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OPTIMIZING IN A DYNAMIC MODE THE CONNECTING ROD FROM AN INTERNAL COMBUSTION ENGINE

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Abstract: *The paper addresses an optimization process of a connecting rod from an internal combustion engine and the research was done in two parts. One part consists of dynamic analysis for the analyzed engine mechanism through a method that combines simplified models from mechanisms dynamics and virtual prototyping with the aid of MSC Adams. Thus, it was established the engine torque variation diagram when the mechanism geometry, material and inertial characteristics are known and gas pressure on the piston head was experimentally obtained. The second part was dedicated to developing an optimization procedure of the engine mechanism connecting rod in a dynamic mode with the use of ANSYS. For this are known the design variables, objective functions and constraints imposed by a structural analysis with finite element method. The obtained results and the parameterized system allow identifying an important premise for an optimal dynamic study of the mechanisms characterized by fast motions.*
Key words: *Dynamic optimization, finite element, design variables objective functions.*

1. INTRODUCTION

Nowadays, optimization analysis procedures become more and more flexible with the aid of computing programs. Even though there are several theoretical procedures for optimization processes but with the aid of computers this becomes more attractive and solves many engineering problems.

In the automotive industry optimization processes gain success which can be obviously seen through the increased performance of automobiles.

By having in sight the internal combustion engines, each part of this mechanism can be optimized to obtain high performances, lightweight, lower manufacturing costs or material improvements as it can be remarked in [1, 2].

The mechanical design of automotive connecting rods is essentially guided by analytic calculations followed by numeric methods to assess the stresses, displacements, contact pressures, fatigue and buckling.

Stresses on the connecting rod are always high due to the combustion chamber pressure, inertia forces, which induces a high value of stresses. But even though, this component can be supposed through optimization processes.

These processes can be developed for weight reduction similar to [3], or to identify a safety factor to which these components can be exposed at high risks. Here it can be mentioned the example of a connecting rod failure when in 2015 Porsche 911 GT3, spent millions of dollars to recall or to replace the entire engine due to the redesign of this component [4, 5].

The optimizing methods pay attention to the stress decreasing and these can be performed in a flexible manner with the aid of the finite element method as the ones reported in [6].

Other optimizing methods were concentrated to the connecting rod geometry and surface improvements.

In [7], theoretical models and finite element analyses were done for connecting rod. By having in sight to optimize the geometry, through the conducted research, the obtained results demonstrate a failure causes at the fillet of connecting rod both ends due to the stress.

Other research like the ones from [8, 9, 10] develops finite element analyses for deformation, fatigue and weight optimization of a connecting rod. These were used ANSYS workbench and the obtained results leads to shape changes.

Another very important aspect, is the fact that this component it is often exposed to high impact forces when combustion stroke occurs. Thus it can be remarked in [11] that there was developed a dynamic model of the impact force, for analyzing this for a multi-body mechanical system. For this research case, it was considered a mechanism from an internal combustion engine with a radial clearance between the piston bolt and connecting rod head. There was acquired important result which leads to optimization procedures development.

By having in sight the related research in this domain, the main objective is to develop a method with a flexible character for optimizing components from an internal combustion engine mechanism. This method will combine theoretical optimization peculiarities with modern software which owns optimization instruments. Thus the research is organized as it follows: the first part it was developed a state-of the art where it can be remarked that the optimization processes were mostly done in case of connecting rods; the second part will allow to present theoretical considerations, which assure the proper tools for virtual simulations in optimization processes; in third part there will be analyzed an internal combustion engine mechanism for calculating the proper torque and connection forces in dynamic conditions; fourth part represents the core of this research where connecting rod optimization process was performed. Thus important results were obtained and at the end conclusions will certify the accomplished objective.

2. THEORETICAL CONSIDERATIONS

To solve optimization problems in the case of mechanical structures, with the use of ANSYS software there where use three types of parameters, respectively design variable, steady variables and optimization function.

For an optimization analysis the design variables are independent. The variables vector has the following form:

$$x = (x_1, x_1 .....x_n) \tag{1}$$

The design variables are supposed to n constraints with upper and lower limits, more over n represents the number of design variables with i=1,…,n.

$$x_{i\min} \leq x_i \leq x_{i\max} \tag{2}$$

The design variable constraints define the achievable design space. Thus, we consider minimizing the following function:

$$f = f(x), \tag{3}$$

with the following constraints:

$$g_i(x) \leq \breve{g}_i \qquad (i = 1, 2, 3, \ldots, m_1) \tag{4}$$

$$h_i \leq h_i(x) \qquad (i = 1, 2, 3, \ldots, m_2) \tag{5}$$

$$\hat{w}_i \leq w_i(x) \leq w_i \quad (i = 1, 2, 3, \ldots, m_3), \tag{6}$$

where: f is the optimization function; g_i, h_i, w_i are state variables that represents the upper and lower limits, $m_1+m_2+m_3$ represents the number of state variables constraints with more values of lower and upper limits.

The state variables can be referred at dependent variables which will get modified once with the x vector of the design variables.

The equations (3) and (6) represent a problem of a minimal constraint which has the target to minimize the optimization function unde the constraints imposed by the equations (2), (4), (5) and (6).

It will be considered a set of design variables as it follows:

$$\overset{*}{\mathrm{x}} = \begin{bmatrix} \overset{*}{x}_1 \overset{*}{x}_2 \overset{*}{x}_3 \overset{*}{x}_n \end{bmatrix} \tag{7}$$

The design will be considered achievable if the following conditions will be accomplished:

$$\overset{*}{g}_i = g_i\left(\overset{*}{x}\right) \le \overline{g}_i + \alpha_i \qquad i = 1,2,3,\ldots,m_1 \tag{8}$$

$$h_i - \beta_i \le \overset{*}{h}_i = h_i\left(\overset{*}{x}\right) \qquad i = 1,2,3,\ldots,m_2 \tag{9}$$

$$w_i - \gamma_i \le \overset{*}{w}_i = w_i\left(\overset{*}{x}\right) \le \overline{w}_i + \gamma_i \qquad i = 1,2,3,\ldots,m_3 \tag{10}$$

Where: α_i, β_i, γ_i represents the admitted tolerances, respectively:

$$\underline{x}_i \le \overset{*}{x}_i \le \overline{x}_i \qquad i = 1,2,3,\ldots,n \tag{11}$$

The equations (8) to (11) represent the defined expressions of the achievable variable sets in case of an optimal design with the use of ANSYS. These mathematical equations were defined by taking into account the fact that it was not introduced any tolerance to the design variable constraints.

3. KINEMATIC AND DYNAMIC MODELS

In this section, it will be analyzed an internal combustion mechanism under kinematic and dynamic aspects. The proposed mechanism has a configuration of a three cylinders engine with the angular position known as the ones reported in figure 1.

By knowing its functional role and also the functional cycle, this mechanism is characterized by fast motions with an accentuated dynamic character. Thus, the crankshaft revolution reaches a value of 6000rpms.

The mechanism dynamic model follows to obtain the engine torque variation law applied to the crankshaft component and obviously the angular speed/revolutions. Theoretically, it will be used the reduced model method when the angular speed can be evaluated with:

$$\omega_1 = \sqrt{\frac{2}{J_{red}}\left[\int_{\varphi_0}^{\varphi} M_{red}\cdot d\varphi + \frac{1}{2}J_0\omega_0^2\right]} \tag{12}$$

Where:

$$M_{red} = \sum_{i=1}^{3}\left(\frac{\overline{F_i}\cdot\overline{V_{c_i}}}{\omega_1} + \frac{\overline{M}_i\cdot\overline{\omega}_i}{\omega_1}\right) \tag{13}$$

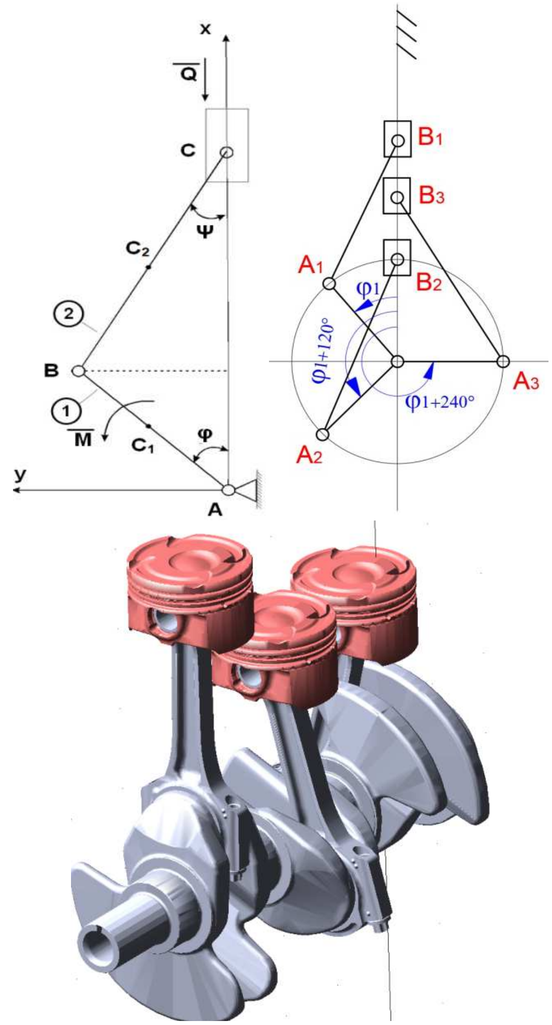

Fig. 1. Engine mechanism kinematic model.

$$J_{red} = \sum_{i=1}^{3}\left(\frac{m_i v_{c_i}^{\ 2}}{\omega_1^2} + \frac{J\Delta c_i\,\omega_i^2}{\omega_1^2}\right) \tag{14}$$

In the equations (13) and (14) can be identified the following terms: $\overline{V_{c_i}}$ - mass center speed of the i element; $\overline{F}_i$ - force which actuates on the i element; $\overline{M}_i$ - torque which actuates on the i element; $\overline{\omega}_i$ - angular speed of the i element; J_{red} – the inertia momentum reduced at the crankshaft level; M_{red} – reduced torque at the crankshaft level; J_0 - initial inertial momentum (when t=0); ω_0 – initial angular speed (when t=0), ω_1 – angular speed of the crankshaft.

By having in sight the proposed objectives in this research, the kinematic and dynamic analyses were performed through virtual prototyping with the aid of MSC Adams. A first step, it was developed a numerical simulation of the engine mechanism in a kinematic mode, considering as input data: the crankshaft revolutions (6000 rpms), geometry characteristics, mass properties and local coordinate systems attached to elements and kinematic joints. Second step it was to create the dynamic model, where it was considered as input data: the gas pressure variation under the pyston head, geometry, inertial and material characteristics for the kinematic links.

In this manner, it was obtained the torque variation law for the crankshaft and the angular speed for the drive joint. Thus, the gas pressure variation law it was obtained through experimental tests and this variation it is presented in diagram from figure 2, depending on crankshaft angular variation.

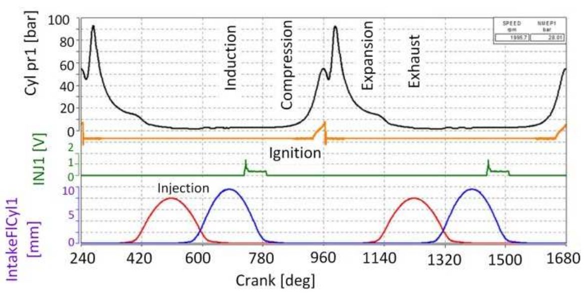

Fig. 2. Dynamic parameters variation vs. crankshaft angular position.

The experimental data were modeled and interpolated in MSC Adams environment as a spline function and the output of the gas force variation on the piston head during time is shown in figure 3.

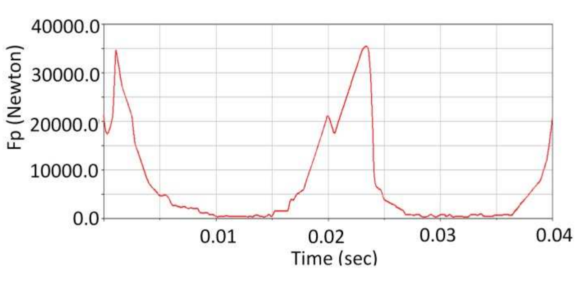

Fig. 3. Force variation under gas pressure on the piston head vs. time.

In this way, it was created a function that allows to the user an interface with the MSC Adams programming environment as it is:

$$M_m = M_0 * \left(1 - WZ\left(\begin{matrix} MARKER\_I, \\ MARKER\_J, MARKER\_J \end{matrix}\right)/\omega_0\right); \quad (15)$$

where: M_m – represents the engine torque; M_0 – initial torque; ω_0 – initial angular speed; ω_z – the obtained angular speed which corresponds to the drive joint.

For the actual case, the engine torque function is:

$$M_m = -170000 * \left(1 - WX\left(\begin{matrix} MARKER\_1, \\ MARKER\_2, MARKER\_2 \end{matrix}\right)/100\right); \quad (16)$$

In figure 4 it can be remarked the crankshaft torque variation during a complete cycle vs. time. Other results were obtained like the reaction components R_{32} from the revolute joint no.:35 equivalents to the connecting rod bushing and piston bolt. These are shown in diagram from figure 5.

Some snapshots during virtual simulations were acquired and presented in figure 6.

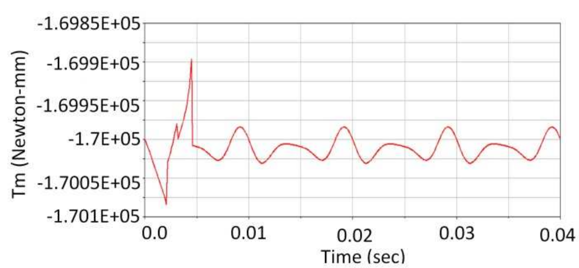

Fig. 4. Crankshaft torque variation during a complete cycle vs. time.

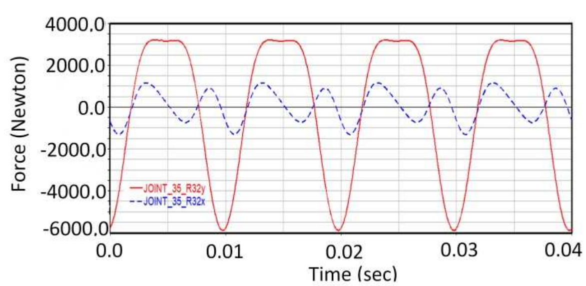

Fig. 5. Reaction force components variation from the revolute joint no:35 vs. time.

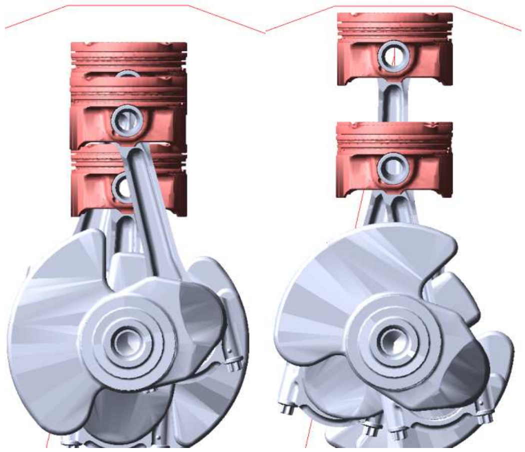

Fig. 6. Snapshots during engine mechanism virtual simulations.

4. CONNECTING ROD DYNAMIC OPTIMIZATION PROCESS

In this second research part, there were considered the geometry characteristics for only two connecting rod elements of the analyzed engine combustion mechanism. These elements are the crankshaft – connecting rod roller bearing and the connecting rod –piston bolt bushing. The 3D connecting rod model was designed in a parameterized format and in this manner, it can be remarked a large area for choosing the proper design variables.

Thus, the parameterized model was analyzed through the finite element method in ANSYS environment where there were considered as input data the following parameters: the applied pressure in a constant mode to the connecting rod bushing as a result of connection force between piston bolt and connecting rod bushing; proper dynamic characteristics of the contact between connecting rod parts (body, connecting rod cap, bolts, roller bearing, bushing).

For this study, the following design variables were defined: P1 variable which corresponds to the inner diameter of the roller bearing; P2 variable which corresponds to the outer diameter of the roller bearing; P3 variable for the inner diameter of the connecting rod bushing; P4 variable for the bushing outer diameter. Also, there were defined two objective functions namely: P5 – maximizing the safety coefficient according to the maximum stress (Safety factor), with a minimum value of the restriction parameter; P6 – minimizing the maximum equivalent stress with a restriction onto maximum value.

Thus the design variables domains, as a numerical form, are: P1 – nominal value of 40 millimeters with a proper domain of 36 to 41 millimeters; P2 – nominal value of 43 millimeters with a proper domain of 41 to 43 millimeters; P3 – nominal value of 20 millimeters with a proper domain of 18 to 21 millimeters; P4 – nominal value of 23 millimeters with a proper domain of 22.5 to 23 millimeters.

After processing the results through finite element method, there were obtained the safety factor and equivalent von Misses stress presented in figure 7. It can be remarked a minimum safety factor of 10.998 and a maximum value of 15. In the case of equivalent von Misses stress, it can be observed a maximum value of 22.731MPa at the level of rod small end.

The obtained results during connecting rod dynamic optimization process are shown in figure 8 to figure 17.

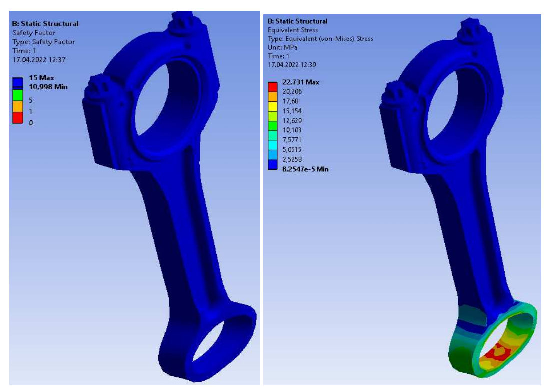

Fig. 7. Objective functions distribution (Safety factor P5 and equivalent von Misses stress P6).

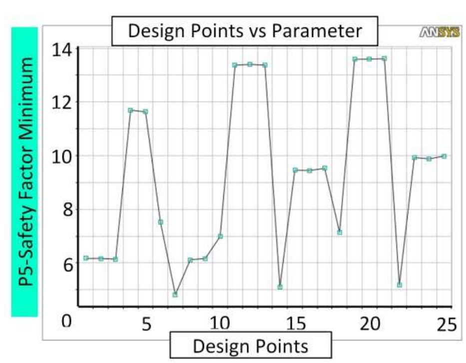

Fig. 8. Safety factor variation vs. design points.

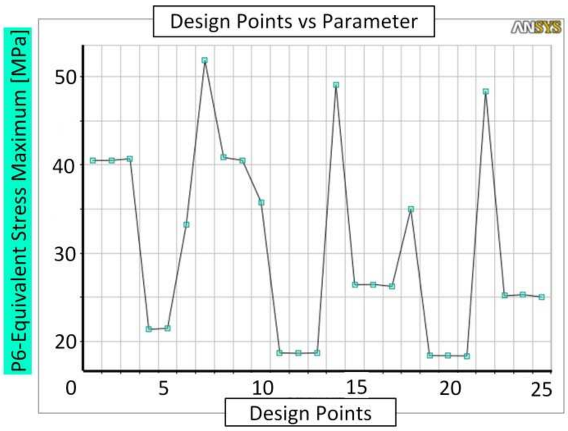

Fig.9. Equivalent maximum von Misses stress vs design points.

From the results reported in the diagram from figure 8 it can be remarked that the safety factor rises proportionally with the inner diameter of the connecting rod roller bearing.

Thus the safety factor reaches a maximum value when the connecting rod roller bearing outer diameter reaches a value of 43 millimeters, according to figure 11.

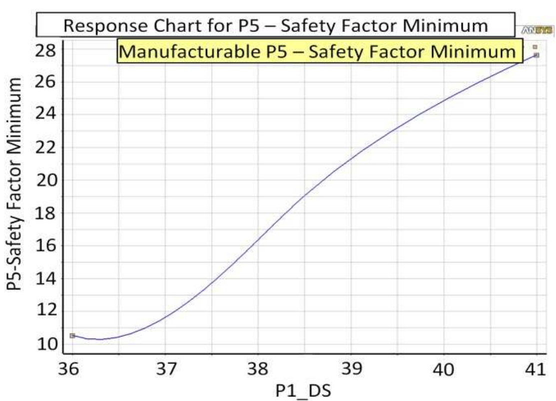

Fig. 10. P5 objective function evolution depends on P1 design variable.

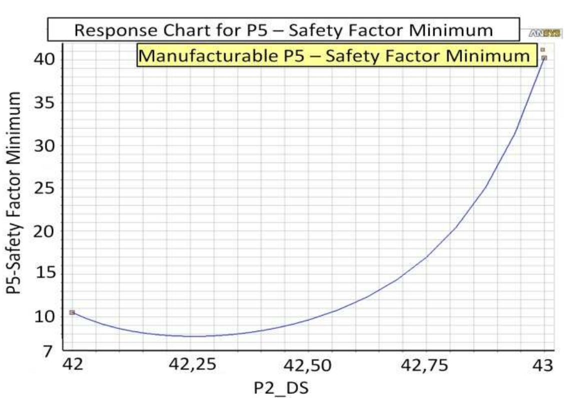

Fig. 11. P5 objective function evolution depends on P2 design variable.

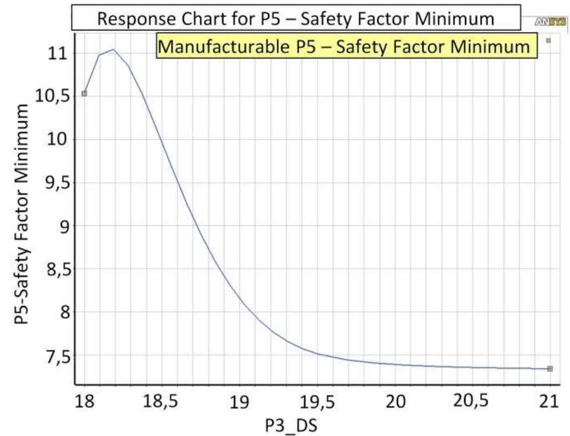

Fig. 12. P5 objective function evolution depends on P3 design variable.

From figure 12 it can be observed that the maximum safety factor will be obtained when the connecting rod bushing inner diameter has a variation between 18 to 18.5 millimeters.

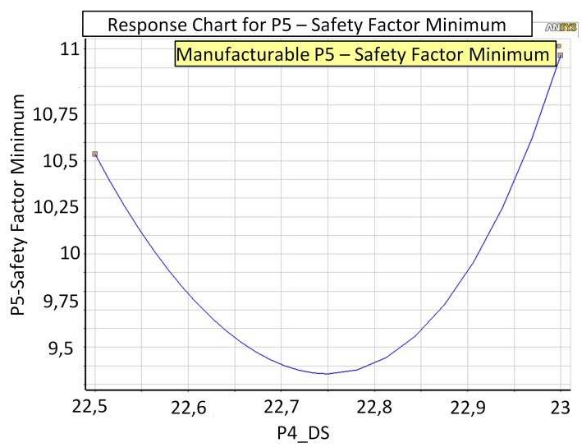

Fig. 13. P5 objective function evolution depends on P4 design variable.

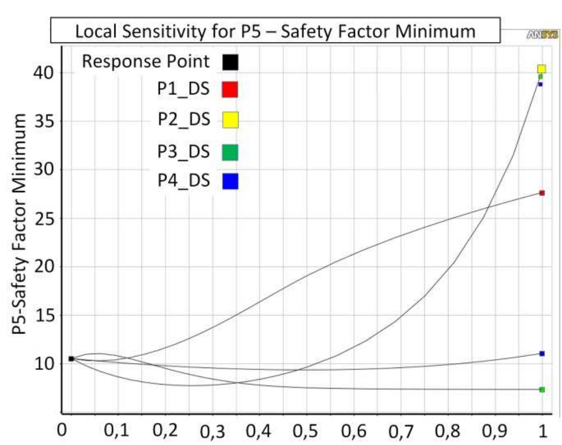

Fig. 14. Design variables influence on the safety factor.

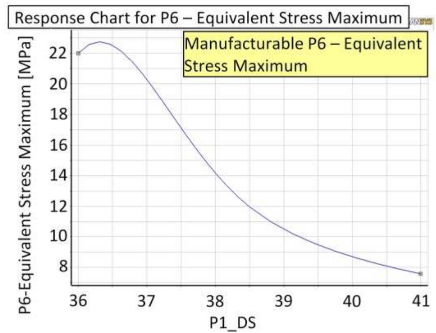

Fig. 15. Maximum equivalent von Misses stress evolution depends on P1 design variable.

Thus, the safety factor danger zone appears when the connecting rod bushing outer diameter has a variation between 22.6 to 22.9 millimeters according to the graph reported in figure 13.

As it is normal, the maximum equivalent von Misses stress gets lower values once with the increase of connecting rod roller bearing inner diameter, according to the reported results in the diagram from figure 15.

More results of the connecting rod optimization process can be found in figures 16 to 18.

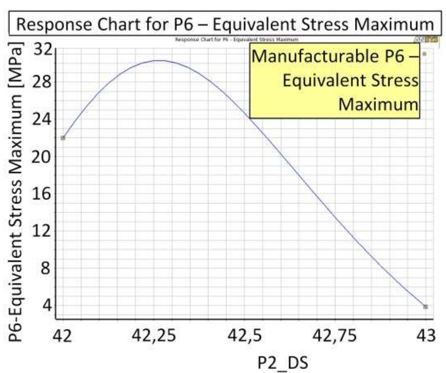

Fig. 16. Maximum equivalent von Misses stress evolution depends on P2 design variable.

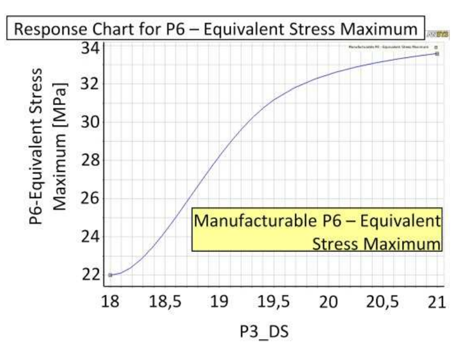

Fig. 17. Maximum equivalent von Misses stress evolution depends on P3 design variable.

The equivalent von Misses stress reaches low values when the connecting rod roller bearing inner diameter reaches low values, respectively 18millimeters according to the plotted graph from figure 16.

According to figure 17 the maximum stress evolution has a parabolic path with a proper maximum which corresponds to connecting rod roller bearing outer diameter equal with 22.75 millimeters and a minimum value when this reach 23 millimeters.

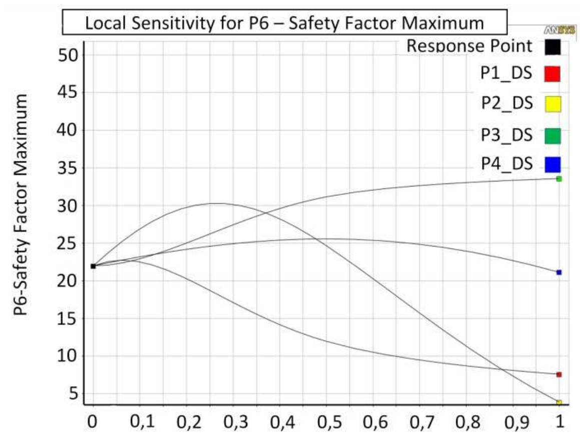

Fig. 18. Design variable influence for the equivalent von Misses stress.

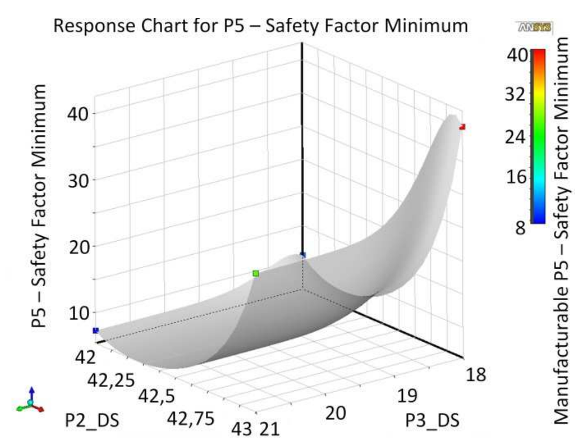

Fig. 19. Evolution of the P5 objective function depends on P2 and P3 design variables.

During this optimization analysis, there were processed several diagrams reported in figure 19 to figure 26. Thus, there were obtained dependencies between defined objective functions and design variables, according with the safety factor extreme values.

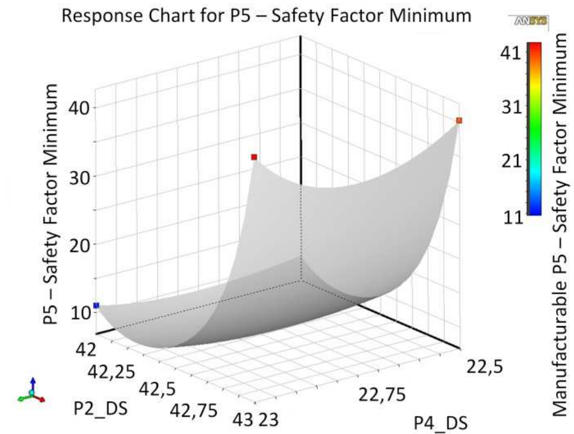

Fig. 20. Evolution of the P5 objective function depends on P2 and P4 design variables.

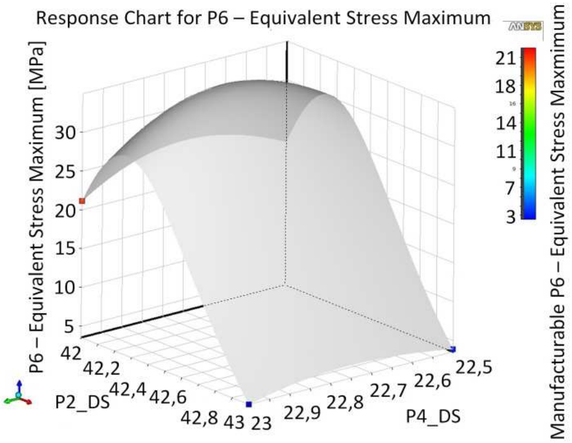

Fig. 21. Evolution of the P6 objective function depends on P2 and P4 design variables.

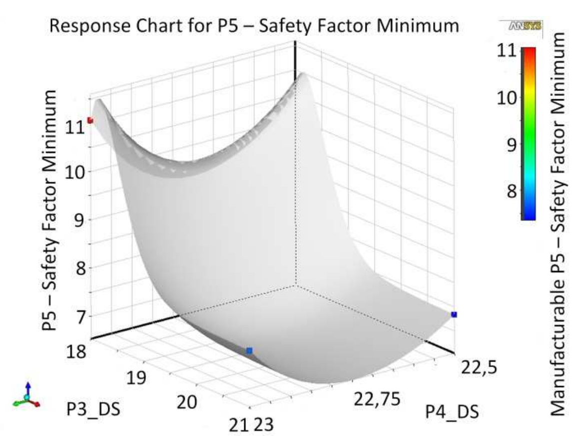

Fig. 22. Evolution of the P5 objective function depends on P3 and P4 design variables.

The optimization analysis was extended to obtain the sensitivity of the objective functions by having in sight the design variables. These are reported in figure 26, where it can be observed that the safety factor become more sensitive during P1 and P2 design parameters and in the case of equivalent von Misses stress, these becomes sensitive at P3 and P4 design variables variations.

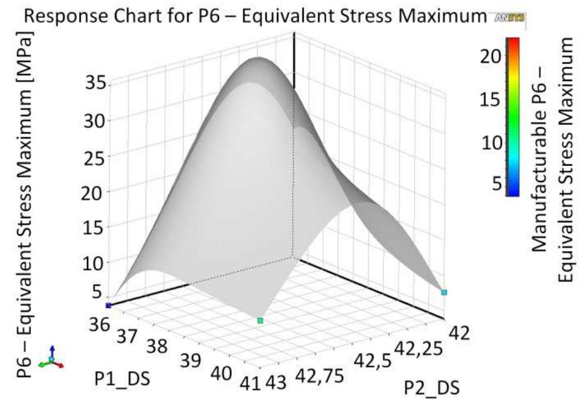

Fig. 23. Evolution of the P6 objective function depends on P1 and P2 design variables.

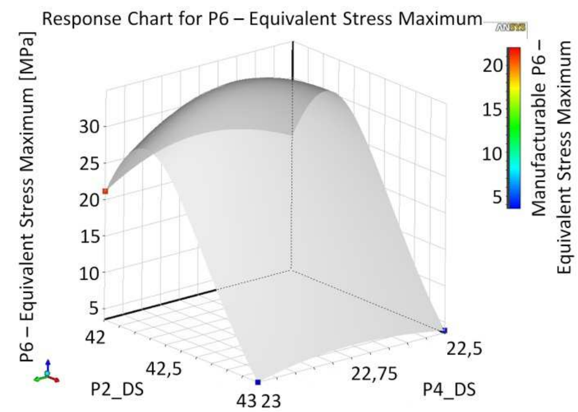

Fig. 24. Evolution of the P6 objective function depends on P2 and P4 design variables.

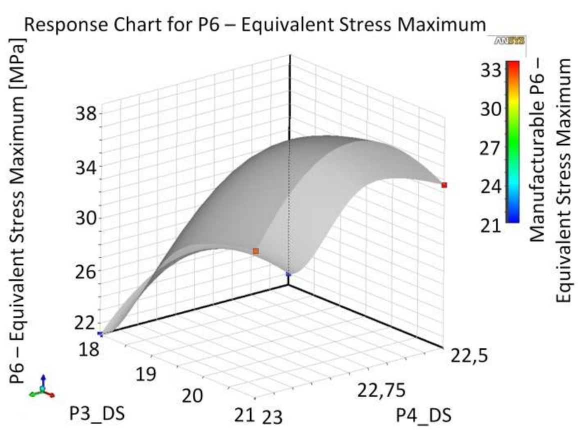

Fig. 25. Evolution of the P6 objective function depends on P3 and P4 design variables.

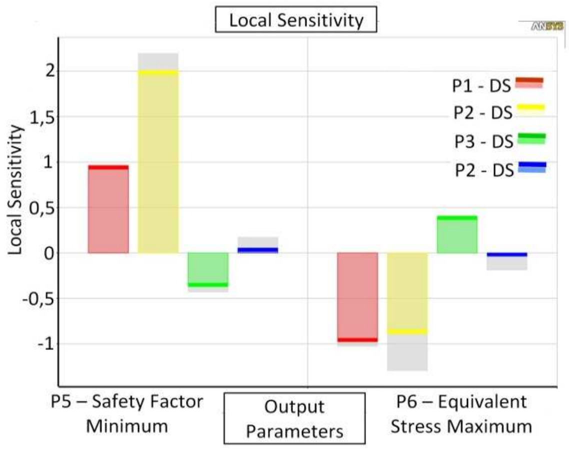

Fig. 26. Objective function sensitivity vs. design variables.

The reported graph from figure 26 allows the user to identify the increase of safety factor and P5 objective function, when this has the restriction P5≥5 MPa. Similarly it can be identified the decrease of safety factor and P6 objective function when this has a restriction of P6≤22,731MPa.

5. CONCLUSIONS

In this research it was achieved an internal combustion engine mechanism dynamic analysis, starting from a mathematical model with reduced models method and through virtual prototyping with MSC Adams software.

Thus, it was established in an analytical mode the equation for identifying and defining the proper engine torque. Based on this mathematical relation, it was identified the proper function for engine torque definition under MSC Adams environment.

By having in sight the engine torque variation, established through virtual prototyping, this oscillates from a numerical viewpoint around the nominal value, which was identified through experimental tests, respectively $T_m=1,7*10^5$[Nmm].

The numerical simulation performed in a dynamic mode with the aid of MSC Adams allows obtaining the connection forces variation law between connecting rod roller bearing and piston bolt.

In this way, it was possible to build a dynamic optimization procedure for the connecting rod of the analyzed engine mechanism. For this it was developed a virtual prototyping process of a parameterized connecting rod. For this, there were defined four design variables, respectively inner and outer diameters of roller bearing and connecting rod bushing. The objective functions consist in maximizing the safety factor at proper maximum stress which occurs insight of the parameterized connecting rod, respectively to minimize the equivalent stress.

It was analyzed the influence of each design variable on objective functions, represented through 2D diagrams. Also, this analysis allows obtaining the cumulated influence of two design variable above each objective function, represented through 3D diagrams.

The optimization procedure was achieved based on finite element method. There were used tetrahedral finite elements for the connecting rod body, by having in sight the manufacturing process based on powders metallurgy. In the case of the connecting rod roller bearing and bushing component, these were analyzed by taking into account proper material definition for aluminum alloys.

The presented procedure through this research frame has a flexible character and the geometrical model was a parameterized one for all connecting rod dimensions. Thus, this procedure can be extended for different objective functions, which include different material properties, natural frequencies and much more, with the respect of several restrictions.

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OPTIMIZAREA IN REGIM DINAMIC A BIELEI MECANISMULUI DE LA UN MOTOR CU ARDERE INTERNA

Rezumat: Lucrarea este structurata in doua parti. O parte care contine studiul dinamic al mecanismului printr-o metoda care combina modele reduse folosite in dinamica mecanismelor si prototivarea virtuala cu programul MSC Adams. Se determina diagrama de variatie a momentului motor cand se cunosc, geometria mecanismului, caracteristicile inertiale si de material si distributia presiunii gazelor pe capul pistonului, care a fost stabilita experimental. O alta parte, in care se dezvolta o procedura de optimizare in regim dinamic a bielei mecanismului motor, cu programul Ansys, in care se studiaza variabilele de proiectare, functiile obiectiv si eventuale restrictii impuse de analiza structurala cu metoda elementului finit. Sistemul parametrizat de lucru si Rezultatele obtinute creaza o premiza importanta pentru studiul dinamic optimal al mecanismelor cu miscari rapide.

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