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# Optimization and verifying designed parts for hevy loads

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**Abstract** This work focuses on the design of a rotating station for weighty and bulky products. It was a combination of diverse requirements based on practice, technical and design requirements and appropriate real design possibilities. Verification of technical designs and safety requirements was carried out using a CAD Solidworks simulation, which was used for the strength calculation. By checking the structural design using the finite element method, it has been found that we need to optimize the stiffeners of the supporting elements. By optimizing, it has been achieved the weight reduction of the station and the stiffness increase at critical design points.

**Keywords** optimization, simulation, FEM, CAD

## 1. INTRODUCTION

The subject of the study was the design, a construction and a subsequent manufacture of the station for rotating heavy parts. Design check should be part of the development of each product. By checking, we can ensure its reliability and security. In mass production, also material optimization and saving are considered.

Checking was done using the Finite Element Method (FEM). This method verified stress strength values and displacement of components.

Solidworks simulation allows us to load the model with linear statics, in which we can load the model with forces, torques (moments), deformations and temperature field. In linear statics, material behavior is linear and described by Hook's law.

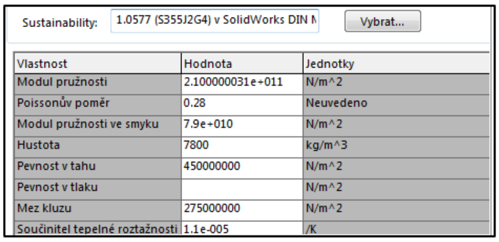
### 1.1 FEM principle

The principle of the finite element method consists in dividing a component into small parts – elements. These elements must be small enough to best describe the given component. The smaller and regular elements will be, the more accurate the calculation. At the corners of these elements, there arise nodes that determine the shape and position of the element. Partial solutions on these elements are combined into one unit after the calculation. For bulk elements, these elements have the shape of a block or a quadrangle. The recommended element size setting for bulk bodies is at least two elements per element thickness.

### 1.2 Setting

Safety factor: It is used for a simple expression of design reliability. The safety factor is chosen according to the used material and the field of application of the product. For a normal construction of standard steel machines, the safety factor k = 1.5 – 2 is chosen.

Material for the manufacture of the station was chosen from commonly available materials for thick rolled sheets – S355J2.


Sustainability: 1.0577 (S355J2G4) v SolidWorks DIN N
Vybrat...

| Vlastnost | Hodnota | Jednotky |
|---|---|---|
| Modul pružnosti | 2.100000031e+011 | N/m^2 |
| Poissonův poměr | 0.28 | Neuvedeno |
| Modul pružnosti ve smyku | 7.9e+010 | N/m^2 |
| Hustota | 7800 | kg/m^3 |
| Pevnost v tahu | 450000000 | N/m^2 |
| Pevnost v tlaku | | N/m^2 |
| Mez kluzu | 275000000 | N/m^2 |
| Součinitel tepelné roztažnosti | 1.1e-005 | /K |

*Fig. 1: Properties of the material from the library Solidworks*

### 1.3 Description of the task

The technical solution was based on practical requirements for the rotating station. The basic limitations included the installation dimension of the station, especially the built-up height of the station, which should not exceed 300 mm. Weight and load distribution in the horizontal plane. The load of the station up to a workpiece weight of 15,000 kg. The rotation of the station work plate must be manual or performed by manual drive. The adjusting feet at the corners of the station.

The construction design of the rotating station was investigated as a low plate table with a work plate rotatably mounted on a large-diameter turn element „bearing“ with external teeth. Due to a high weight of the parts being handled (high passive resistances and high dynamic torque needed to rotate large masses), the work plate of the station was rotated by a gear mechanism (a circular turn element with external teeth – pinion – worm gear – crank for manual rotation of the station).

The station is equipped with a wooden grate to prevent product damage.

The products can be laid on the station with a straight surface (plate) or on the auxiliary conical struts with a diameter of the feet 200 mm.

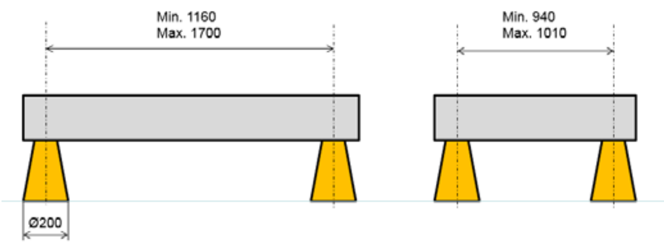



*Fig. 2: Load distribution on the rotating station*

## 2. CONCEPTS OF THE ROTATING STATION

The original construction design was based on the initial rough requirements, where after the first idea, a study of a rotating table consisting of two plates was created. A large-diameter bearing was inserted between the plates. The feet of the station were made of solid flat bars where it was assumed to be used on a planar base.

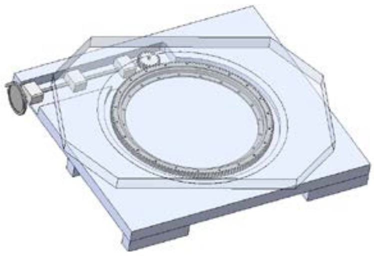


*Fig. 3: Basic design of the station*

In the next stage of the development, observations were implemented and the design development was being completed. First, it was necessary to design and select a central bearing with sufficient load capacity. For bearing design, we need to know the basic static load that will be stressing the bearing. This load will reflect the weight of the parts being handled and the weight of the upper loading plate with the accessories. Determination of equivalent static axial load and equivalent static torque.

$$F_{0q} \mathrel{\widehat{=}} F_{0a} * f_A * f_S$$
$$M_{0q} \mathrel{\widehat{=}} M_{0k} * f_A * f_S$$

Application factor fA (fA = 1.5) and Additional safety factor fS (fS = 1) according to the manufacturer of the bearing.

$$F_{0a} = (m_{dílu} + m_{desky}) * g = (15 + 3) * 9{,}81 = 1$$

The calculation of the tilting torque is based on the load from the parts being handled and their maximum possible offset when laying on the loading surface (upper plate). It follows from the calculation that a higher tilting torque occurs when the whole part being handled is eccentrically laid rather than in a non-symmetrical suspension of the part and laying the part over the edge of the upper plate. Center of gravity offset l = 0.45 m.

$$M_{0q} = (m_{dílu} * g * l) * f_A * f_S = (15 * 9{,}81 * 0{,}45) * 1{,}5 * 1 \cong 99{,}5\ KNm$$

Eccentric load parameter ε = 1.

From F0q and M0q values, the load point and the static load capacity check of the fastening bolts were determined. Due to a possible overloading of the bearing owing to poor handling and an offset of the product being laid, I chose an INA VSA200944N four point contact ball bearing, which is shown in the Graph under the number of 6.

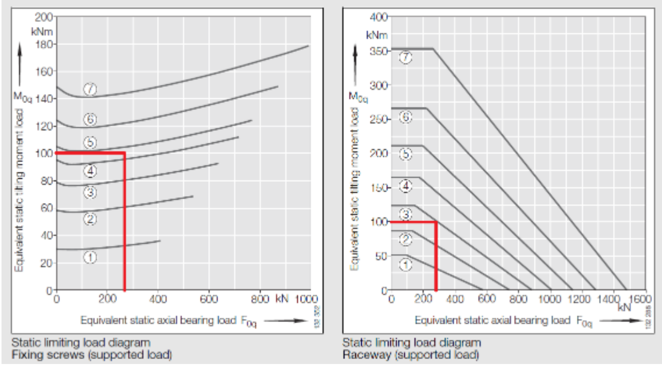



*Fig. 4: Load diagram of the fastening bolts [3].*

In the next step, it was necessary to provide a manual drive design of the station, which was designed as follows: circular turn element with external teeth – pinion – warm gearbox – crank for manual rotation of the station. It was necessary to ensure a reasonable ratio of speed by the hand crank and the deduced force required to operate the station.

Determining the forces needed to overcome the passive resistances and acceleration:
The moment of inertia of the offset part being handled (Iz = 9322 kg.m2) to the axis of the rotating station was obtained from a 3D model. The rolling resistance coefficient of the bearing is μ = 0.01.

Friction force Ft

$$F_t = (m_{dílu} + m_{desky}) * g * \mu = (15000 + 3000) * 9{,}81 * 0{,}01 = 1766\ N$$

The speed of rotation of the station is required for 2 rpm, resulting in φ = 4π [rad]. The angular velocity of the rotating station: gear ratio for pinion $i_1$ = 5.86 and gearbox $i_2$ = 40.

$$a_\omega = \frac{\varphi}{i_1 * i_2} = \frac{4\pi}{5{,}86 * 40}$$

Acceleration resistance of rotational masses

$$M_s = I_Z * a_\omega \cong 536\ Nm$$

The bearing rotation diameter is di = 945.5 mm
The torque of resistance we have to overcome on the pitch circle of the bearing external teeth is:

$$M_o = M_T + M_s = F_t * \frac{di}{2} + M_s \cong 1358Nm$$

From the calculated values of the resistances we have to overcome when rotating the station and from the required values of the forces needed for the manual drive of the station, other necessary components, such as pinion gearing and worm gear dimensioning, were designed. The calculation showed the need to use a higher gear series – other built-up dimensions than those in the original design. The number of teeth of the pinion was set at z = 20, worm gear of a CMIS075 series with a gear ratio of i2 = 40, an output nominal torque 25 Nm and an efficiency of 44%.

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From these values, the theoretical force needed to manually rotate the station resulted in 63N.

## 3. OPTIMIZATION OF THE ROTATING STATION

An analysis using simulation begins with geometry represented by a model. This geometry must be converted to a set of appropriate large elements. This networking (crosslinking) requirement is important for the analysis results. We simplify the model with the sole goal of supporting the system and shortening the calculation time. Simplification must not be too aggressive to distort the simulation results.

The simplification of our model was accomplished by removing cosmetic threads, bolt (screw) holes, cosmetic chamfers and radii.

After adjusting the geometry, it is necessary to define the boundary conditions – material values, catches, load of a part, type of analysis. A parabolic tetrahedral was used to define the basic element. For our task, the value of Jacobi's method 4 was chosen.

### 3.1 Optimization of the loading plate

The design of the loading plate was made of a thick-walled sheet metal in a thickness of 100mm. The optimization of the plate consisted in adjusting its thickness and recessing the seating surface of the bearing. To simplify networking (crosslinking), we removed the cosmetic threads on the upper surface serving to hold (catch) the wooden grate and the bolt holes for attachment to the bearing that were replaced by the corresponding partition curve of the seating surface. The load of the plate was used to asymmetrically lay the load onto four conical struts (webs).

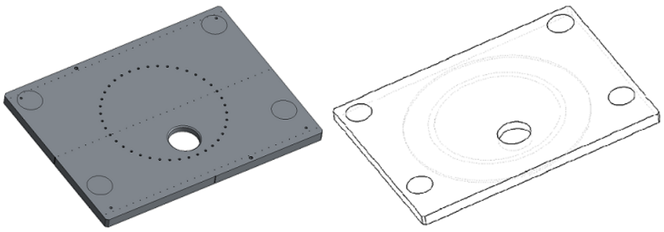

*Fig. 5: Displaying the loading plate before and after simplification*

**Boundary conditions and calculation:**
An attachment (catch) as a fixed geometry was used for the partition curve (spline) of the seating surface of the bearing. A load of 150,000N was applied to the cylindrical seating surfaces.

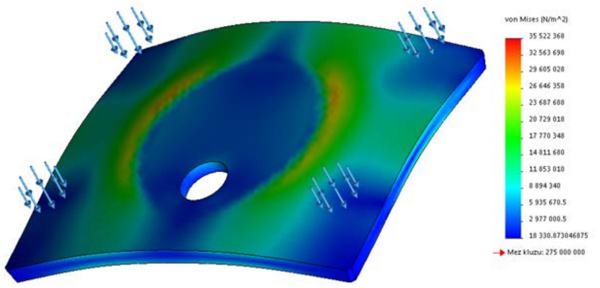

*Fig. 6: Stress course in the loading plate - von Mises*

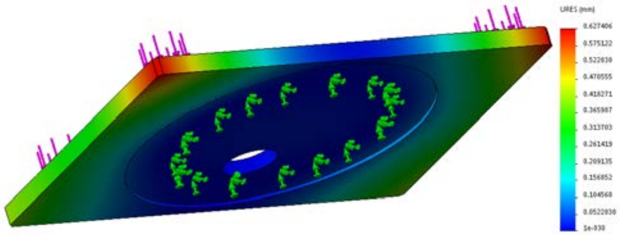

*Fig. 7: Displaying the deformation by displacement in the loading plate*

From the results of numerical simulations, the thickness reduction of the loading plate by 16 mm and weight reduction by 370 kg were carried out.

### 3.2 Optimization of the base frame of the station

Modifying the original design:
Fixed feet were replaced for adjustable feet used only at the corners of the station. The position of the worm gearbox was adjusted due to the larger built-up dimensions and the associated connecting parts (bearing houses and drive shaft) that had to be recessed under the base plate. A hole was created through the plate for an easy removal or replacement of the pinion on the worm gearbox.

Simplification of our model was accomplished by removing cosmetic threads, bolt (screw) holes, cosmetic chamfers (chunks) and fillets (rounding). There were also suppressed recesses for stops and seating surfaces which are at the edges of the frame and do not affect its strength.

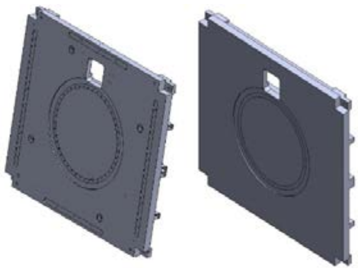

*Fig. 8: Displaying the base frame before and after simplification*

**Boundary conditions and calculation:**
The seating surface of the bearing was loaded with a force of 180,000 N. The base frame was attached (caught) by a roller/slider fixture (restraint) that was applied to the surfaces at the adjusting feet. One threaded foot hole was set as fixed to catch the reaction forces.

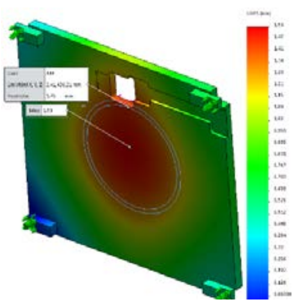

*Fig. 9: Initial checking calculation*

After a checking calculation, an undue deformation under the bearing was found. The resulting displacement value was 1.45 mm.

The measured stress values exceed the material yield strength. Those peak values occur at sharp edges. In practice, such sharp edges would not occur or would be replaced by a weld in the given point.

The permitted deformation under the bearing according to the manufacturer is 0.145 mm.

Therefore, the longitudinal stiffeners under the base plate were used. For a faster calculation (element networking), a full (solid) profile of (the bars) the stiffeners was chosen.

**Modifying the base frame:** adding longitudinal stiffeners. The boundary conditions remained the same.

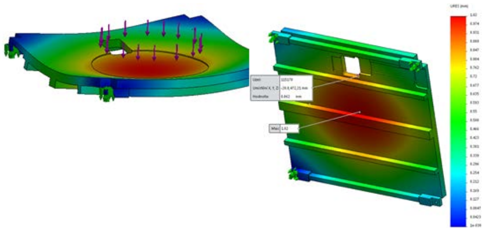

*Fig. 10: An example of deformation of the part under load*

When using longitudinal stiffeners, the resulting displacement values dropped to a displacement value of about 1mm. From the example of the deformed part, there is evident a collapse of the stiffeners in the transverse direction in the area of the opening for placing the gearbox.

Another approach was to add transverse stiffeners to prevent deformation in the area of the opening.

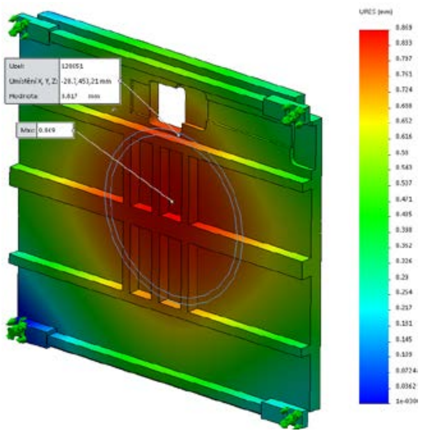

*Fig. 11: Deformed frame with transverse stiffeners*

After this adjustment, the values of deformation – displacement improved, but the values are still above the serviceability limit set by the manufacturer of the bearing.

Further reinforcing using stiffeners would not make any sense in terms of both increasing the stiffness and the economic point of view where another adding of stiffeners did not bring the desired effect.

Therefore, another approach was a modifying of the basic structure of the station with a design of a central fixed support and the principle of the so-called balance table. This support would significantly increase the stiffness of the table in the critically stressed area. Another advantage of this solution is the distribution of the total weight of the station to a higher number of contact points.

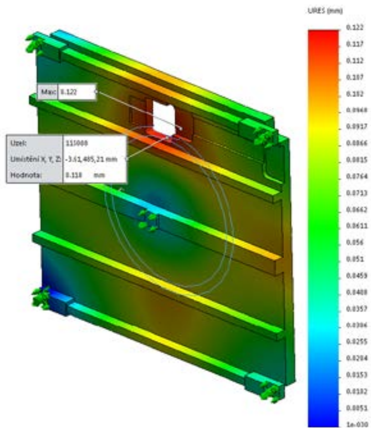

*Fig. 12: The displacement course in the modified frame structure*

From the results it is concluded a rapid improvement in the resulting displacement when we reduced the deformation of the base frame by one order to a value of 0.12 mm.

The final optimization of the model took place in reducing the weight of the stiffeners when the rod profiles were replaced by a conventional tubular rectangular profile with a wall thickness of 5 mm.

Example of the final reinforcement of the base frame (plate) using stiffeners from tubular profiles.

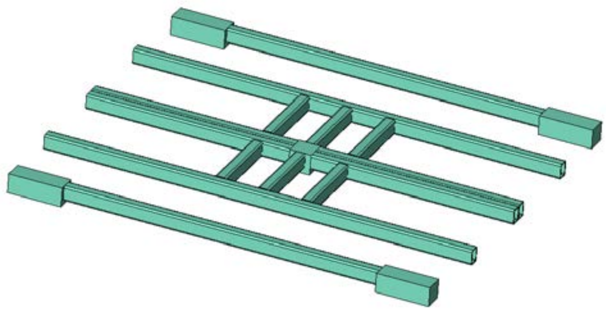

*Fig. 13: Example of the base frame stiffeners*

In the final calculation, those values were set: The network type was retained from the previous analyses – volume network, the Jacobi method at value 4. The model attachment: the seating surface and the surfaces at the adjusting feet were set as roller/slider, avoiding Z-axis deformation. One threaded feet hole was set as fixed. The load was induced by a force of 180,000N perpendicular to the distribution curve representing the bearing seating surface.

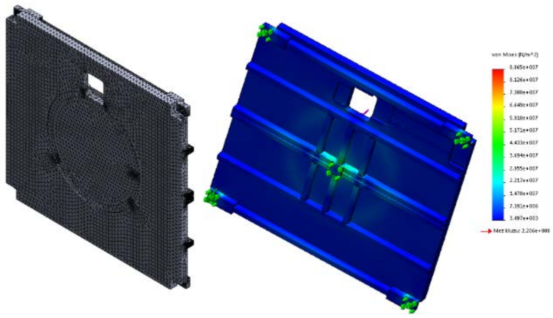

*Fig. 14: An example of network - elements creating on the left and stress course on the right*

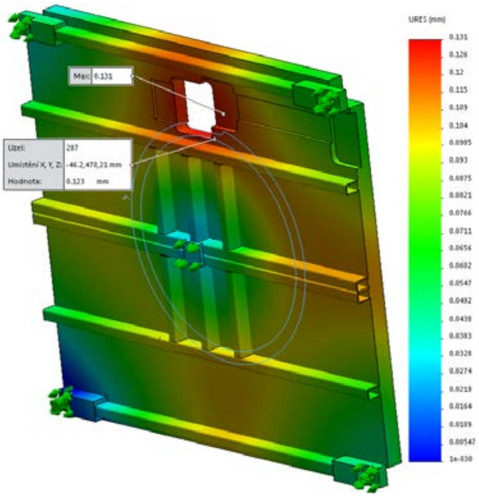

*Fig. 15: Displacement course in the optimized structure*

**Results**

From the resulted values it is concluded that we have achieved an acceptable deformation under the bearing using a new design concept and an optimization of the stiffeners. The resulting displacement value is about 0.132 mm under the bearing and the maximum value is 0.131 mm. By replacing the solid stiffeners for the tubular profile, it occurred a decrease of the weight of the base frame by approx. 255 kg.

## 4. CONCLUSION

The design was verified in a Solidworks simulation environment shows a simple design of a component and its check using the finite element method. It is a static analysis that provides quick and accurate results. Using this software, the designer can analyze the static structural behavior of the engineered parts and quickly respond and modify the parts. If the parts are stressed with a simple load only, so the simulation program saves the time needed for thorough calculation and modeling. In its result, the carried out simulation solved by the designer himself or herself examines the basic optimization both in terms of weight of the used material and it mainly leads the designer to more suitable modifications. Design modifications are in the designer's hands, are verified with mathematical simulation and retroactively create a guideline for investigating other similar assignments, they have, also an educational value in addition to the design checking function.

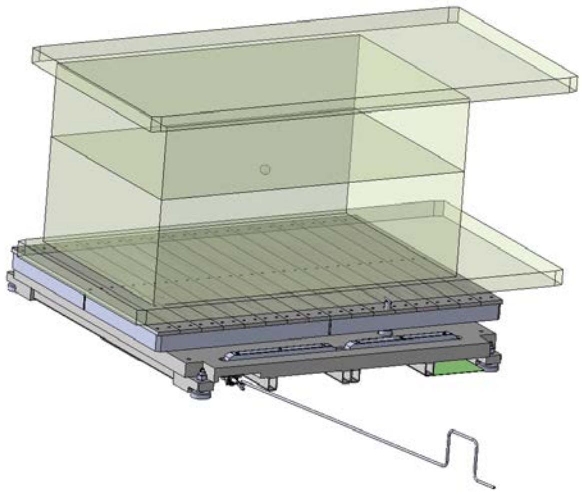

*Fig. 16: A computer image of the rotating station in Solidworks*

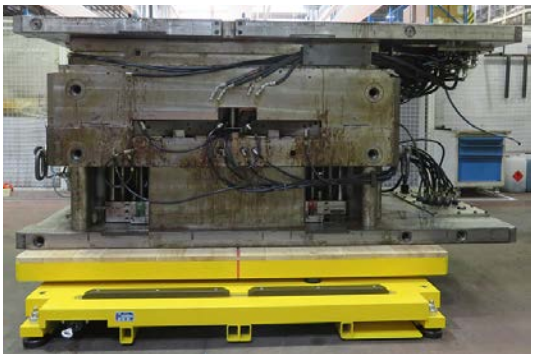

*Fig. 17: A real image of the rotating station*

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