DIZAJN SKLOPA HIDRAULIČNOG CILINDRA NATKOLJENIČNE PROTEZE SA CILJEM SMANJENJA MASE

DESIGN OF THE HYDRAULIC CYLINDER ASSEMBLY OF AN ABOVE-KNEE PROSTHESIS WITH THE OBJECTIVE OF MASS REDUCTION

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Hidraulični cilindri koji se koriste na natkoljeničnoj protezi noge moraju odgovarati veoma preciznim geometrijskim, funkcionalnim i masenim zahtjevima. Takve je moguće naći samo u inostranstvu uz komplikovanu nabavku i po visokim cijenama. Tema ovog rada je analiza mogućnosti izrade i nabavke takvih cilindara na domaćem tržištu, te optimizacija rješenja dobivenih od domaćih firmi sa ciljem smanjenja mase. U radu je izvršena analitička i numerička analiza minimalne potrebne debljine stjenke cijevi cilindra i minimalna debljina klipnjače. Na kraju rada je dato optimizirano rješenje sa smanjenom masom za oko 70% u odnosu na idejno rješenje.

Professional paper

SUMMARY

The hydraulic cylinders used on the above-knee leg prosthesis must meet very precise geometric, functional and mass requirements. Such can only be found abroad, but procurement is complicated and prices are high. The topic of this paper is the analysis of the possibility of manufacturing and purchasing such cylinders on the domestic market, and the optimization of solutions obtained from domestic companies with the aim of reducing mass. In the paper, an analytical and numerical analysis of the minimum required wall thickness of the cylinder tube and the minimum thickness of the connecting rod was performed. At the end of the paper, an optimized solution with reduced mass by about 70% compared to the conceptual solution was given.

1. INTRODUCTION

The need for anatomical prosthesis arises in cases of people who miss their limbs, as a characteristic they are born with or acquired in various ways. The case of a person who lost his/her leg above the knee is particularly difficult, because the prostheses that replace such loss are either ineffective or very expensive. The team led by Prof. Dr Remzo Dedić developed a hydraulically powered prosthesis that imitates the movement of a healthy leg and enables the user to move normally, even up stairs. One of the problems for the future mass production and use of these prostheses is the availability to acquire hydraulic cylinders. The cylinders used for the development of prototype were purchased abroad at high prices and complicated procurement procedures. Therefore, there was a need to research the possibility of production and procurement of suitable cylinders on the domestic market [3].

The topic of this work is the analysis of the possibility of optimizing hydraulic cylinders that can be produced on the domestic market with the aim of reducing their mass, and with the ultimate goal of achieving the performance of a cylinder that could be produced in Bosnia and Herzegovina, which could meet technical, geometric and mass criteria. First, an analysis of production possibilities and availability of cylinder materials on the domestic market was performed. Then the conceptual solution from the company Termika d.o.o. Zenica was analyzed, and the given solution was analyzed from the point of view of technical and mass requirements.

After that, the optimization of the wall thickness and the diameter of the connecting rod of the mentioned solution was carried out with the aim of reducing the mass.

The optimization process proceeded in four directions. The first direction is to adjust the cylinder assembly with the aim of saving mass. Different ways of performing the oil inlets, cylinder bottoms and receiving elements were analyzed and the most favorable solution was given. The second direction is optimization by changing the material, where the possibility of replacing the steel tube of the cylinder with an aluminum tube was analyzed. The third direction is optimization by reducing the inner diameter of the cylinder tube. Here, it was tested whether by reducing the working surface, i.e. the inner diameter of the tube, is possible to achieve the minimum desired force on the rod. In addition, the mass of such a cylinder was numerically calculated and compared with the mass of a cylinder with a larger diameter. The fourth direction is the analysis of the possibility of reducing the wall thickness of the hydraulic cylinder tube, where the stresses that occur due to oil pressure on the cylinder tube for different tube wall thicknesses are analyzed. The analysis was done analytically and numerically, and the minimum required wall thickness that meets the strength requirements was given.

At the end, the conclusions of the analysis of optimization possibilities were summarized, and the optimal solution was given as the best combination of the analyzed versions, and a 3D model of the assembly of the most favorable solution was given. Instructions were also given in which direction the further optimization of the hydraulic cylinder could move.

2. DEFINING CYLINDER ASSEMBLY

The selection of appropriate components and the performance of the cylinder play a major role in the effort to reduce mass. The initial version from which the optimization started is the version of the cylinder produced by HP-FLUID d.o.o. Gradačac, which was made from available elements, but geometrically and in terms of mass did not meet the required criteria.

Figure 1 shows a hydraulic cylinder manufactured by HP-FLUID d.o.o. Gradačac at the request of the prosthesis development team. In addition to the excessive mass, the mentioned cylinder was also too large in terms of geometry and it was not possible to mount it in the intended place.

The next step in the development of the cylinder, which can be produced by domestic companies, was made in cooperation with the company 'Termika' d.o.o. Zenica. Based on the materials and components available to them, the company offered a solution that solved part of the problems that appeared with the previous solution. Figure 2 shows the conceptual solution of the company 'Termika' d.o.o. Zenica.

Figure 1. *Cylinder of HP-FLUID d.o.o. Gradačac compared to the geometry of the prosthesis*

Their solution was more favorable because, unlike the earlier solution, the front and bottom are connected to the cylinder tube from the inside, and this saves material because in that case the front and bottom are smaller in diameter. However, this solution also had a mass of over 1 kg and was therefore unusable.

Figure 3 shows the assembly of the best performance of the cylinder with a welded bottom and a face connected to the cylinder from the inside.

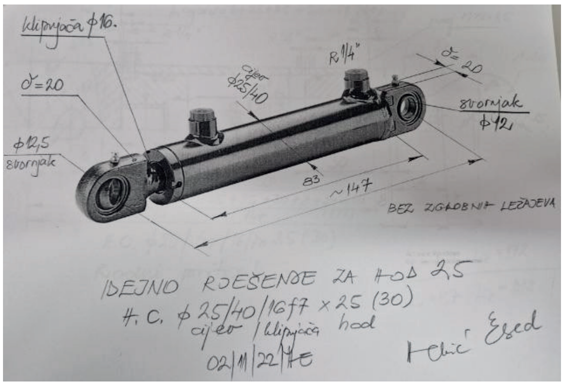

Figure 2. *Conceptual solution of the hydraulic cylinder from the company "Termika" d.o.o.*

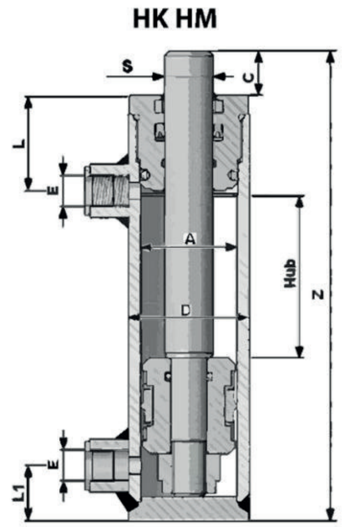

Figure 3. *The principle of the cylinder design with internal face assembly, welded bottom and oil fittings directly connected to the cylinder tube [6]*

3. OPTIMIZATION OF THE MATERIALS AND GEOMETRY OF THE HYDRAULIC ASSEMBLY

The hydraulic cylinder tube is the largest element of the hydraulic assembly that is analyzed, both geometrically and by mass. One of the ideas to reduce the mass of the cylinder is to reduce the thickness of the tube wall and change the material of the tube. The assumption is that the thickness of the tubes offered in the catalog is intended for a wide range of possible applications, and therefore for cylinders of significantly longer lengths than the cylinder being analyzed, which also experience significantly higher bending and other loads. Therefore, it is possible to make the assumption that these loads could also be handled by a cylinder with a significantly smaller diameter. In order to confirm the assumption, a numerical and analytical analysis of the minimum necessary thickness of the tube wall that can withstand a nominal pressure of 100 bar was carried out [4]. Below is an analysis of steel and aluminum tubes with a diameter of ϕ25 and ϕ20.

3.1. Determination of the minimum required wall thickness of aluminum and steel hydraulic tubes with an internal diameter of ϕ25

The SolidWorks software with the add-in Simulation was used for numerical analysis. The tube is modeled in a simplified form. The domestic market offers a 25/40 steel tube, which inner diameter is 25 mm, outer diameter is 40 mm, i.e. the wall thickness is 7.5 mm. The most commonly used materials for making hydraulic cylinders are steel 1020, 1035, 1045, ST 52-3 etc. For the analysis of the stresses that occur at the maximum pressure on the tube, a tube made of steel 1020 will be analyzed. The specified material has a lower yield stress value compared to all other steels, and further improvement of the element could be done by choosing one of the stronger materials. In this paper, the analysis will be limited to AISI 1020 steel, since its yield stress is $R_{eH} = 375\ MPa$ and tensile strength $R_m = 520\ MPa$ [7][8][9].

Aluminum AA6063 T5 with yield stress $R_{eH} = 110\ MPa$ and tensile strength $R_m = 160\ MPa$ is most often used for steel production. [10]

The cylindrical tube is shown simplified, without oil connection and without other components such as bottom, guide housing etc.

Given that body movement must be prevented in all directions, one front side of the tube is marked as stiff, and the other free. Fixed support has no effect on the results, since it is not near the part where the fluid pressure is acting. The pressure is indicated as uniform over the part of the surface corresponding to the working stroke of the cylinder. A cylinder with a stroke of 35 mm was analyzed because the surface of the fluid pressure is larger, and thus the loads are higher compared to a cylinder with a stroke of 25 mm. The obtained results will certainly be satisfactory for a 25 mm stroke cylinder where smaller loads are applied [4]. The cylinder was analyzed in the fully extended position, i.e. when the working surface is the largest, and thus the load is the largest. Figure 10 shows a simplified model with applied connections and pressure loading.

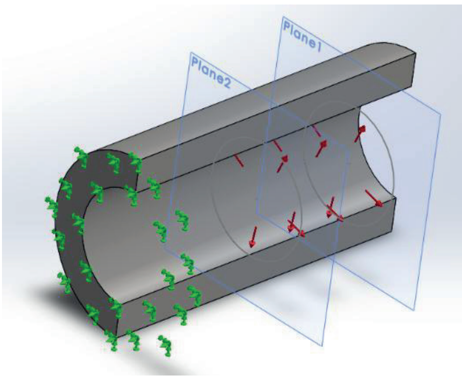

Figure 4. *Simplified cylinder tube model with applied connections and loads*

After creating the network and starting the analysis in SolidWorks, the stress and strain results are obtained. Figure 5 shows the values of the maximum equivalent stress according to Von Mises by points of the body, and the colors indicate the values, where the blue color corresponds to the smallest value of the stress, and red to the largest.

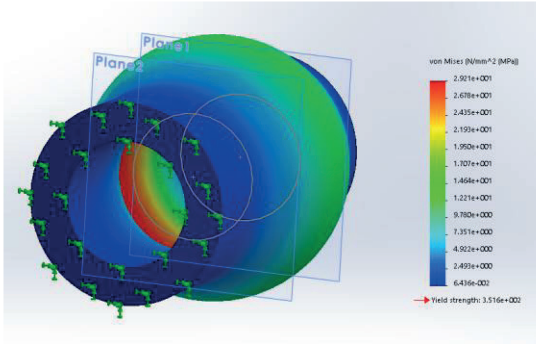

Figure 5. *Von Mieses stress values at different points of the body*

In order to determine the minimum wall thickness, a numerical analysis was performed for different wall thickness values from 7.5 mm to 0.75 mm, and the obtained stress values were compared. In addition, an analysis was made for the case of an aluminum tube, where the wall thickness was changed from 7.5 mm to 1 mm.

The obtained values for different thicknesses of aluminum and steel are shown in Table 1.

Table 1. Value of the maximum stress for different thicknesses of steel and aluminum

Material	**Wall thickness**	**Maximum stress (Mpa)**	**Tube mass (g)**
Steel $R_{eH} = 351,6\ MPa$	7,5 mm	29,21	453,71
	4 mm	45,80	215,92
	2 mm	78,59	100,52
	1 mm	150,3	48,40
	0,75 mm	199	35,95
Aluminum $R_{eH} = 145\ MPa$	7,5 mm	28,93	155,07
	4 mm	45,58	73,80
	2 mm	78,48	34,35
	1 mm	149,3	16,54

Based on the values listed in the table, it can be seen that the stress does not change linearly with the decrease in wall thickness. The above can be interpreted as the dominance of pressure loading due to the action of the fluid on the inner surface, which does not change with the reduction of the wall. Later, tensile stresses in the profile become dominant, and below a thickness of 2 mm the stresses change more significantly. The same happens in case of aluminum.

According to ASME B30.1, the defined safety factor is ν = 2. For the specified safety factor, the minimum required wall thickness is 1 mm for steel, and 2 mm for aluminum. If the mass of aluminum and steel tubes of the specified thicknesses is analyzed, it is concluded that there is no big difference, i.e. the steel tube is 14 g heavier. The mentioned difference is not big, and due to easier workability and connection by welding with the bottom and the oil connections, it is concluded that the most favorable variant is a steel tube with a thickness of 1 mm. Weight reduction of the optimized profile compared to the profile recommended by Termika d.o.o. Zenica is 90%, i.e. 400 g.

The described optimization is easy to implement because tubes that are available on the market are used, and it is not necessary to buy whole rods and pay for transportation costs, customs etc. It is possible to reduce the diameter on a lathe, by scraping the outer surface and removing it from the initial dimension from 25/40 to 25/27. This procedure would not damage the factory quality of the inner surface, which is necessary for the proper functioning of the hydraulic cylinder.

In order to finally adopt the selected optimized tube, it is necessary to check whether it is suitable for use from the aspect of assembly and connection technology with other elements. The cylinder tube is connected to the oil ports on the side and to the bottom of the cylinder by welding. The welding process can be performed on such thin elements, but there is a high risk of deformations that may occur in the weld zone. The risk is especially great if you take into account the tolerances of the internal moving parts, which should not be reduced because it could lead to difficult work, accelerated wear or oil leakage from the working area to the return. The connection to the housing of the guide is made through a threaded connection, and since the guide is on the inside, it is necessary to thread the cylinder tube from the inside. Since, according to [2], for a small M20 thread, the pitch is 1.5 mm, and the depth of threading is 0.75 mm, the use of a tube with a wall thickness of 1 mm is not appropriate, because the remaining load-bearing part would be 0.25 mm thick. Therefore, the minimum required thickness that would meet the stated requirements of joining technology should be 2 mm.

3.2. Analysis of the possibility of applying a cylinder with an inner diameter of ϕ20

One of the ways of possible optimization of the cylinder is the analysis of the possibility of using a hydraulic tube with an internal diameter of 20 mm. This type of tube is not offered by domestic manufacturers of hydraulic cylinders, but it is possible to order it from foreign suppliers [6]. In the case of mass production of these prostheses, this problem would be overcome, because the additional costs would be divided into several pieces. An analytical calculation of the force produced by such a cylinder, as well as, mass analysis and comparison with a cylinder with an internal diameter of 25 mm is below.

The settings of the numerical analysis are as in the case of the 25 mm tube. Figure 6 shows a cylinder with applied supports and loads.

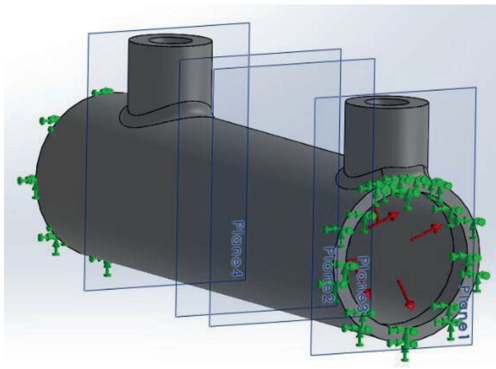

Figure 6. *Tube ϕ20 with applied loads and supports*

After the analysis, the results were obtained as shown in Figure 7.

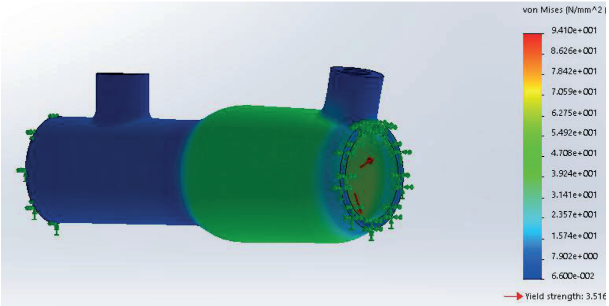

Figure 7. *Obtained stress values for tube ϕ20/24*

Figure 8 shows the maximum stress occurs at the point of stress concentration and is 94.1 MPa at that point. The stress on the outer surface is 43.25 MPa, and on the inner surface of the cylinder is 58.71 MPa.

Given the specified stress is far below the permitted one, it can be concluded that this tube would be oversized, and that it is possible to use tubes with a smaller wall thickness. Table 2 shows the values of maximum stresses and stresses on the outer and inner surface for different wall thickness values.

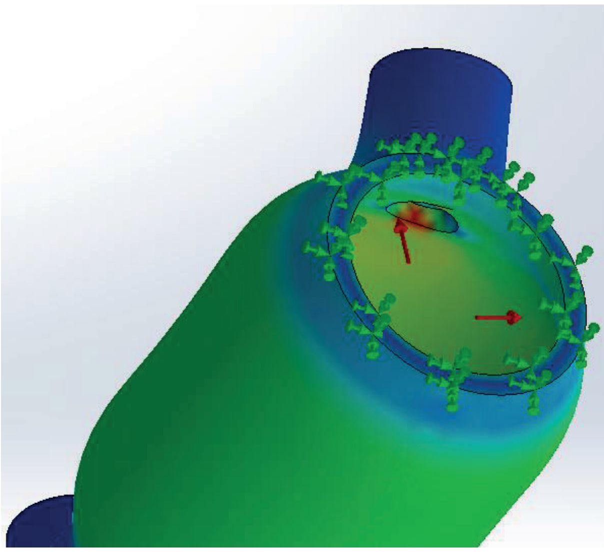

Figure 8. *Stress concentration at the place of the oil connection*

Table 2. Stress values for different tube wall thicknesses with an internal diameter of ϕ20

Wall thickness	**The maximum eq. stress**	**Stress on the outer surface**	**Internal surface tension**
2 mm	94,1 MPa	43,25 MPa	58,71 MPa
1,5 mm	122,5 MPa	58,74 MPa	74,29 MPa
1 mm	170 MPa	90,7 MPa	97,6 MPa
0,5 mm	260 MPa	193,5 MPa	196,6 MPa

It can be seen from this table that even a tube with a wall of 0.5 mm would be sufficient for the specified load with a pressure of 100 bar. However, although it is possible to use tubes with a smaller wall thickness, they are not available as standard in the supply, and it would be necessary to produce them by processing the available tubes through scraping the outer surface. This could affect the possibility of making threads for the threaded connection with the guide housing, and the possibility of welding the bottom of the cylinder. In addition, the mentioned procedure would additionally increase the price of the product.

3.3. Analytical stress analysis of the cylinder tube

In order to confirm the results obtained by numerical analysis, it is necessary to perform a check using analytical calculation. For verification purposes, the results of numerical and analytical analysis of a cylinder with an inner diameter of ϕ20 mm and an outer diameter of ϕ24 mm will be compared, i.e. wall thickness 2 mm.

The calculation of the stress state of the tube of the hydraulic cylinder corresponds to the calculation of the state of pressure vessels. The ratio of the wall thickness to the inner diameter of the tube determines whether it is a thin-walled or thick-walled element. If the ratio t/d_u is less than 0.1, then it is a thin-walled element, otherwise it is a thick-walled element. In the analyzed case, this ratio is 2/20 = 0.1, so it is possible to apply the calculation for a thick-walled element [2]. There is a triaxial stress state in the tube wall, i.e. axial, tangential and radial stresses act. The value of the tangential stresses on the inner surface of the cylinder is

$$\sigma_{t-in} = -P\frac{u^2+1}{u^2-1} \quad \text{... (1)}$$

The radial stresses are equal to the pressure P, which maximum calculated value is 100 bar or 10 MPa.

There are no radial stresses on the outer surface, and the tangential stresses are equal

$$\sigma_{t-out} = -P\frac{2}{u^2-1} \quad \text{... (2)}$$

where u is the ratio of the outer and inner diameter

$$u = \frac{D}{d} = \frac{24}{20} = 1{,}2$$

Stress values are obtained on the basis of the above formulas

$$\sigma_{t-in} = -55{,}45\ MPa$$

$$\sigma_{t-out} = -45{,}45\ MPa$$

The radial stress on the interior is

$$\sigma_{r-in} = 10\ MPa$$

After determining the value of the radial and tangential stresses on the inner and outer surface, it is necessary to determine the value of the equivalent stress according to one of the fracture theories. Given that it is a ductile material, the equivalent stress will be determined according to the theory of the largest specific deformation work (Von Mieses):

$$\sigma_{ekv} = \sqrt{\frac{1}{2}[(\sigma_1-\sigma_2)^2+(\sigma_2-\sigma_3)^2+(\sigma_1-\sigma_3)^2]} \quad \text{... (3)}$$

By entering the values, the values of the equivalent stress on the inner and outer surface are obtained, given in Table 3. Also, the stress values obtained numerically are listed in the table.

Table 3 Values of analytically and numerically obtained stress values

Position	**Analytical [MPa]**	**Numerical [MPa]**
Inner surface	51,18	58,71
Outer surface	41,36	43,25

By analyzing the obtained values, it can be concluded the analytically obtained results are largely in agreement with the numerically obtained results. The deviation can be attributed to the difference in geometry, because the numerically analyzed model was performed with oil connections and oil inlet openings. Oil inlets weaken the structure, while connections affect the difference in stress distribution along the depth of the wall. In addition, the proximity of the support on one side has an impact on the results. On that side, the cylinder tube is welded to the bottom of the cylinder, and this connection is shown as an ideal fixed support.

In any case, both analytical and numerical analysis showed the actual stresses in the element are far below the permitted ones, almost 5 times lower. According to the ASME B 30.1 standard for hydraulic cylinders, it is recommended to use safety factor 2. Therefore, a tube with a smaller wall thickness could also be used for production of this cylinder.

3.4. Stress analysis of the rod

3.4.1. Analytical analysis of the minimum required hydraulic rod thickness

According to [9], class 35 and 45 steels are used for the production of hydraulic rods. In the supply offer according to the catalog [6], the hydraulic rod material is steel EN 10294-1 20MnV6. For the purposes of analytical calculation and numerical analysis, steel 1035 with yield stress $R_{eH} = 282\ MPa$ will be considered. The force acting on the hydraulic rod is a pressure force and its intensity is determined according to:

$$F = P \cdot A \qquad \text{... (4)}$$

where P is the maximum pressure of the working fluid, and A is the surface area of the cylinder.

The force intensity is 3000 N for a cylinder with a diameter of ϕ20 and 4900 N for a cylinder with a diameter of ϕ25 mm. In the return stroke, the force is smaller, so that case will not be analyzed. A connecting rod with a diameter of 10 mm is analyzed.

The value of the pressure stress in the connecting rod is

$$\sigma_p = \frac{F}{A_{kl}} = \frac{F \cdot 4}{D^2 \pi} = \frac{3000 \cdot 4}{10^2 \cdot \pi} = 38{,}19\ MPa \qquad \text{... (5)}$$

For the case of a cylinder with an inner diameter of ϕ25 mm, it is obtained

$$\sigma_p = \frac{4900 \cdot 4}{10^2 \cdot \pi} = 62{,}39\ MPa \qquad \text{... (6)}$$

Based on the above, it can be seen that the piston rod is oversized, and that the safety factor is 7.38.

The Euler buckling check is done according to the diagram available in [5] on page 13, where - for a connecting rod 10 mm, length 100 mm with both joint supports - the permissible pressure force is 10 kN for a safety factor ν = 5, where it can also be seen that the specified connecting rod is oversized.

It would be possible to use a connecting rod with a smaller diameter, but since there are no connecting rods with a smaller diameter in the available catalogs, and given the mass of the connecting rod piston is 52.96 g, attempts to further reduce the diameter would not bring much weight savings, and could significantly increase costs, because special parts - such as gaskets that are not available in the catalogs - would have to be ordered.

Based on the above, a standard piston rod ϕ 10 mm will be used, as well as the gaskets available for it.

3.4.2. Numerical analysis of the minimum required hydraulic rod thickness

As a check of the analytical calculation, a numerical analysis of the stress state of the connecting rod was, also, performed. The material and geometrical parameters were selected as for the previously mentioned analytical calculation. The support is placed on the part of the joint and defined as completely immobile, and the pressure force is applied to the part where the piston rests on the connecting rod and thereby transmits the pressure force further to the support. From the diagram shown in Figure 9, it can be seen the maximum stress that occurs is 113.1 MPa. However, that is the stress on the sharp edge of the surface on which the force is applied. This high stress is a consequence of stress concentration and is not relevant for the calculation of the thickness of the connecting rod. In order to determine the stress on the piston rod, the 'probe' function was used, and Figure 9 shows that the stress on the piston rod body is 38.2 MPa, which fully corresponds to the analytically obtained stress.

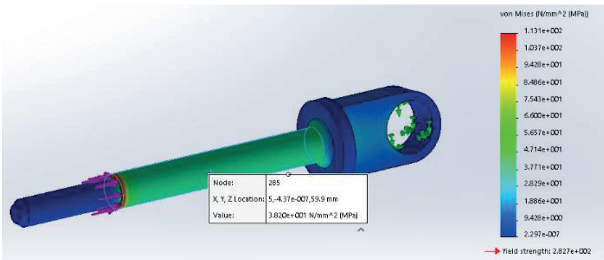

Figure 9. *Stress state of a connecting rod ϕ10 mm loaded with a force of 3000 N*

3.5. Model of the hydraulic cylinder assembly

In order to determine the difference in the mass of cylinders with an inner diameter of ϕ20 and ϕ25, their models were made. Both cylinders are modeled with a wall thickness of 2 mm, and with the same components and materials. Figure 13 shows the assembly in its original form and in cross-section.

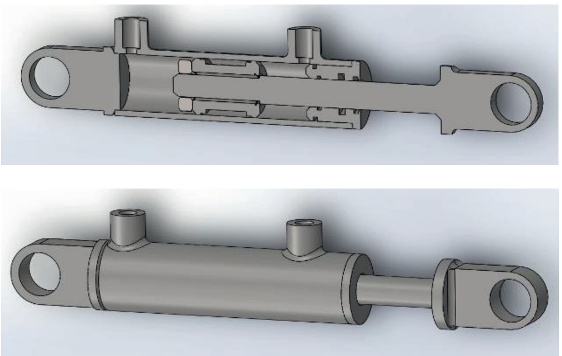

Figure 10. *Hydraulic cylinder assembly*

Using the *Mass properties* tool located in the *Evaluate* section of the toolbar, the total mass of hydraulic cylinder assemblies of both diameters was determined. The mass of a cylinder with an inner diameter of ϕ20mm is 331.12 g. The mass of a cylinder with an inner diameter of ϕ25 mm is 452.93 g. The difference in mass is 121.81 g, that is, the saving in mass is 27%.

Figure 11 shows the disassembled cylinder assembly.

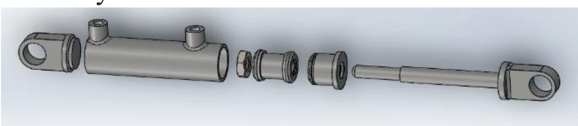

Figure 11. *Cylinder components in an exploded view*

4. CONCLUSION

The possibility of optimizing the cylinder with the aim of reducing its mass was analyzed. The optimization was analyzed in several directions, and conclusions and the optimal solution for cylinder production were given. During optimization, the cylinder diameter and wall thickness were reduced, so instead of the 25/40 tube, a 20/22 tube was used. In addition, the thickness of the connecting rod was reduced, and a ϕ10mm connecting rod was used instead of a ϕ16mm connecting rod. Also, with the aim of reducing dimensions and weight, some elements were changed on the cylinder, the thickness of the connecting rod guide was reduced, the oil connections were mounted directly on the cylinder tube and the position of the bolt hole was optimized. All selected elements are standard and are selected from available catalogs.

The finite element method was used to check the stress state of the newly selected components using the Solidworks Simulation software. All elements satisfy stress analysis with a safety factor higher than 2. The results obtained for the cylinder tube were also confirmed by analytical calculation.

The production of an optimized cylinder is simple and does not differ in terms of production technology from the conceptual design, so production could be carried out by domestic companies. Most of the cylinder elements are available in the domestic market, with the exception of the tube and connecting rod, which are available in the region. The cost price of the optimized hydraulic cylinder should be lower than the conceptual solution because less material was used, and the production procedures and technology were not changed.

The final result of the optimization is that the mass of the entire cylinder has been reduced from 1120 grams to 330.47 grams, i.e. by about 70%. The mass of the optimized cylinder is acceptable for installation on the prosthesis and such a cylinder would not cause too much load for the user of the prosthesis. Also, this cylinder meets all geometric requirements at the same time.

Further research in the direction of reducing the mass of the cylinder should go in the direction of additional reduction of the wall thickness, up to the numerically confirmed minimum of 0.75 mm. In this sense, it would be necessary to analyze the possibilities of making such a cylinder, the possibilities of welding oil connections to a tube with such a small wall thickness, and the possibilities of making threads for connecting the connecting rod guide. In addition, further optimization could go in the direction of optimizing the geometry of the cylinder ends, i.e. bolt assembly locations, where it would be possible to further reduce the wall of those elements.

The aforementioned would improve the already achieved optimization results, but it is possible it would, also, lead to an increase in the price of the product, due to the use of more special technologies and materials.

In the end, it is possible to state that the optimization was carried out successfully and a solution obtained meets the requirements from the aspects of functionality, mass, price, availability of elements and manufacturing technology.

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