Analytical study regarding the topological optimization of an internal combustion engine piston

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Abstract. Continuous progress in the automotive field involves the use of new working methods in the calculation and optimization of the internal combustion engine parts. On this occasion, in this paper, the impact of a topological optimization on the piston in the combustion engine of the conventional or hybrid electric car was studied. Starting from the simulation of the Otto cycle, the dimensional calculation is reached, after which a classical FEA analysis is carried out in order to establish the reference values. In the last part, the data of the classical analysis are compared with those of the topological analysis in order to perform a comparison of their results. The purpose of this study is to reduce the volume of the optimized part and to increase its rigidity.

1. Introduction

For the thermal calculation we will choose as input parameters the following values: piston bore $B = 70$ [mm], piston stroke $S = 71$ [mm] and nominal speed $n = 6000$ [rpm]. Using these input data, it was possible to generate both the engine operating cycle (figure 1) and the graph of the pressure arrangement in the cylinder depending on the angle between the crank and the vertical axis $\alpha$, $p = f(\alpha)$ (figure 2).

In these diagrams, the variation of the pressure inside the cylinder can be observed. The region of interest is in the upper half of the graph, where the pressure is at its maximum value. In order to be able to measure precisely the maximum numerical value of the pressure, the graph $p = f(\alpha)$ is generated, where it will be easy to observe its variation throughout the engine cycle.

2. Engine workload simulation

The maximum value of the pressure in the engine cycle is necessary to be able to determine the force generated by this pressure, that is, the pressure force of the gases in the cylinder $F_p$. This force can be calculated using the following formula:

$$F_p = \frac{\pi B^2}{4} (p - p_{\text{cart}})$$

(1)

where $B$ is the bore of the piston, $p$ is the gas pressure in the cylinder, which is determined graphically, and $p_{\text{cart}}$ is the pressure of the gas in the engine bay, which works on the inside of the piston head.

This usually has a value of 0.1 [MPa] [1]. Once known the calculation formula and having all the input parameters, the calculated value of this force is:
Knowing the value of the force $F_p$, it can be continued with finding the other component of the resulting force, that is, the inertia force of the masses in translation $F_i$. This force can be calculated using the following equation:

$$F_i = -m_p j_p,$$

where $m_p$ is the sum of the masses of the piston, the segments and the bolt and $j_p$ is the acceleration of the piston.

In this case, the sum of the piston masses $m_p$ is equal to 0.4 [kg] and the piston’s acceleration is determined by the following formula:

$$j_p = r \omega^2(\cos \alpha + \Lambda \cos 2\alpha),$$

where $r$ is the radius of the crank, $\omega$ is the angular velocity, and $\Lambda$ is the ratio between the crank length and the connecting rod length. In general, for cars, $\Lambda$ has a ratio between $1/3...1/3.8$ [1]. Once replaced all the parameters in the formula, the value of the inertia force will be the following:

$$F_i = -5487 \text{ [N]},$$

corresponding to the maximum gas pressure force.

By summing the numerical values of the two forces, the maximum value of the resulting force is obtained, $F_{R \text{ MAX}}$, at $\alpha = 370^\circ$ RAC.

The resulting force will be decomposed into two components, one will be in the direction of the axle of the connecting rod, and the other will be in the direction of the normal axis of the cylinder.

$$F_{R \text{ MAX}} = F_p + F_i = 26707 - 5487 = 21220 \text{[N]}$$

$$F_p = \pi \frac{702}{4}(7.0399 - 0.1) = 26707 \text{[N]}$$

**Figure 1.** Simulation of the engine’s Otto cycle  
**Figure 2.** Pressure - crank angle diagram
The force distributed in the axis of the connecting rod will have the following relation:
\[ F_B = \frac{F_{\text{R_MAX}}(\cos \beta)}{\sqrt{\cos \beta}} \]  
(7)

Consequently, the normal force at the axis of the cylinder is:
\[ F_N = \frac{F_{\text{R_MAX}}}{\tan \beta} \]  
(8)

where \( \beta \) is the angle between the axis of the connecting rod and the vertical axis of the piston.

Using:
\[ \cos \beta = \frac{1 - \Lambda \sin \alpha}{\Lambda} \]  
(9)

For \( \alpha = 370 ^\circ \) RAC, at which the forces reach the maximum, the following values result:
\[ F_B = 21616.2 \text{ [N]} \]  
(10)
\[ F_N = 4115.25 \text{ [N]} \]  
(11)

These two forces serve as input data for the analysis part of this paper.

3. Analysis and interpretation of results

In this study, the aim is to topologically optimize a piston used in an internal combustion engine. In this case, the topological optimization is represented by a set of calculations that determines the most optimal shape of the piece, according to the constraints imposed [3]. The calculation relationships underlying this process aim to determine the values of the minimum stresses, the minimum deformations, the minimum displacements, the minimum mass as well as the minimum volume of the part. Based on a set of calculations that uses genetic algorithms, optimal criterion method, level analysis and topological derivatives, in addition to the dedicated software programs, this optimization process also needs a computer with powerful hardware resources. The software used for this analysis is AUTODESK FUSION 360, specifically, the Shape Optimization module.

In the experimental analysis approached, the piston will have the geometry from figure 3.

After defining the geometry, the second step consists in adopting the material for manufacturing the piston. It requires a material that can withstand the amplitude of the forces to which it is subjected and also has high machinability. For this case, the Al7075 aluminum alloy was chosen. As the main element of zinc alloy, this alloy is a resistant one, comparable to many steel alloys; it has excellent fatigue resistance and average workability. In the composition of this alloy there are the following elements: between 5.6% and 6.1% Zn, between 2.1% and 2.5% Mg, between 1.2% and 1.6% Cu and below 0.5% Si, Fe, Ti, Cr. The mechanical properties of this material depend on whether it was subjected to heat treatments or not. The untreated 7075 has a maximum tensile strength of 276 [MPa] and an elasticity limit of 145 [MPa]. The maximum elongation of the material is between 9% and 10%. The 7000 series alloys are mainly used in the automotive, marine and aeronautical industries due to the good resistance and density ratio [2].

Once the material used in the analysis is defined, a static finite element analysis was performed on the piston with the initial geometry; this facilitates the establishment of the reference values for the comparative analysis. Also, for the accuracy of the results the active surface of the piston was exposed at an average temperature of 200 ° [C], according to the recommendations in the literature [1].

The initial results of the model are presented through figure 4, figure 5, figure 6, figure 7 and figure 8:
Figure 3. The mesh of the analyzed piston

Figure 4. Safety factor

Figure 5. Von Mises stress

Figure 6. Displacement

Figure 7. Equivalent strain

Figure 8. Thermal gradient
They highlight the following values, summarized in table 1 [5]:

Table 1. The highlight values of static finite element analysis for initial piston geometry

| Minimum safety factor | Maximum Von Mises stress [MPa] | Maximum displacement [mm] | Maximum equivalent strain [mm] | Thermal gradient [°C/mm] |
|-----------------------|--------------------------------|---------------------------|-------------------------------|-------------------------|
| 0.1228                | 1685                           | 0.1417                    | 0.01312                       | 0.002961                |

It is mentioned that the thermal gradient represents the dimensional quantity, expressed in [°C/mm], of units of degrees per unit of length. It describes in what direction and at what pace the temperature changes around a specific location.

After establishing the results of the parameters of interest, the next step is to prepare the topological optimization process. The desired target mass is defined, and for this study, a reduction of 1% from the initial mass is imposed [4]. The target rigidity is defined below. In this case, it was chosen to maintain 100% rigidity. In some cases, when the target mass is too small to maintain an imposed level of rigidity, within the program a warning message is being received. This fact enables the possibility to either change the target mass or to relax the rigidity condition.

The final results of the optimization analysis are shown in figure 9 and figure 10.

![Figure 9. Isometric view of the topological optimized piston](image1)

![Figure 10. Isometric view of the topological optimized piston, a second plane](image2)

![Figure 11. Isometric view of the final geometry of the topological optimized piston](image3)

![Figure 12. Isometric view of the final geometry of the topological optimized piston, a second plane](image4)
The highlighted green surfaces can be observed, indicating the areas of the piston where the mass was reduced. The final geometry of the optimized piston can be seen in figure 11 and figure 12. In the next phase, a finite element analysis was performed on the optimized piston. The obtained values are being compared with the initial results.

The results of the optimized model analysis are presented in figure 13, figure 14, figure 15, figure 16, figure 17 and they lead to the following values contained in table 2.

![Figure 13. Von Mises stress of the topological optimized piston](image1)

![Figure 14. Safety factor of the topological optimized piston](image2)

![Figure 15. Equivalent strain of the topological optimized piston](image3)

![Figure 16. Displacement of the topological optimized piston](image4)

![Figure 17. Thermal gradient of the topological optimized piston](image5)
Table 2. The highlight values of static finite element analysis for optimized piston geometry

|                | Minimum safety factor | Maximum Von Mises stress [MPa] | Maximum displacement [mm] | Maximum equivalent strain [mm] | Thermal gradient [°C/mm] |
|----------------|-----------------------|-------------------------------|---------------------------|-------------------------------|-------------------------|
|                | 0.1101                | 1396                          | 0.2517                    | 0.03238                       | 0.0001658               |

At the same time, the obtained results are presented comparatively, using the diagrams in figure 18, figure 19, figure 20.

Figure 18. Comparative representation of parameters: safety factor and maximum total displacement, before and after topological optimization

Figure 19. Comparative representation of the parameters: maximum equivalent deformation and thermal gradient, before and after topological optimization

Figure 20. Comparative representation of the parameter: Von Mises stress, before and after topological optimization

Based on these results, concerning the reference values, the percentage differences are obtained. Regarding the piston characteristics, the updated geometry is validated yet again by the finite element analysis.
They are highlighted in the diagram from figure 21:

![Diagram showing comparative representation of the optimized parameters.]

**Figure 21.** Comparative representation of the optimized parameters: the minimum safety factor, the maximum total displacement, the maximum total deformation, the maximum Von Mises stress and the thermal gradient, relative to the reference values.

In figure 22 is presented the topologically optimized piston model.

![Topologically optimized piston model.]

**Figure 22.** The topologically optimized piston model is presented.
4. Conclusions
Comparing the results of the two finite element analyses, we can conclude the following:

- The reduction of the piston mass by 1% is plausible, keeping the initial rigidity;
- In the comparative representation of figure 21, it can be observed the 17% decrease of the equivalent Von Mises stress, this means that the piston is less stressed during operation;
- The maximum deformations and displacements increased as a percentage, but numerically, the values are in the optimal range of values;
- For the feasibility of a possible topology optimization, the economic side of this problem must also be considered. Because this analysis involves specialized equipment and software solutions, which are expensive, the possible gain from the optimization must be very well estimated, before starting all the steps required by this;
- Possible gains from a quality analysis result in a consistent reduction in the production price, regardless of the nature of the optimized part.

5. References
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