubing (6) and the upper bearing does not touch the square tubing surface (6).

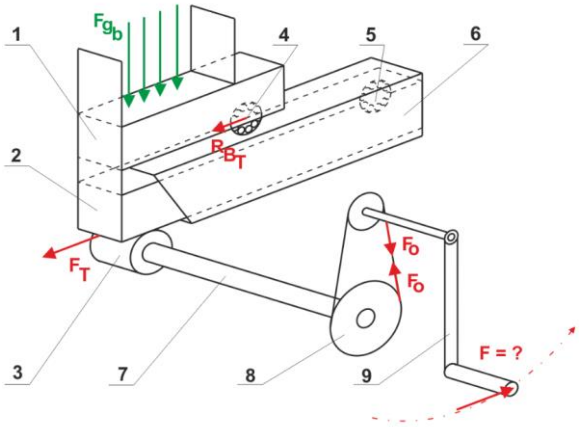


Fig. 6: The schematic design of the rack pull-out mechanism

Supposing low working speeds and smooth motion we did not pay attention to dynamic effects while designing the mechanism, so we simplified the dynamic model and designed the static model. The mechanism is released into two positions because of having two extreme positions. One position is when the rack is pull-in and it is needed to be pulled out.

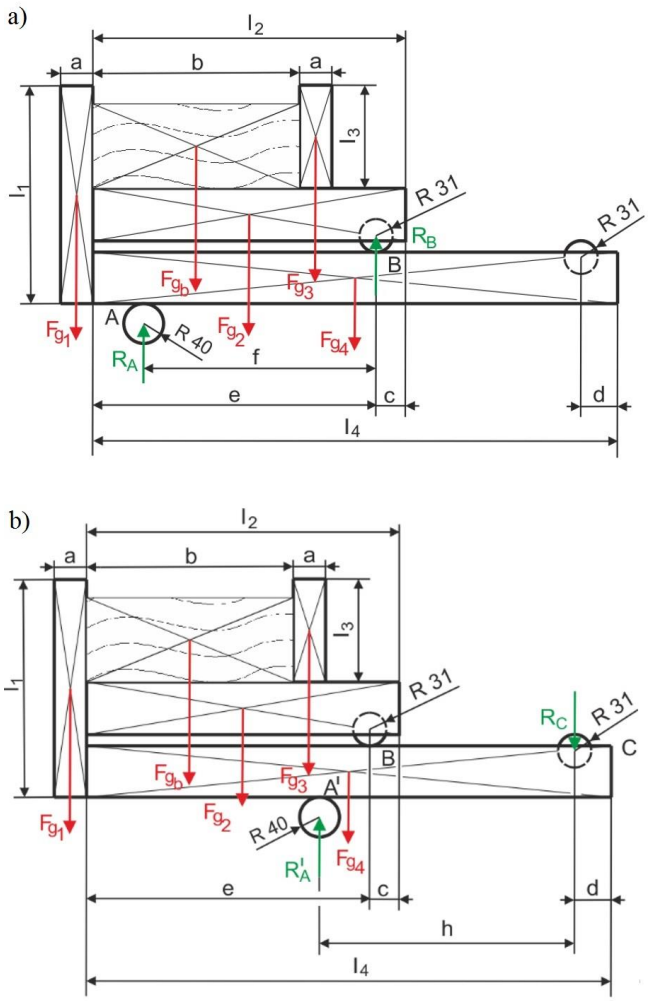


Fig. 7: Releasing of the mechanism for pulling out a) and pulling in b)

In Figure 7a forces and reactions are described in the beginning of pulling out the rack. We suppose that the force which releases the mechanism into motion will be greater than the force which is needed to keep it in motion, because the slide friction coefficient is greater in stillness than in motion. We also take into account the other situation when the rack is pulled in from the fully pull-out position. Then the reactions will be different, as it is seen in Figure 7b.

Table 2 presents the data relating to mechanism parts and they were used during calculations.

Table 2: Values of the rack mechanism geometry

Dimension	Value [mm]	Dimension	Value [mm]
a	40	f	958
b	900	l_1	400
c	82	l_2	1200
d	103	l_3	200
e	1118	l_4	2095

After calculating on the values of load forces in the mechanism, i.e. weight forces from load and weight of the moving parts of the construction, it is possible to determine important reactions in the friction wheel and auxiliary wheel B in the pull-out position, and reactions in the friction wheel and auxiliary wheel C in the pull-in position. The values of weight forces of a rack result from linear mass density of used tubing and values of weight acceleration. The effects of forces are: $F_{g1} = 19.29$ N, $F_{g2} = 115.72$ N, $F_{g3} = 9.64$ N, $F_{g4} = 202.03$ N and $F_{gb} = 4905$ N. During pulling out we can count the reaction itself from the amount of moments towards the point B (eq. 3) (Fig 7a) and from the amount of forces in vertical direction (4):

$$\sum M_{iB} = 0 \Rightarrow F_{g1} \cdot \left(\frac{a}{2} + l_2 - c\right) + F_{g2} \cdot \left(\frac{l_2}{2} - c\right) + \\ + F_{g3} \cdot \left(l_2 - b - a - c\right) + F_{g4} \cdot \left(e - \frac{l_4}{2}\right) + \\ + F_{gb} \cdot \left(e - \frac{b}{2}\right) - R_A \cdot f = 0 \quad (3)$$

If we put numeric values in the given equation (3) instead of variables, for the sought reaction force we find out: R_A = 3520.86 N.

$$\sum F_{iy} = 0 \Rightarrow R_A + R_B = F_{g1} + F_{g2} + F_{g3} + F_{g4} + F_{gb} \quad (4)$$

If we put numeric values in the given equation (4) instead of variables, for the sought reaction force we get R_B = 1730.82 N. During pulling in we count the reactions from the amount of moments towards the point C (5) (Fig. 7b) and from the amount of forces in vertical direction (6):

$$\sum M_{iC} = 0 \Rightarrow F_{g1} \cdot \left(l_4 - c + \frac{a}{2}\right) + F_{g2} \cdot \left(l_4 - d - \frac{l_2}{2}\right) + \\ + F_{g3} \cdot \left(l_4 - d - b - \frac{a}{2}\right) + F_{g4} \cdot \left(\frac{l_4}{2} - d\right) + \\ + F_{gb} \cdot \left(l_4 - d - \frac{b}{2}\right) - R_A^{'} \cdot h = 0 \quad (5)$$

If we put numeric values in the given equation (5) instead of variables, for the sought reaction force we get $R_A^{'}$ = 8685.42 N.

$$\sum F_{iy} = 0 \Rightarrow R_C = \left(F_{g1} + F_{g2} + F_{g3} + F_{g4} + F_{gb} - R_A^{'}\right) \quad (6)$$

If we put numeric values in the given equation (6) instead of variables, for the sought reaction force we get R_C = 3433.74 N. From the reaction on the friction wheel R_A (pulling out) and $R_A^{'}$ (pulling in) we count the needed tensile force F_T. The friction wheel rolls onto the square tubing (in fact the wheel has concentrated parameters and that is why the tubing pulls out) resulting in a rolling resistance. Therefore we can count the amount of the traction force (7):

$$F_T = R_A \cdot \frac{\xi}{d_k}, \quad (7)$$

where d_k is the diameter of the friction wheel and ξ is the coefficient of rolling resistance. The diameter of the driven friction wheel is d_k = 80 mm and the rolling resistance arm ξ in the supposed diameter of the friction wheel and material of kinematic pair (steel-steel) we choose ξ = 0.5 mm. By putting the appropriate values into the equation (7) we get the amount of friction force in pulling out the mechanism F_{T1} = 22 N and in pulling in the mechanism the tensile force F_{T2} = 54.28 N. Not to slide the tubing on the surface of the wheel, one condition must be secured, i.e. the static condition of rolling (8):

$$F_T = R_A \cdot \frac{\xi}{d_k}, \quad (8)$$

where $F_{Tt} \geq F_T$ is the tensile force supposed the sliding and f is the coefficient of slide friction, thus (9):

$$R_A \cdot f \geq R_A \cdot \frac{\xi}{d_k}, \quad (9)$$

Resulting in (10):

$$d_k \geq \frac{\xi}{f}, \quad (10)$$

After putting numeric values we get 80 mm > 5 mm and the condition of rolling is fulfilled without sliding and increasing of needed tensile force. Then it is necessary to count the tensile force R_{BT} (11) on the bearing B (pulling out) and C (pulling in), which serve to facilitate movement (there is rolling resistance instead of the slide friction):

$$R_{BT} = R_B \cdot \frac{\xi}{r_L}. \quad (11)$$

When we put real values into the equation (11): r_L = 31 mm the radius of the auxiliary wheel constructed as roller bearing and ξ = 0.005 mm is for rolling resistance arm, so we get R_{BT} = 0.28 N for the tensile force in the bearing B. We can use the same equation for the calculation of the tensile force in the bearing C by means of appropriate values. For the tensile force in the bearing C we get R_{CT} = 0.55 N. The total force needed for pulling out and in of the rack part F_{C1} will be given by the force on the bearing and by the tensile force on the driving friction wheel increased by the loss η in placing (12):

$$F_{C1} = R_{BT} + \frac{F_{T1}}{\eta \cdot \eta}. \quad (12)$$

Placing is carried out by means of two rotary couplings on the roller bearings, each with the medium efficiency η = 0.98 (-). When we put the values into the equation (12) we get the amount of total force in pulling out F_{C1} = 23.19 N and pulling in F_{C2} = 57.01 N. This value needs to be increased six times because the calculation was carried out by means of the sixths model with the sixths load. Then the total force in pulling out is F_1 = 139.15 N and in pulling in F_2 = 342.06 N. These forces are still too great for the arms to be pulled out and in by a hand. That is why we add a chain transmission between the arms and the crank handle. The torsion moment of the friction wheel must be as great as the torsion moment of the big chain wheel (without considering the losses). When we analyses both moments and put the efficiency of the chain wheel into the calculation, in the chain of the chain transmission we get for the force F_{O1} when pulling out and F_{O2}, when pulling in (13a and 13b):

$$F_{O1} = \frac{F_1 \cdot r_k}{\eta_k \cdot r_V}, \quad (13a)$$

$$F_{O2} = \frac{F_2 \cdot r_k}{\eta_k \cdot r_V}. \quad (13b)$$

In the equation (13) r_k is the friction wheel radius r_k = 40 mm, η_k is the efficiency of the chain wheel η_k = 0.94 (-), r_V is the big chain wheel radius r_V = 60 mm, F_{O1} is the force inside the chain while pulling out, F_{O2} is the force inside the chain while pulling in. When we put the values into the equation (13), we get F_{O1} = 98.68 N and F_{O2} = 242.6 N. Then we also consider the moment equilibrium. The torsion moment of the small chain wheel must be equal to the torsion moment of the crank. When we analyse the moments and include all efficiency, we get the amount of force which must be used on the crank for the rack to be pulled out F_1 and pulled in F_2 (14):

$$F_1 = \frac{F_{O1} \cdot r_m}{\eta_k \cdot R}, \quad (14a)$$

$$F_2 = \frac{F_{O2} \cdot r_m}{\eta_k \cdot R}. \quad (13b)$$

In the equation (14) r_m is the radius of the small chain wheel: r_m = 30 mm, η_k is efficiency of the chain wheel η_k = 0.94 (-), R is the radius of the crank and we choose R = 300 mm, F_1 is the force needed for the crank when pulling out the rack, F_2 is the force needed for the crank when pulling in the rack. Putting the values in the equation (14), we get for F_1 = 10.5 N and for F_2 = 25.8 N. So it is obvious that the forces F_1 and F_2 are low enough so that the worker is able to develop them on the crank.

5. Functional calculation of the force on the hand crank

The stress analyses of the storage mechanism were performed in the ADINA software package strength analysis by means of finite element method [5, 6, 10]. As there was described above, the analysed structure is composed of tubes with various cross-sections, therefore in computational model it was created using shell elements. According to known practice we have substitute original model of structure by so-called mid-surfaces with defined thicknesses of material (Fig. 8).

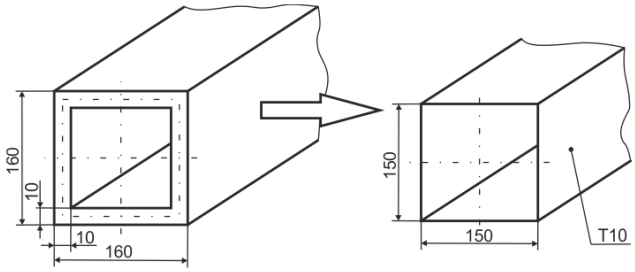


Fig. 8: Releasing of the mechanism for pulling out a) and pulling in b)

Modelling of analysed structure was going on in the Pre-Processing module of the used software while we have modelled sixth model of the main-load bearing part and pull-out arms (Fig. 9).

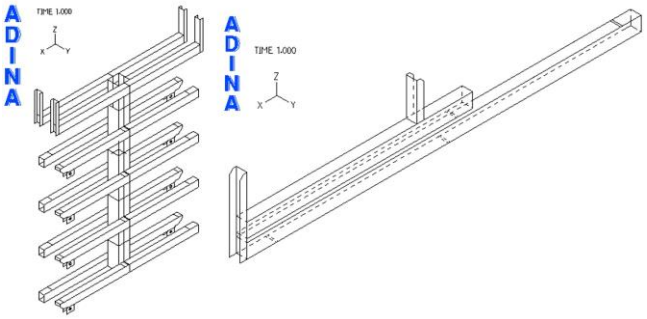


Fig. 9: The geometry of the rack bearing part (left) and the rack arm (right)

We have divided the whole solved task into individual parts, i. e. for two extreme case, when arms are in the pull-in position and when arms are in the pull-out position (Fig. 10). Just because of this we have analysed the storage mechanism in two steps. In the pull-in case arms are loaded by calculated reactions R_A and R_B and in the pull-out case, the reactions $R_A^{'}$ and R_C act on the arm. The structure is made of structural steel with defined material data as following: E = 210 GPa – the Young`s module of tensile elasticity, μ = 0.33 (-) – the Poisson coefficient and $\rho = 7850\ \mathrm{kg \cdot m^{-3}}$ – the density of material. Movement of the arm is ensured by means of the friction wheel. In our computational model this element is represented by the structural node, which is coupled with two lifting holes in the structure, the reaction R_A acting on the friction wheel operates in the centre of holes and it has only a half magnitude. In comparison with previous case, the reaction R_B acting on the upper bearing acts as the distributed load in the contact area of the beam tube and the bearing and the square tubing. In the case of the pull-out position of the arm, the reactions $R_A^{'}$ a R_C are modelled similar to the manner described above.

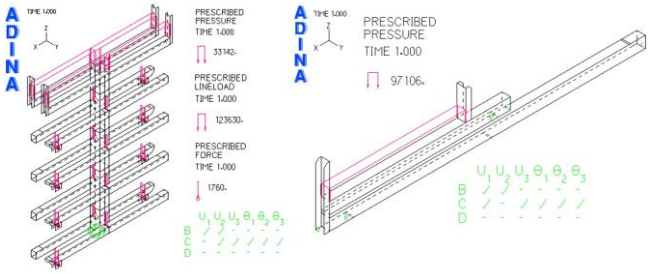


Fig. 10: Boundary conditions of loaded pulled-in rack system

The boundary conditions were defined following: there were removed four degrees of freedom and moving in x and y directions were allowed for the central load-bearing part in the lower rim. Further, we have defined the group of finite elements. As we have used steel profiles with various dimensions, we had to define also shell elements with various thicknesses in such manner, that they would correspond to the real elements dimensions. Also, there was needed to model group of rod elements (Fig. 11 left) which ensure possibility of application of the forces F_1 or F_2 in the proper position of the hole (i. e. in the middle). To ensure the and optimal compromise of the calculation accuracy and calculation consumption time we have set the element size of the FE mesh on 10 mm and as finite elements we have chosen tetrahedrons with linear characteristics.

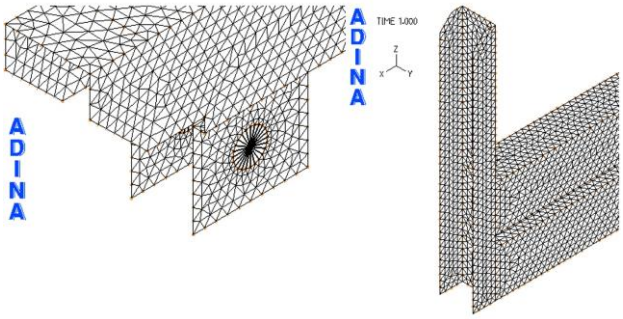


Fig. 11: Detail of the mesh

When we have setting up the entire model and performed all required analyses, we can evaluate results from analyses. It is possible in the Post-Processor module of the ADINA software, where we can display assess several important parameters of an analysed structure, such as stress distribution, displacements, deformations etc. [9].

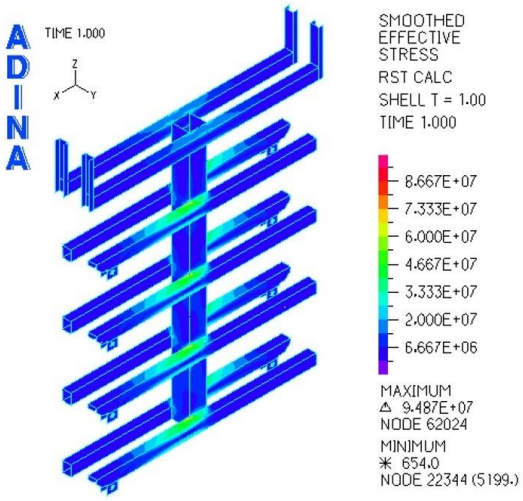


Fig. 12: Smoothed stress in central rack part in pulled-in position

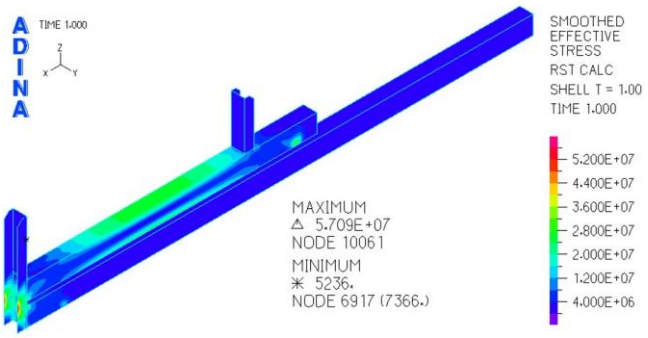


Fig. 13: Smoothed stress in rack arm in pulled-in position

The used software allows setting calculation in such manner, that individual load cases can be calculated in time steps. It means that we have defined two time steps, i. e. for the pull-in position of the arm and for the pull-out position of the arm. Results from analyses are shown in Figure 12 and Figure 13. For strength analysis of the structure we have chosen the HMH hypothesis.

Firstly, we evaluate results from numerical calculation for the first time step (for the pull-in position of the arm). Results are shown in Figure 12. We can see that the maximal HMH stress of 86.7 MPa is identified in the main-load bearing part. The calculated value of the stress in the structure is below the permissible stress of 98 MPa determined for our welded structure. From results follows the design of the storage mechanism for the pulled-in position of the arm meets given requirements. Thereinafter we can assess the calculated stress distribution in the arm structure (Figure 13). The maximal HMH stress in the analysed arm for the pulled-in position achieves value of 57.1 MPa. It is in the boundary condition location therefore we do not take into account it. On closer investigation we can see, that the stress ranges about the value of 52.0 MPa, which means, that the strength of the structure is safely under the permissible stress value (211 MPa).

Let's pass to the second time step of the strength analysis, i.e. when the arm is in the pulled-out position. The calculated stress is distributed in the main-load bearing part structure and in the arm as it is shown in Figure 14 and Figure 15, respectively.

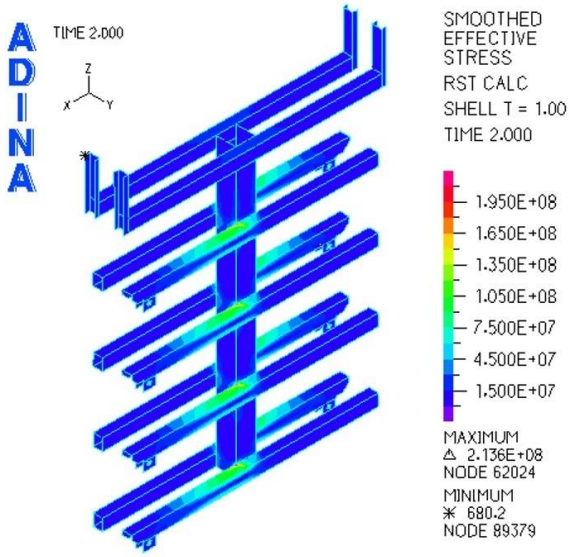


Fig. 14: Smoothed stress in central rack part in pulled-out position

When we compare results of stress analysis, we find out, that for this position of the arm the maximal calculated stress of 214.0 MPa in the central bearing part is higher than the permissible value of 195.0 MPa. The stress distribution in the arm structure for the pulled-out position (Figure 15) shows that the maximal calculated stress is about 50 MPa and it is completely acceptable and it meets the determined criteria.

The strength analysis of the construction showed deficiencies of the design in the pulled-out position. That is why it is necessary to make some changes in the construction. One possible solution to this problem is to weld 10 mm thick sheet between the central and horizontal beams, and thus increase thickness. The geometry of bearing construction model is formed from three separated parts which are joined by means of interconnecting the displacements in each of the corresponding nodes of the mesh (Fig. 16).

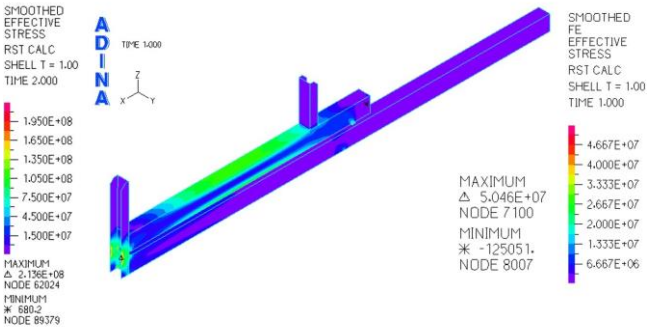


Fig. 15: Smoothed stress and rack arm in pulled-out position

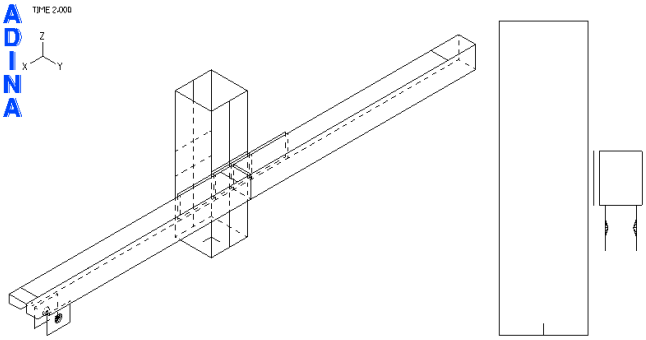


Fig. 16: Geometry of the bearing construction with the welded sheet

We have decided for this method so that the model was more real. If only one mid-surface was modeled with the thickness of the corresponding amount of two beams and the sheet, the results would be less accurate because the sheet is welded to the beams only on the circuit and is not connected with the whole surface.

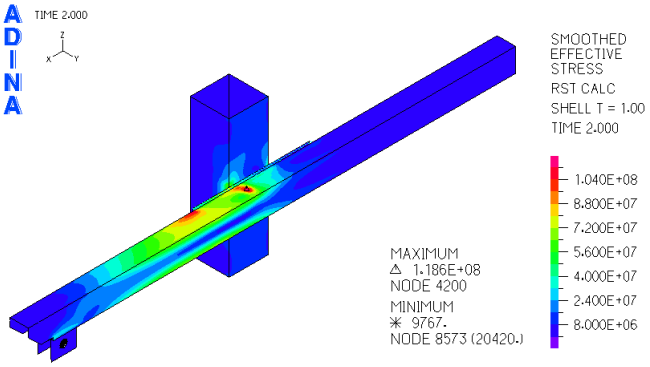


Fig. 17: Detail of stress in the construction after designed change

As it is seen in Figure 17, such construction change can ensure a decrease of the stress in the beams from 195 MPa to 118.6 MPa.

By comparing the obtained stress value with the allowed stress of a weld for the material S355 steel (148 MPa) we conclude that after using this construction change, the maximum stress value will decrease to a safe value, and therefore the construction safely transmits the required load.

The strength analysis was carried out in parts. Finally we can briefly evaluate also deformations of the main-load bearing part of the designed storage mechanism analysed in the case that the arm is in the pulled-out position (Fig. 18.). So the endpoints of the

arms move by 4.18 mm, when arms are in the pulled-out of 2095 mm (Fig. 18).

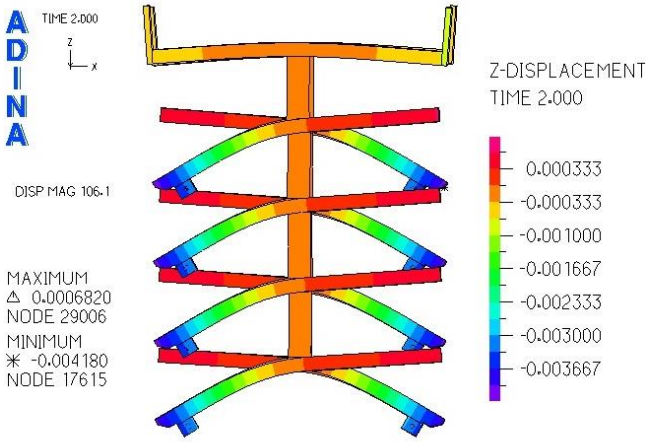


Fig. 18: Simulated displacement of the construction without the construction change in the vertical direction

The construction change included the weld between the central beam and horizontal beams and thus increases the thickness of the construction. This solution proved to be right as the stress values of welds went down to the value of 118.6 MPa. This value is acceptable and the construction meets safety requirements.
In the case of necessity, further investigation of the designed rack system will be focused on setting-up so-called flexible multibody system, which means that our FE model will be imported into the MBS software [3, 4], where will be possible to study dynamic effects load on the structure resulting from the long-term operation conditions [7, 8].

6. Conclusion

This work presented technical solution of the mechanical rack system including strength analyses of its main load-bearing parts. The strength analyses were carried out in parts and we have found out that in the case of pulled-out position of the arm the designed structure does not meet prescribed strength criteria. The original design of the central load-bearing part was not able to carry required load and at the same time the arms are deformed. That is why we have designed the technical modification of the construction. This modification includes the weld between the central beam and horizontal beams and thus increases the thickness of the structure in most loaded locations. Based on numerical calculations we have found out, that our modification ensures stresses reducing on the acceptable value.

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