

## Comparison of Analytical Stress Analysis and Numerical Calculation of Mobile Work Machine Part

Mária Blatnická<sup>1</sup>, Miroslav Blatnický<sup>2</sup>, Ján Dižo<sup>2</sup>, Milan Sága<sup>1</sup>

<sup>1</sup>Department of Applied Mechanics, Faculty of Mechanical Engineering, University of Žilina. Univerzitná 1, 010 26 Žilina. Slovakia. E-mail: maria.blatnicka@fstroj.uniza.sk, milan.saga@fstroj.uniza.sk

<sup>2</sup>Department of Transport and Handling Machines, Faculty of Mechanical Engineering, University of Žilina. Univerzitná 1, 010 26 Žilina. Slovakia. E-mail: miroslav.blatnický@fstroj.uniza.sk, jan.dizo@fstroj.uniza.sk

The aim of the article is to verify dimensions of the hydraulic arm column (the necessary cross-sectional area) by analytical dimensional calculation and thus to design a lifting rotary arm which will be located on the pick-up car body (Figure 1). After the analysis of dimensions, the next step is creation of the structure in FEM program and then a numerical analysis will be carried out for verification of stress in the structure already with the values that are not available for the preliminary design (e.g. the structure weight). The next step in the solution will be to import the proposed and by strength calculations checked geometry into the multibody system program, where the dynamic response of the structure will be monitored, depending on the size of the load and the movement possibilities of this mechanism.

**Keywords:** Numerical analysis, Hydraulic arm, Stress, Work machine

### 1 Introduction

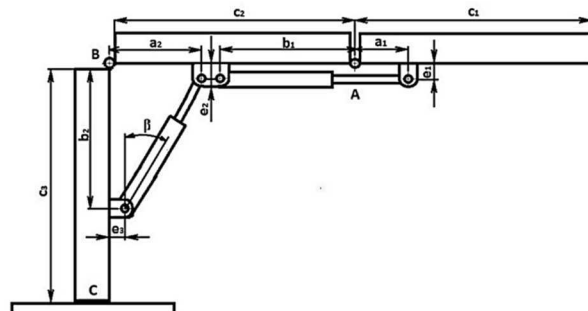
The successful development of mankind to the state we see today has its origins in the ages when man had to fight for survival in the original living conditions. Repeating successful activities that have brought improved living conditions has also triggered brain activity, i. e., from simple logical operations to memorization and analysis [1]. The result of this process was creations of man, whose improvement led to progress consisting of the construction of machines and heavy machinery. This process of developing technical systems that relieve man from repetitive physical work is called mechanization, and there is no need to talk too much about the importance of machinery for manipulating material. Mechanisation is an important means of increasing productivity, quality and competitiveness of production. The effort is to make the individual work actions as short and simple as possible, to make them easy to learn while demanding a minimum of human effort. Mechanisation greatly relieves man from heavy physical work, for example, in dangerous or harmful environments. Such a mechanism is also the proposed hydraulic arm, which greatly facilitates the manipulation process with great operability and with the condition of minimal production costs we achieve the most cost-effective ratio [2].



**Fig. 1** Platform for mounting the proposed hydraulic arm

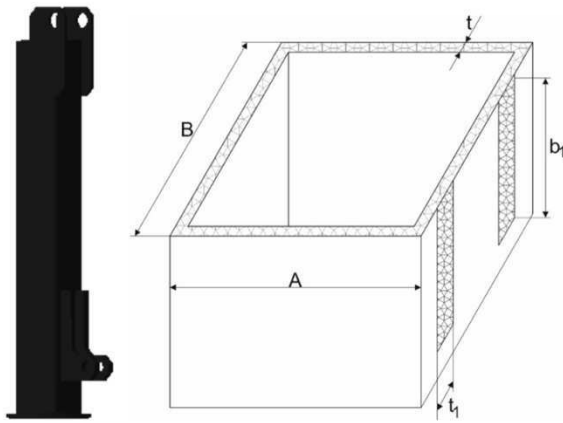
### 2 Verification of column in critical cross-section

This calculation was preceded by the determination of the maximum load resulting from the maximum weight of the lifted load, which was set on  $m = 300$  kg [3]. Subsequently, calculations were made to determine the position at which the column is most stressed (Fig. 2).



**Fig. 2** The most effective position of the arm for maximum column loading

The largest possible stress in the critical cross-section of the column body arises when loading the arm with geometry according to Fig. 2, and that is to be defined. After considering loading effects of the forces acting on a structure it was found that they cause tensile/compressive stress, shear stress and bending moment. We assume that the most exposed area on the column's body is a common border of critical cross-sections (Fig. 3 right), i. e., vertically oriented surfaces to which is welded the holder of the second hydraulic motor with dimensions  $t1 \times b1$  and a horizontally oriented surface of the intended section taken close to the welding edge with the hydraulic motor holder (Fig. 3). This is due to the fact that the greatest bending moments and at the same time the most force effects act on this surface. Therefore, it was determined that the location of the greatest load on the structure is between the holder of the second hydraulic motor and the column [4].



**Fig. 3** 3D model of the arm's column (left) definition of the expected extreme stress location in the common crossing of the critical cross sections (right)

The effects of the force  $R_B$  on the stress value in the critical cross section were detected earlier. To this, it is still necessary to add the effect of another acting force by the superposition method, i. e. force  $F_{HM2x}$  acting in the axis of the hydraulic motor between the middle arm and the column (see Fig. 2). The force  $F_{HM2x}$  in the given cross

section causes the compressive stress  $\sigma_d$ . Its value is determined by (1):

$$\sigma_d = \frac{F_{HM2x}}{2 \cdot S_1} \quad (1)$$

By solving equation (1) we get  $\sigma_d = 0.93$  MPa. Furthermore, the force  $F_{HM2x}$  causes the shear stress  $\tau_{s2}$  whose value we determine by (2):

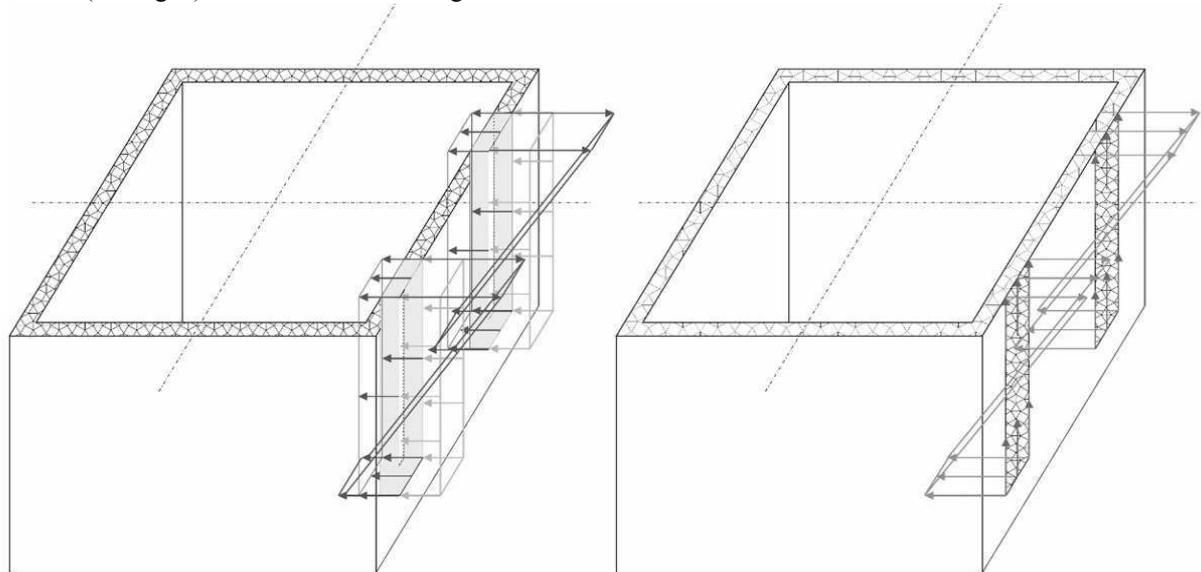
$$\tau_{s2} = \frac{F_{HM2x}}{2 \cdot S_2} \quad (2)$$

The size of the shear surface  $S_2$  is shown in Fig. 4 left (grey area) and its size is (3):

$$S_2 = (2 \cdot t_1 \cdot t + 2 \cdot b_1 \cdot t) \quad (3)$$

By solving equation (3) we get  $S_2 = 3,600$  mm<sup>2</sup> and from equation (2) we get  $\tau_{s2} = 0.82$  MPa. Further, the force  $F_{HM2x}$  causes eccentric compressive stress because the holder is not loaded symmetrically by the force of  $F_{HM2x}$ , and thus produces a bending stress  $\sigma_{o3}$  whose value is determined by (4):

$$\sigma_{o3} = \frac{M_{o3}}{2 \cdot W_{od}} \quad (4)$$



**Fig. 4** Critical cross-section stress induced by  $F_{HM2x}$  (left) and  $F_{HM2y}$  (right)

Value of bending moment  $M_{o3}$  is (5):

$$M_{o3} = F_{HM2x} \cdot e_1 \quad (5)$$

The elastic section modulus  $W_{od}$  of the penetration of the holder and the column area  $S_1$  (rectangular cross-section of the dimension  $t_1 \times b_1$ ) at its orientation towards the load is determined from equation (6):

$$W_{od} = \frac{t_1 \cdot b_1^3}{6 \cdot b_1} \quad (6)$$

By solving the equations (4), (5) and (6) we get  $\sigma_{o3} = 2.1$  MPa. The stress in the considered cross-section caused by the force  $F_{HM2x}$  is shown in Fig. 4 left. Light-green, violet and red colours represent the compressive stress  $\sigma_d$  in the area  $S_1$ , shear stress  $\tau_{s2}$  in the area  $S_2$ , and

bending stress  $\sigma_{o3}$  from the bending moment  $M_{o3}$  due to the eccentric loading of the holder with the surface  $S_1$ , respectively. The force  $F_{HM2y}$  in the given cross section causes the shear stress  $\tau_{s3}$  whose size is determined by (7):

$$\tau_{s3} = \frac{F_{HM2y}}{2 \cdot S_1} \quad (7)$$

By solving equation (7) we get  $\tau_{s3} = 4.8$  MPa. Further, the force  $F_{HM2y}$  causes eccentric tensile stress and thus causes the bending stress  $\sigma_{o4}$ , whose size is determined by (8):

$$\sigma_{o4} = \frac{M_{o4}}{2 \cdot W_{od}} \quad (8)$$

For  $M_{o4}$  bending moment applies (9):

$$M_{o4} = F_{HM2y} \cdot e_3 \quad (9)$$

By solving the equations (6), (8) and (9) we get  $\sigma_{o4} = 10.3$  MPa. The stress in the considered cross-section caused by the force  $F_{HM2y}$  is shown in Fig. 4 right. Blue and orange represent uniform distribution of the shear stress  $\tau_{s3}$  in the surface  $S_I$  and tension  $\sigma_4$  of the bending moment  $M_{o4}$  in the area  $S_I$ , respectively.

### 3 Determination of the maximum equivalent stress in the structure

The next step of the task solving methodology is calculation of the maximum equivalent stress  $\sigma_{redmax}$ . This is composed by the normal stresses  $\sigma_{NI}$ , for which respecting the stress from the previous (Article J.D. EVM 2017) and the current calculations can be written (10):

$$\sigma_{NI} = \sigma_t + \sigma_{o1} + \sigma_{o2} \quad (10)$$

By solving equation (10) we get  $\sigma_{NI} = 56.64$  MPa. The equivalent stress is further composed by the normal stresses  $\sigma_{N2}$ , for which respecting the stress from the previous (article J.D. EVM 2017) and the current calculations can be written (11):

$$\sigma_{N2} = \sigma_{o3} + \sigma_{o4} - \sigma_d \quad (11)$$

By solving equation (11) we get  $\sigma_{N2} = 11.47$  MPa. The equivalent stress is further composed by the tangential stresses  $\tau_I$ , for which respecting the stress from the previous (article J.D. EVM 2017) and the current calculations can be written (12):

$$\tau_I = \tau_{s1} + \tau_{s2} \quad (12)$$

By solving equation (12) we get  $\tau_I = 2.62$  MPa. The resulting normal stress  $\sigma_{VN}$  (13) is:

$$\sigma_{VN} = \sqrt{\sigma_{N1}^2 + \sigma_{N2}^2} \quad (13)$$

By solving equation (13) we get  $\sigma_{VN} = 57.8$  MPa. For the resulting tangential stress  $\tau_V$  the following applies (14):

$$\tau_V = \sqrt{\tau_1^2 + \tau_{s3}^2} \quad (14)$$

By solving equation (14) we get  $\tau_V = 5.47$  MPa. Then the equivalent stress  $\sigma_{red}$  is calculated according to HMH theory (15):

$$\sigma_{red} = \sqrt{\sigma_{VN}^2 + 3 \cdot \tau_V^2} \quad (15)$$

By solving equation (15) we get  $\sigma_{red} = 58.57$  MPa. Since the critical cross section between the holder and the column gives rise to a notch effect due to right angles between them and that the load is mainly bending and tension, the calculated reduced stress must be multiplied by the shape coefficient  $\alpha$  which, for such conditions, acquires the value  $\alpha = 2$  (-). Then the maximum reduced stress in the critical point (16) is:

$$\sigma_{redmax} = \alpha \cdot \sigma_{red} \quad (16)$$

By solving equation (16) we get  $\sigma_{redmax} = 117.14$  MPa.  $\sigma_{redmax}$  is the greatest stress in the construction of the hydraulic arm column determined by the analytical calculation.

### 4 Numerical calculation of stress in the column

In order to verify the accuracy of the analytical calculation and thus to increase the overall probability of the correct result, the column is also solved using the numerical method. It was first necessary to model the column geometry [5]. Because it is made of a square seamless tube, it was modelled by shell elements (Fig. 5).

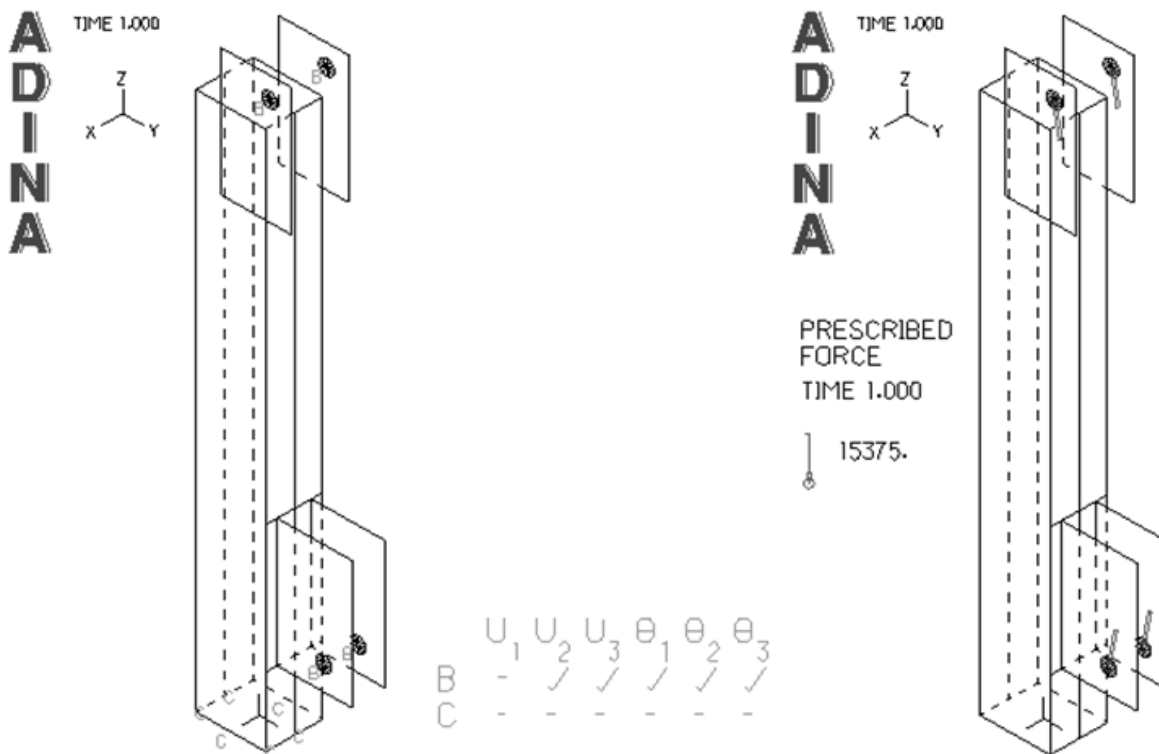


Fig. 5 Shell model of the column is made by means of mid-planes with the fixity (left) and loading (right)

This was followed by entering boundary conditions [6,7]. The column was fixed as it is in reality (Fig. 5 left). The load was represented by the resultant components of

the two acting forces distributed into corresponding number of holders - 2 (Fig. 5 right).

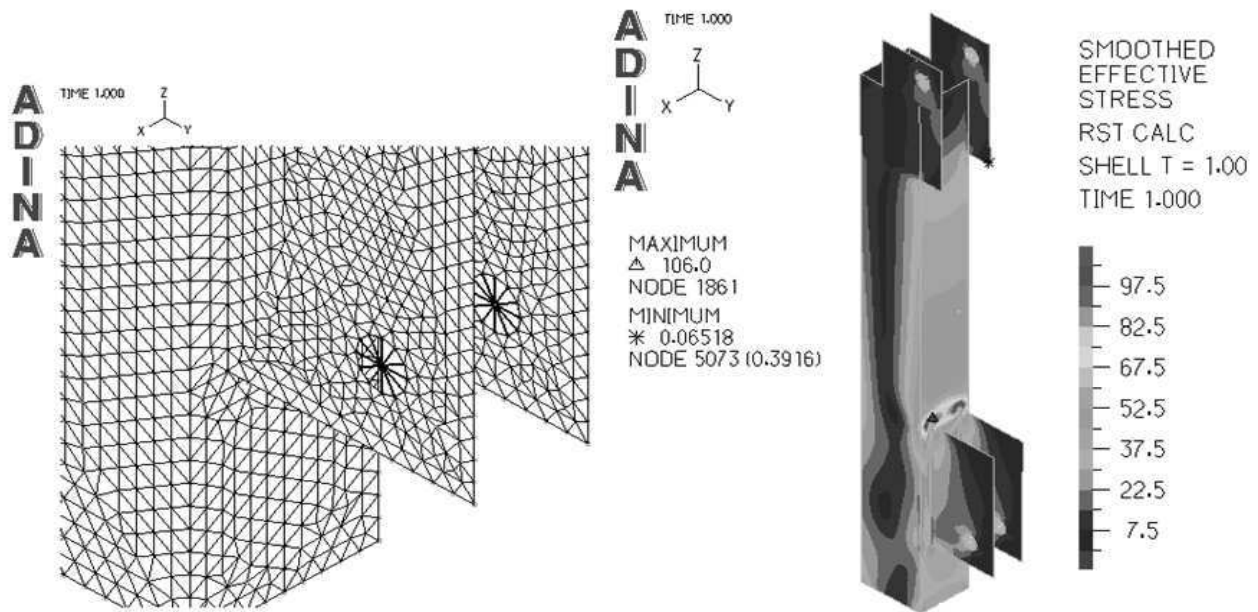


Fig. 6 Detail of column mesh (left) and reduced von Mises stress in the structure

Linear tetrahedron elements with a 4 mm element size were used to mesh the model (Fig. 6 left). After running the solution, we can display results of the equivalent von Mises stress in the post processor (Fig. 6 right). Results show that the chosen profile is designed correctly, since the maximum stress in the structure is lower than the allowable stress = 140 MPa, which can be written by equation (17):

$$\sigma_{redmax} = 117.14 \text{ MPa} \leq \sigma_{redov} = 140 \text{ MPa} \quad (17)$$

## 5 Conclusion

The aim of the paper was to analytically solve column part of the versatile small lifting hydraulic arm (Fig. 2). Since the numerical calculations have verified the accuracy of the analytical solution, it can be stated that this goal has been met. After designing and recalculating all arm dimensions, it will be possible to create a precise 3D model of this structure in the CAD program and then numerically calculate the entire arm. Finally, the model will be imported into the MBS program, where dynamic behavioural responses will be monitored when operating with a maximum load of up to 300 kilograms.

## Acknowledgement

This work has been supported by KEGA grant under the contract No. 015ŽU-4/2017.

## References

- [1] BAJLA, J., BRONCEK, J., ANTALA, J., SEKERESOVA, D. (2014). *Mechanical Enginee-*

*ring Tables*. (In Slovak). Selection Standard. Slovak Office Standards, Metrology and Testing. ISBN 978-80-8130-039-4.

- [2] KOPAS, P., JAKUBOVICOVA, L., VASKO, M., HANDRIK, M. (2016). Fatigue resistance of reinforcing steel bars. *Procedia Engineering*, vol. 136, 2016, pp. 193-197.
- [3] HANDRIK, M., VASKO, M., KOPAS, P., MOZER, V. (2016). The linear and nonlinear stability loss of structures due to thermal load. *Procedia Engineering*, vol. 136, 2016, pp. 359-364.
- [4] SAPIETOVA, A., BUKOVAN, J., SAPIETA, M., JAKUBOVICOVA, L. (2017). Analysis and Implementation of Input Load Effects on an Air Compressor Piston in Msc. Adams. *Procedia Engineering*, vol. 177, 2017, pp. 554-561.
- [5] DEKYS, V. (2017). Condition Monitoring and Fault Diagnosis. *Procedia Engineering*, vol. 177, pp. 502-509.
- [6] ZMINDAK, M., NOVAK, P., DEKYS, V. (2017). Finite element analysis of the delaminated composite plates reinforced by unidirectional fibres. *Manufacturing Technology*, vol. 17 (2), 2017, pp. 275-280.
- [7] BATHE, K. J. (1996). *Finite element procedures*. Prentice-Hall, Inc. New Jersey. ISBN 0-13-301458-4.