FEM ANALYSIS OF MAIN PARTS OF A MANIPULATOR FOR MOUNTING A COMPRESSOR TO A CAR EQUIPPED WITH A PNEUMATIC SUSPENSION SYSTEM

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Abstract
The article deals with a FEM analysis of main parts of an engineering design of a pneumatically controlled manipulator, which will serve for mounting a compressor to a POS56 car chassis. This car chassis belongs to SUV cars and it is equipped with a pneumatic suspension system. It is the chassis of a Porsche Cayenne car produced in the Slovak Republic. The designed manipulator will be located on the assembly line of this vehicle. Except of FEM analyses of designed important parts of the manipulator, which are necessary for verification of safety of the design in order to obtain distribution of stresses and deformations in a structure, the it contains also the design of a pneumatic circuit of the manipulator together with pneumatic logic and it individual components. Based on three-dimensional model, there will be possible to generate drawings of the manipulator for manufacturing of the device and its application to the real operation.

Keywords: engineering device, FEM simulations, pneumatic mechanism, vehicle

1. INTRODUCTION

Raising requirements about quality and quantity of products together with demands of large range of goods are conditioned by implementation of mechanisation and automation. More operators in production process lead to increasing of production, however better productivity and effectiveness is possible to reach just by application of means of mechanisation and automation. Recently, it is the only one way by reason that labour reserves are limited. Therefore, the request of a special device has arisen. The device will participate in improving of productivity and production of a manufacturing corporation in Slovakia.

The objective of the solved problem is the production and installation of the device – the manipulator – which will serve for mounting an air compressor to the car chassis. A 3D model of the compressor and a part of a car chassis are shown in Fig. 1. The solved task has to meet all requirements insisted on the manipulator [2], theoretical principles of designing of the given mechanism and working principle as well as the engineering design itself [3] including all aspect of safety, ergonomics, sustainability, geometrical parameters [5, 13] and other requirements, which are demanded during assembly process of cars with pneumatic suspension system [6, 14]. The very important part of the designed manipulator is a pneumatic brake.

2. ENGINEERING DESIGN OF A PNEUMATIC BRAKE OF THE MANIPULATOR

The required working principle of the pneumatic brake is depicted in Fig. 2. When the brake switch is activated, the manipulator has to be fixed in the x direction. In this case, neither control element nor external effects can cause some movement of the manipulator, i.e. neither lifting nor rotating. Interlock of the manipulator in the x direction is ensured by means of the pneumatic cylinder Festo DSBC-32-25-PA-N3, which prevents after activation the movement by a rubber part leaning against an aluminium profile. An
adjustable pressure valve mineralizes an extensive loading of the manipulator structure.

Fig. 2. A scheme of a working principle of the sensor of the lower position

It allows setting the needed pressure of the rubber part to the aluminium profile. Residual pressure in pneumatic cylinders and pneumatic reverse valves Festo HGL avoid moving the manipulator in vertical direction as well as in rotational direction because of external force effects or accidental using of control elements. When the brake is activated, compressed air is released from control elements and the given accidental command cannot be performed.

Another important factor in term of the brake design was prevention of the brake activation in improper position of the manipulator. It is ensured using the sensor of the lower position Festo R3-M5 (Fig. 2). It is needed to note, that all inputs of pneumatic cylinders are equipped with pneumatically controlled reversed valves Festo HGL. They insure the needed pressure in a cylinder despite of loss of pressure. It is important mainly for the lifting cylinder DSBC-50-800-PA-N3, which acts in the vertical direction z. In case of loss pressure, the arm could fall down because of its mass in the direction of the lower position and even it could injure operators. Moreover, these valves improve movement smoothness of individual cylinders. A model of the workplace is shown in Fig. 3.

Fig. 3. A model of the created workplace

The used compressed air is treated by means of the compressed air treatment unit Festo MSB4-1/4-C4-J2-WP. A scheme of the designed manipulator is depicted in Fig. 4.

3. ANALYSES OF SELECTED COMPONENTS BY MEANS OF THE FINITE ELEMENT METHOD

Strength analysis of selected parts of the designed device was subsequent step in designing of the manipulator for mounting an air compressor to a car chassis. It was carried out by means of Ansys software [7, 17] working based on the finite element method [15, 16, 18, 19] and these analyses

Fig. 4. A scheme of the designed pneumatic system
are important to verify, whether the main carrying components meet all requirements in term of carrying capacity and integrity. Simulations were focused on obtaining of values of stresses and deformations after loading by real forces values. There were analysed following components:
- a steel plate carrying the manipulator,
- an adjusting mechanism,
- a console of a RPS pin.

3.1. A steel plate
During designing of the manipulator there was supposed, the steel plate carrying the whole weight of the manipulator will be one of the most loaded element. Therefore, the thickness of the plate has been chosen of 10 mm and it will be made of S235JR steel. Material properties are listed in Tab. 1.

| Property                    | Value            |
|-----------------------------|------------------|
| Density \( \rho \) (kg \( \cdot \) m\(^{-3} \)) | 7,850            |
| Poisson’s coefficient \( \mu \) (-)       | 0.3              |
| Young’s module \( E \) (MPa)       | \( 210 \cdot 10^3 \) |
| Yield of strength \( R_y \) (MPa)     | 235              |
| Ultimate strength \( R_m \) (MPa)     | 360              |

For better accuracy of calculation, there was necessary to define all welded elements. Contact surfaces of welds and welded parts were defined by means of the “Connections” function. Welds and welded surfaces in contacts were defined using the “Bounded” function. It resulted to solid joints of selected elements. On the contrary, contacts between surfaces, which have been just in contact without welded joints, were defined as “Friction” function. It means that they behaved during calculation as surfaces with a friction contact [1, 4, 12, 19]. In this case, it is a steel-steel contact. Boundary conditions have been chosen as following: a rotation coupling has been defined on four surfaces, in which the steel plate in connected with trolleys (Fig. 5). It allows rotating about all axes and also translation movement in the \( x \) direction. The rotation coupling defines a joint coupling the trolley and the plate. The translation in the \( x \) direction defines movement of the trolley on a rail.

The loading force acts on the mounting surface of a bearing in the vertical direction. Its value represents the gravitational force of the device hanging on the plate. The manipulator weight has been determined of 150 kg. The loading force has been rounded on the value of 2,000 N taking into account the gravitational acceleration and the safety coefficient. Obviously, the effect of the device dead weight was taken into the calculation. Figure 6 shows a three-dimensional model of the designed manipulator and its location on the workplace.

After definition of needed boundary conditions, parameters of FE mesh and other parameters, the calculation has been performed. Results of stresses calculated in compliance with the von-Misses theory are shown in Fig. 7 and Fig. 8.

Fig. 5. A three-dimensional model of a trolley travel

Fig. 6. A three-dimensional model of the designed manipulator and its location on the workplace

Fig. 7. Distribution of von-Misses stresses in the steel plate (3D view)

Fig. 8. Distribution of von-Misses stresses in the plate (2D view)
From these figures we can conclude that the maximal calculated value of the stress is of 53.91 MPa. It means that we have chosen the proper construction material as well as its utilization is optimal. Moreover, the safety in term of strength of the designed device is fulfilled.

As it was supposed, maximal values of stresses are reached in locations of welded joints, where they connect the working area of the bearing with the plate.

Stresses have greater values in plate edges. Just for this reason, edges are rounded. Despite of this fact, these locations show higher, however acceptable values of stresses. We suppose to verify visual check of these exposed welds during auditing inspections of the device.

The calculation of deformation has again confirmed the location of the supposed deflection of the steel, which is depicted in Fig. 9. Obtained maximal value of the plate deflection of 3.1 mm does not mean the engineering problem and it is acceptable.

Total deformation in the vertical direction is completely compensated by the lift of the pneumatic piston as well as by setting of end surfaces. Hence, the engineering design of the device has sufficient reserve for setting of the device with regard to these facts.

### 3.2. An adjusting mechanism

Other component, which has been the subject of FE analysis, an adjusting mechanism is (Fig. 10, Fig. 11). It contains a pin joint [3]. This calculation is in comparison with the previous one more difficult mainly in term of degrees of freedom, number of used materials, larger number of welds and contacts between individual components. It is composed from several types of materials.

Welded parts are made of the structural steel S235JR by reason that it is suitable for welding and it has sufficient mechanical properties (Tab. 1). A pin of the adjusting mechanism is made of C45G premium steel C45G, which important properties are introduced in Tab. 2.

![Fig. 10. A three-dimensional model of the manipulator with marking of the adjusting mechanism position](image)

| Table 2. Properties of the premium steel C45G |
|-----------------------------------------------|
| Property | Unit | Value |
| Density $\rho$ ($kg/m^3$) | 7,700 |
| Poisson’s coefficient $\mu$ (-) | 0.3 |
| Young’s module $E$ (MPa) | $200 \times 10^3$ |
| Yield of strength $R_y$ (MPa) | 550 |
| Ultimate strength $R_m$ (MPa) | 880 |

Adjusting screws compose two materials. The screw is made of the steel AISI 12L13 (Tab. 3) and the contact surface is made of polyoxymethylene, which is known as POM material (Tab. 4).

![Fig. 9. Deformation of the plate under the determined load](image)

| Table 3. Properties of the steel AISI 12L13 |
|-------------------------------------------|
| Property | Unit | Value |
| Density $\rho$ ($kg/m^3$) | 7,870 |
| Poisson’s coefficient $\mu$ (-) | 0.29 |
| Young’s module $E$ (MPa) | $200 \times 10^3$ |
| Yield of strength $R_y$ (MPa) | 235 |
| Ultimate strength $R_m$ (MPa) | 395 |

| Table 4. Properties of polyoxymethylene |
|----------------------------------------|
| Property | Unit | Value |
| Density $\rho$ ($kg/m^3$) | 1,420 |
| Poisson’s coefficient $\mu$ (-) | 0.35 |
| Young’s module $E$ (MPa) | 3,000 |
| Yield of strength $R_y$ (MPa) | 70 |

For required accuracy, there was necessary to again define types of contact surfaces between individual elements, e. g. in the pin joint as well as in the contact of plastic surfaces of adjusting screws with the steel surface.
All contact surfaces of the pin were defined as friction surfaces [8-11]. Moreover, a contact between the adjusting screw and a bearing surface was defined as the friction contact. All welds were defined as “Bounded” elements and thus as the solid joint.

Boundary conditions were defined as following: the upper part of the adjusting mechanism (Fig. 11), from which a vertical guiding part of the manipulator arm continues, was defined as fixed joint. The dead weight of the device and the force representing the compressor gravity rounded to the value of 200 N have acted to the rest of the mechanism. The weight of this component is of 3.8 kg. The given force will vary during operation of the mechanism just of the gravitational force value of the compressor, i.e., either it is present or not and that is of 50 N. It is supposed, that the force will increase because of the stroke of the piston causing contact of the arm to the car chassis, whereas this stroke is limited by means of the RPS pin sensor.

Based on performed calculations and display their distributions (Fig. 12, Fig. 13) there is possible to found out, steel parts with holes for pins are the most loaded parts of the analysed component. From distributions we can see, that external edges are subjected to maximal tension stress of 88.44 MPa.

As the yield of stress of the steel S235JR is of 235 MPa, it is possible to assume, that also in this case the requirements of safety will be fulfilled and during operation the material will not be damaged. However, there is recommended visually to verify the actual state of the component within auditing inspections (Fig. 14).

Besides results described above we have found out, that the shearing stress in the pin as well as pressure caused by adjusting screws are negligible.

### 3.3. A console of a RPS pin

The RPS pin (reference point system) is the only one element of the manipulator, which is in the contact with a car chassis during assembly of the compressor to the chassis. The force needed for pulling the manipulator along the assembly line is transmitted by the RPS pin and its console (Fig. 15). Therefore, it is necessary to analyse arose stress in these components.
As well as in the previous cases, there was necessary to prescribed properties of individual contact surfaces. In this case, the friction contact and welded contact were supplemented by prescribing the screw joint behaviour. The friction of type steel-steel was defined between parts connected by the screw joint. The screw was divided into several surfaces. A surface touching down on a washer was defined as friction surface. A surface of an external thread of the screw was fixed with an internal thread of the lower connected part. A surface of the screw shank free-getting through the upper connected part was defined as a friction surface.

In order to reach, that the screw joint will behave as a real joint, preload of the screw joint has to be defined. For the given screw DIN 912, strength category 8.8 and for the dimension of M8, the preload force is of 14.6 kN. Further, all important properties have to be input into the software. The whole structure except of the RPS pin is made of steel S235JR (Tab. 1). The RPS pin is made of steel C45G. It is premium steel (Tab. 2). Screws M8 DIN 912 8.8 have strength parameters introduced in Tab. 5.

Finally, values of acting forces have to be determined. The force, which introduces the manipulator in uniform motion, is calculated and rounded to the value of 400 N. The other force, which is just theoretical, was used for verifying. It is the maximal force, which is produced by the pneumatic piston just in case of RPS pin failure. In such a case, the arm could be pulled by the piston force in the direction against the car chassis. There was necessary to detect, if critical stresses in RPS console are not exceeded.

When the manipulator is loaded by the force needed for its actuation, the greatest stresses are generated just in the screw joint.

The preloaded screw joint causes in the screw the stress of 386 MPa. As it can be identify (Fig. 16, Fig. 17), neither the structure of the console nor welds are too loaded under the considered force and thus, no part of the analysed components is critical loaded.

If the analysed structure is loaded by the force produced by the pneumatic piston, distribution of stresses in the structure is not different significantly (Fig. 17). Extreme values of stresses are again in preloaded screw joints. In comparison with the previous simulation values of stresses are even lower despite of the loading force is of 1,178 N. This difference is caused by the resultant directional vector of the loading force. In the first analysed case, the loading force tends to bend and to crinkle the whole console structure, whereby the screw joint is multiaxially loaded. In the second analysed case, the bending load is dominant;
therefore, the screw joint is loaded uniaxially, i. e. by the tension stress.

4. CONCLUSION

The practical part of the presented problem is based on obtained input data, i. e. requirements, which are demanded from the manipulator in the location of its final position. After studying and consultations, a structural and functional solution was created. The challenge was to design the bearing part of a compressor in compliance with all needed assembly dimensions together with possibility to compensate all inaccuracies. Moreover, there was necessary to take into account factors of safety, health protection and ergonomic parameters. Obviously, all standards and internal regulations of a customer have to be met. Selection of suitable and customers required components from particular providers were another important aspect. There are pneumatic components, control elements, hand-grips, electric components, linear guidance, rail systems etc. The engineering design of the manipulator has to consider not only mechanical functionality of the device, but also positioning of individual pneumatic components. It includes pneumatic cylinders, valves, control buttons, a compressed air treatment unit and other components needed for proper function of the pneumatic system. Except for positioning of individual elements, there was necessary to propose the working principle of the whole pneumatic system including pneumatic control. It contains all pneumatic components and pneumatic logic. These efforts have resulted to the functional three-dimensional model of the device. Last but not least, there were performed numerical analyses of selected components of the device by means of the FEM in term of stress distribution. Numerical calculations have confirmed that analysed components meet all safety requirements. A three-dimensional model of the functional, pneumatic controlled device is the general result of the solved problem. The model is verified by the FE software and after creation of production drawings, the device can be produced and set into operation.

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