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Structural Design of a Rotary Valve Manipulator of Bulk Materials–Strength Design of Connecting Elements of the Frame and Trolley

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The paper is a further step in ongoing research on the incorporation of the proposed bulk material rotary valve assembly into an existing production line serving the food industry in bagging milk powder. The main objective of the present paper is the strength static analysis of the previously presented structural design of the trolley and attachment of the frame structure as a track for the travel. Moreover, analytical engineering calculations whose results provide boundary conditions for the numerical strength design of the assembly of the rotary valve for transporting of bulk materials are included in the paper. The proposed mechanism allows precise manipulation of the rotary valve, especially at the time when it is necessary to clean the pipe connected to the rotary valve. Such manipulation is currently actual because of increasing the safety of maintenance of machines and equipment as well as because of reducing the physical burden of maintenance workers. The results of the analyses demonstrate the suitability of the design and provide a basis for further research in this area. The results discovered will be implemented in the form of additional boundary conditions in the numerical analyses of the frame itself carrying the whole travel of the trolley with the rotary valve (the frame forms the track for the trolley travel). The aim of the research is to reach a condition where the entire structure is safe for the operator during maintenance as well as for its surroundings during normal operation.

Keywords: Structural Design, Numerical Analysis, Rotary Valve, Trolley

1 Introduction

Powdered bulk materials are transported by various feasible ways. Bulk materials in powder form are frequently conveyed pneumatically using compressed air. The familiar systems for introducing the bulk material into the conveying line include screw pump, pressure vessel system and rotary valve [1-3]. Rotary valves represent one of the most common means of feeding pneumatic conveying systems, namely, pressure and vacuum types. Rotary valve is a compact mechanical device for continuously discharging bulk powders or granules under gravity flow [4,5]. In addition, rotary feeders are popular devices in industrial production lines. They allow limiting fill the material of transport devices collecting material from under the hopper, bag filters, cyclones [6-8]. The hydraulic rotary valve offers several advantages in comparison with the hydraulic sliding valve, including a simple structure, a high working frequency, tremendous reliability, and a low susceptibility to oil contamination [9,10]. Therefore, it is being increasingly applied to engineering problems [11,12].

A mechanism is a system of bodies connected to each other by linkages and used to transmit forces in line with the transformation of motion. In general, the mechanism is considered as a device which enables the rotary motion of the shaft of a drive motor to be converted into a working motion. In the issue of material handling equipment, the working motion may be represented, for example, the lifting and lowering of the load, the tipping or telescopic extension of the boom, the travel of an end truck of the crane as a whole or as a part of the crane (crane trolley motion) as well as slewing in the case of slewing jib cranes [13-15]. The frame is the basic load-bearing part of any conveyor, and its main task is to ensure the exact position of the individual components and not to damage them. The frame must ensure the smooth operation of the vehicle with regard to a high degree of safety, and the design of the frame must take into account its aesthetics and economy [16]. In terms of entire design process of the mechanism, the FMEA (Failure Mode and Effect Analysis) methodology can be employed as one of the risk analysis techniques recommended by international standards to identify possible failure causes and to reduce their consequences in order that the intended function of the machine is fulfilled [17,18].

In terms of the proposed manipulator, a screw mechanism is employed to change the rotary movement into a translational motion. The mechanism in question is utilized to transfer energy and allows the manipulation of the rotary valve of bulk materials [19,20]. Manipulation is required at the time when it is necessary to clean the pipes connected to the rotary valve. Therefore, it is indispensable to position the rotary valve. The instantaneous position of the general mechanism is uniquely determined by the number of independent coordinates equal to degrees of freedom of the mechanism. In other words, the mechanism has as many degrees of freedom as independent coordinates are demanded to uniquely determine its instantaneous position [21-24].

The proposed structural unit (Fig. 1) is intended for implementation in a milk powder packaging production line. Inasmuch as the structure must not hinder the operator, the anchorage of the structure in the ceiling at a height of 4.2 meters above the floor is chosen. Thus, the screw manipulator mechanism is designed for two extreme positions (operating and service) in order to increase efficiency and reduce the physical burden of the workers in the course of maintenance and repairs.

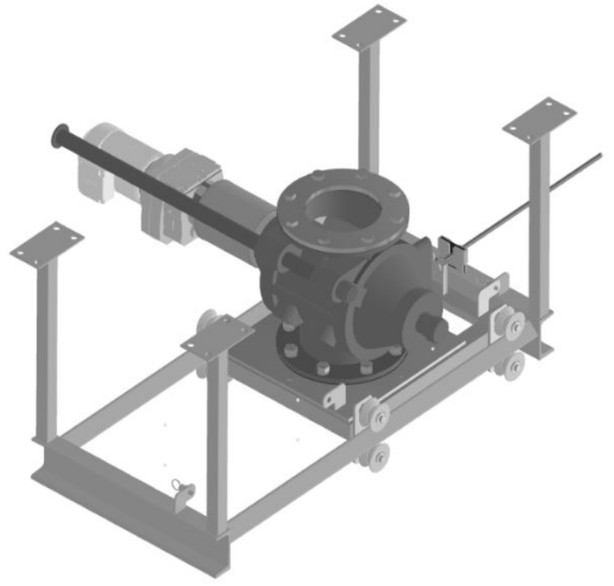

Fig. 1 *A 3D CAD model of the rotary valve for bulk materials with the designed components: trolley, anchoring frame and screw manipulator*

Considering all the input mass parameters described in paper [25] and the dimensional parameters introduced in the paper [26], a structural design of a manually operated rotary valve manipulator of bulk materials was developed by authors [27]. The manual drive appears to be the optimum solution due to the food processing operation (and the resulting high demands on cleanliness) [28], the mass of the rotary valve itself in conjunction with its drive of approximately 230 kg, as well as the economy of the design itself. With regard to the relatively low passive resistance of the rotary valve bed (trolley) as it moves along the track, it will be manipulated by means of turning provided by an operator (wrench, cordless screwdriver, etc.). Authors in the paper [25] performed a determination of the rotary valve centers of gravity in order to quantify the load on the traveling wheels of the trolley. Accordingly, these loads were utilized in the work [26] to calculate the bending moments of the support structure (track) of the travel at the determined most effective load position. Moreover, the authors dealt with the calculation of the resistances limiting the movement of the trolley. These data were employed in the structural design of the detachable rotary valve screw manipulator [26]. In addition, the manipulator needs to be addressed from a computational point of view in order to meet all the demands placed on it. The calculation of mechanisms generally consists of functional and strength calculations. The functional calculation discovers the required power of the driving motors, the kinematic quantities of the machine and its individual elements in order that the desired operating speeds, or start-up times, as well as the trajectories of the individual working motions can be determined. In terms of the strength calculation, the dimensions of the individual parts are determined. Accordingly, the paper in question deals with the issue of strength calculation.

2 Analytical strength design of frame connection elements

As mentioned earlier as well as resulting from Fig. 1, the structure forming the travel (trolley) track will be attached to the ceiling of the room. The arresting of the structure is assumed by means of threaded rods (a total of 16 pieces) threaded into the upper floor of the building and secured by means of two nuts. One of the nuts will fix the threaded rod in the concrete structure of the ceiling (equivalent to the head of a bolt) and the other nut will fix the structure. Because of the threaded rods will be subjected primarily to static pressure in the operation, it is necessary to verify their strength. Furthermore, it is essential to scrutinize the value of the pressure in the thread at the nut-bolt contact. In order to be able to check the individual dimensions at all, the maximum stresses must be determined. These stresses result from the longitudinal position of the center of gravity of the trolley with the rotary valve [25,26].

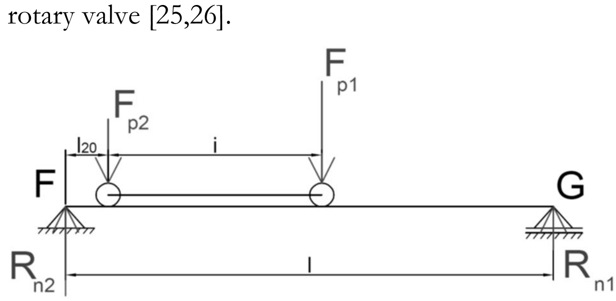

Fig. 2 *Schematic representation of the trolley in its most unfavorable position for achieving the maximum load on the connecting members*

On the basis of the position of the center of gravity, the most loaded beam and the resulting single bolt load will be determined. With regard to the above-mentioned factors, a suitable threaded rod and nut can be designed. First, the moment equilibrium condition of the travel was assumed as shown schematically in Fig. 2, where *F* and *G* symbolize the beam supports (in Fig. 1, vertical square bars welded to the flanges attached to the ceiling by means of threaded rods).

$$\sum M_{iG} = 0 : F_{p1} \cdot \left(l - i - l_{20}\right) + F_{p2} \cdot \left(l - l_{20}\right) - R_{n2} \cdot l = 0,$$
$$R_{n2} = \frac{F_{p1} \cdot \left(l - i - l_{20}\right) + F_{p2} \cdot \left(l - l_{20}\right)}{l} \left[N\right], \tag{1}$$

Where:

i… The wheelbase of the trolley [mm],

l_{20}… The horizontal distance of the center of wheel no. 2 of the travel from the support *F* [mm],

l… Length of the travel track [mm],

F_{p1}… Vertical wheel force of the right wheel on the more loaded side of the structure – the result taken from paper by Blatnický et al. (2023) [N],

F_{p2}… Vertical wheel force of the left wheel on the more loaded side of the structure – result taken from paper by Blatnický et al. (2023) [N],

R_{n1}… Sought reaction (axial force) of the vertical beam no. 1 [N],

R_{n2}… Sought reaction (axial force) of vertical beam no. 2 [N].

After substituting the values into equation (1), the reaction $R_{n2} \approx 1330$ N is obtained.

$$R_{n2} = \frac{1302.038 \cdot \left(1225 - 525 - 93\right) + 738.45 \cdot \left(1225 - 93\right)}{1225} = 1327.56\,N. \tag{2}$$

The value of R_{n1} can be discovered from the vertical equilibrium condition (3):

$$\sum F_{iy} = 0 : R_{n1} - F_{p2} - F_{p1} + R_{n2} = 0 \left[N\right]. \tag{3}$$

By solving equation (3), the reaction $R_{n1} \approx 710$ N is obtained. The highest axial force in the beam with the value of 1330 N emerged from the equation (1). This case occurs when the trolley is moved to the extreme position (closest to the support) with a more loaded wheel. If the use of flanges with symmetrically placed four bolt holes is considered (Fig. 1), the axial force in one threaded rod can be determined as a quarter of the original load, i.e., 332.5 N. On the basis of these results, a formula for sizing the cross-section of the core of the threaded rod can be established. The selected material of the threaded rods is X5CrNi18-10. Therefore, the allowable static tensile stress is selected with the value $\sigma_{D1} = 180$ MPa. The core area of the threaded rod will be subsequently determined by dint of formula (4):

$$\sigma_{D1} \geq \frac{\frac{R_{n2}}{4}}{S_j} \left[MPa\right], \tag{4}$$

Where:

σ_{D1}… Allowable static tensile stress [MPa],

R_{n2}… Sought reaction (axial force) of vertical beam no. 2 [N],

S_j… Core area of the threaded rod [mm²].

The proposed core area of the threaded rod depends on the the core diameter d_j, for which the formula (5) is applied:

$$d_j \geq \sqrt{\frac{R_{n2}}{\pi \cdot \sigma_{D1}}} \left[mm\right], \tag{5}$$

Where:

d_j… Core diameter of the threaded rod [mm],

R_{n2}… Sought reaction (axial force) of vertical beam no. 2 [N],

σ_{D1}… Allowable static tensile stress [MPa].

With regard to the production of only a few pieces, safety and proportional aesthetics, the M16 threaded rod with core diameter $d_j = 13.3$ mm, pitch $s = 2$ mm and allowable thread pressure $p_D = 100$ MPa was chosen. By selecting this structural unit, it was further possible to carry out the calculation (6) aimed at determining the pressure in threads p_{z}. Each nut contains $p = 6$ threads, provided that nuts with height $h = 12$ mm are utilized.

$$p_z = \frac{\frac{R_{n2}}{4}}{\frac{\pi}{4} \cdot \left(d^2 - d_j^2\right) \cdot p} \leq p_D \left[MPa\right], \tag{6}$$

Where:

p_z…Pressure in threads [MPa],

R_{n2}… Sought reaction (axial force) of vertical beam no. 2 [N],

d…Thread diameter [mm],

d_j… Core diameter of the threaded rod [mm],

p…number of threads of nut [-],
p_D… allowable thread pressure [MPa].

By substituting the values into formula (6), it is evident that the pressure in threads achieved in operation is significantly less than the allowable pressure $p_z << p_D$. Inasmuch as the trolley is equipped with wheels that rotate on pins fixed to the trolley structure (Fig. 3), the threaded rods need to be checked for shear stress (7). The bolt material has a permissible shear stress with a value of $\tau_D = 135$ MPa. The resistive force of the trolley $F_o = 307.07$ N was taken from paper Molnár et al. (2023).

$$\tau_s = \frac{4 \cdot F_o}{\pi \cdot d_j^2} \le \tau_D \left[MPa\right], \tag{7}$$

Where:
τ_s…Shear stress of threaded rod [MPa],
F_o… Resistive force of the trolley [N],
d_j… Core diameter of the threaded rod [mm],
τ_D… Permissible shear stress [MPa].

After substituting values into formula (7), the shear stress of threaded rod is (8):

$$\tau_s = \frac{4 \cdot 307.07}{\pi \cdot 13.3^2} \le 135 \Rightarrow \tau_s = 2.21 MPa \le 135 MPa. \tag{8}$$

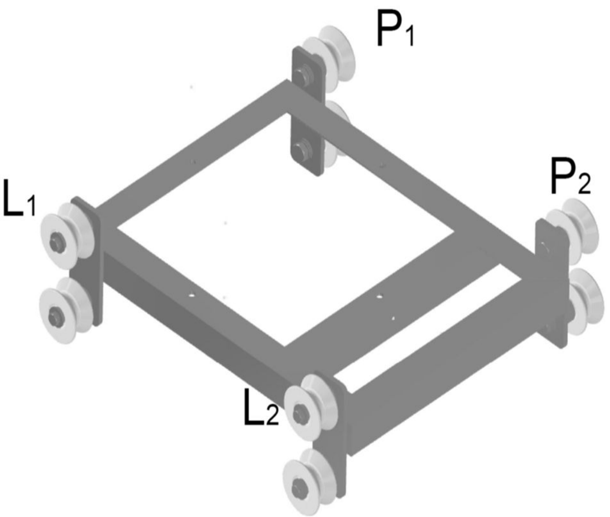

Fig. 3 *A 3D CAD model of the trolley with individual wheel markings*

The results indicate that even when purely one threaded rod is utilized, it is able to carry the shear load with great safety. Nevertheless, shear will purely occur if the nuts are loosened. In terms of tightened nuts, the load will be transmitted by friction. In this case, increased stress on the threaded rods is not expected. Thus, there is no risk of damage in the course of trolley operation.

3 Numerical analysis of the trolley using FEM

The trolley model including all elements (Fig. 3) was imported from Autodesk Inventor CAD (computer-aided design) program into the Ansys program (Fig. 4). The Ansys software is widely use software operating based on the finite element method [29-31]. Connection elements and other (irrelevant for the simulation of the trolley frame) components were removed in order to reduce the number of equations and therefore hasten the calculation (Fig. 5).

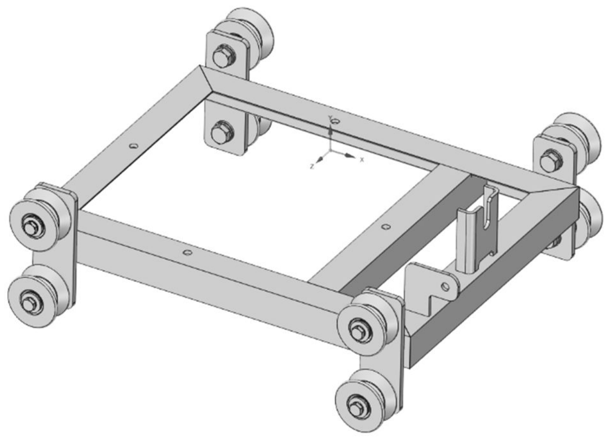

Fig. 4 *3D model of the trolley after importing into the SpaceClaim working environment of ANSYS*

After the aforementioned modifications, the fillets were further removed from the model. This modification is essential due to the quality of the mesh and the reduction of the computation time (Fig. 6). Despite the fact that this step may create areas with stress concentrations, it will refine the global calculations due to the increase in mesh quality. Therefore, the resultant negative effect of this step is negligible.

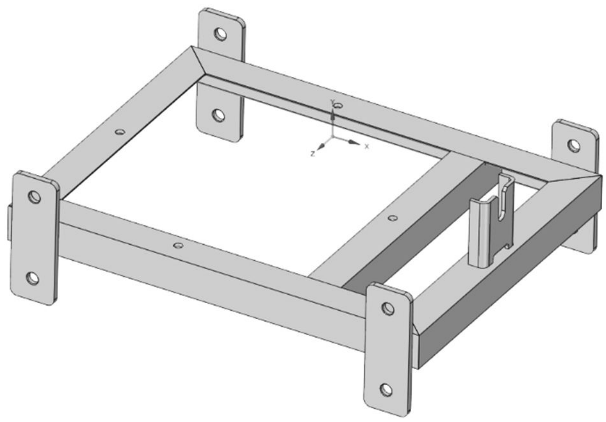

Fig. 5 *Model of the trolley after removal of geometry insignificant for further simulations*

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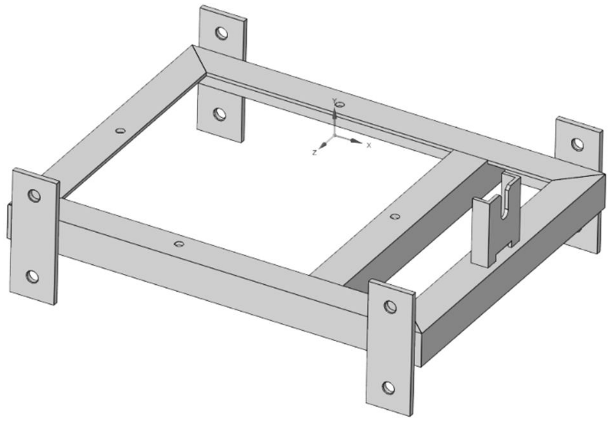

Fig. 6 Model of the trolley after removing the fillets

Due to the entire model is made of thin-walled profiles, it is both possible and highly advisable to create midsurfaces of these profiles. Further, the geometry has been modified in order that it can be combined into a single unit. In this step, it is primarily about the displacement of some parts and their interconnection (Fig. 7). In other words, the midsurfaces are extended or trimmed to adjacent faces. The output of the ANSYS SpaceClaim is depicted in Fig. 8.

Fig. 7 Model imperfections arising in the creation of the midsurfaces

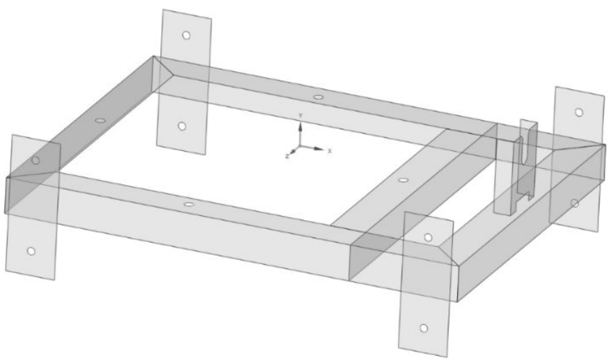

Fig. 8 Trolley frame model after all modifications made in ANSYS SpaceClaim

After modifying the geometry of the model and preparing it for further processing within the ANSYS SpaceClaim working environment, creation of the simulation in the ANSYS Mechanical environment is proceeded. In this environment, a mesh was applied to the midsurfaces, modified in terms of the size and shape of the elements, which are employed for meshing the model. A detail of the mesh can be seen in Fig. 9.

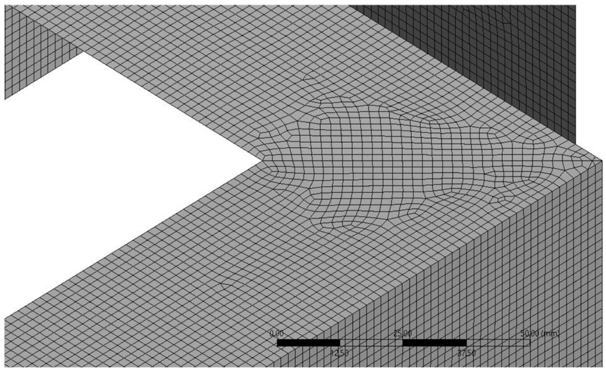

Fig. 9 Detail of the trolley mesh model at the point of contact of the two L profiles at an angle of 45°

The detail of mesh clearly demonstrates that the individual nodal points are contiguous and therefore there is no risk of negatively affecting the simulation in terms of discontinuities in the mesh. In this case, an element size of 2 mm was utilized along with the meshing method using quadrilateral elements. Consequently, boundary conditions in the form of fixed support to holes for the threaded rods, in the form of forces at the locations where the force from the screw manipulator will be transmitted and a force equal to the rotary valve weight at the location of its center of gravity were applied. The above-mentioned boundary conditions are shown in Fig. 10.

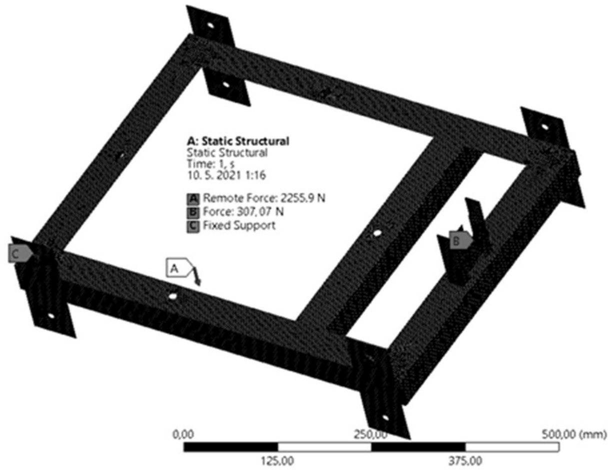

Fig. 10 Selected boundary conditions of the computation

As with most similar projects focusing on framed steel structures, it is essential to determine the displayed quantities in the form of von Mises stress and displacement, i.e., structural deformation. Figure 11 demonstrates the overall stress distribution along with the minimum and maximum stresses marked. Figure 11 evidently shows that the maximum stresses do not reach the yield strength of the EN X5CrNi18-10 steel utilized, namely, approximately 180 MPa. In spite of this fact, the location with maximum stress was scrutinized in more detail (Fig. 12).

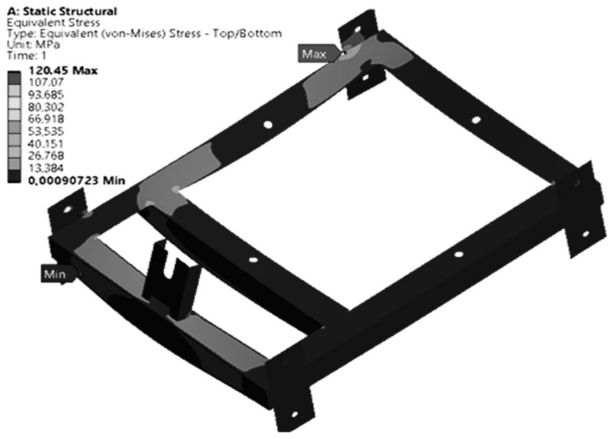

Fig. 11 *Simulated von Mises stresses of the analyzed trolley design*

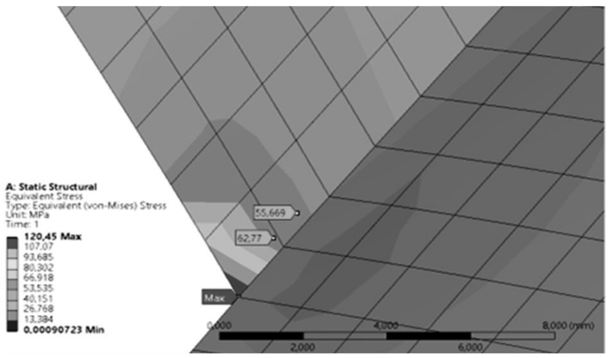

Fig. 12 *Detail of the location where the greatest equivalent stress is discovered in the trolley structure*

Figure 11 and Fig. 12 demonstrate that the highest stress is reached at a point where the L profile of the trolley frame joins the sheet metal bracket holding the wheel. In this case, due to the spot occurrence and the extremely rapid decrease of the stress depending on the distance from this point (merely 2 mm from the point in question the stress drops to approximately 60 MPa), it can be stated that this is a so-called singularity resulting from the geometry of the body and the mathematical solution of the problem in question. Therefore, it is relevant to consider a stress of approximately 60 MPa as an actual value. In addition, the structure can be considered as a safe structural design, due to the safety factor k of the achieved von Mises stress to the yield strength $k = 3$ (-).

Figure 13 demonstrates that the deformations of the trolley (also relative to its dimensions) are considerably diminutive. Moreover, deformations are not expected to cause problems of any kind in operation. On the basis of the forces applied and the results obtained, it is possible to evaluate the design of the trolley as adequately rigid.

The future reseaerch will be focsued on the analysis of the menaipulator form the dynamics point of view. The oscillation of the structure will be analysed. Although it is assumed, that the designed strucutre is suitable for operation and any negative dynamic effects are not expected, the simulation tools (multibody software) allows to perform such analyses [32-34].

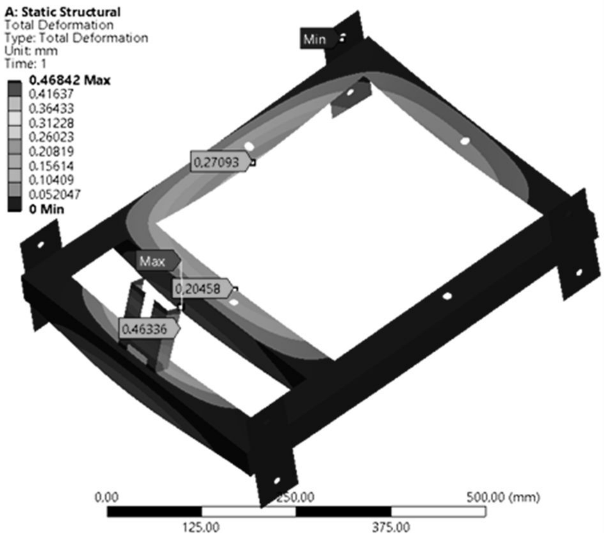

Fig. 13 *Distribution of simulated displacements detected on the trolley structure in operation*

4 Conclusion

The paper is part of the ongoing research concerning the integration of the proposed bulk material rotary valve assembly into the existing production line in a real plant for bagging milk powder. Its chief objective was the strength static analysis of the trolley and the attachment of the frame structure as a track for the travel. In addition, analytical engineering calculations accompanying the research were presented. The initial practical part of the paper focused on the analytical strength design of the frame connection elements. The structure forming the travel (trolley) track will be attached to the ceiling of the room. The fixation is performed by means of threaded rods and nuts. In this part of the paper, the M16 threaded rods were designed through calculations as well as a check of the pressure in the threads was carried out. The obtained results demonstrate that the achieved thread pressure in operation is significantly less than the allowable pressure. In the further part of the paper, a numerical analysis of the trolley was carried out using FEM in Ansys. The arising von Mises stresses and displacements were investigated. The maximum stresses do not reach the yield strength of the EN X5CrNi18-10 steel utilized. The deformations of the trolley (also with respect to its dimensions) are considerably diminutive. Furthermore, they are not expected to cause problems of any kind in service. The results of the analyses proved the suitability of the structural design and will form the basis for further research in this area. The results found will be implemented in the form of necessary boundary conditions in the numerical analyses of the frame itself carrying the whole travel of the trolley with the rotary valve. The aim of the research will be to reach a condition where the whole structure is safe for the operator during maintenance as well as for its surroundings in the course of normal operation.

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