Mechanical and thermal analysis of the internal combustion engine piston using Ansys

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Abstract. The piston is one of the most important components of the internal combustion engine. Piston fail mainly due to mechanical stresses and thermal stresses. In this paper is determined by using the finite element method, stress and displacement distribution due the flue gas pressure and temperature, separately and combined. The FEA is performed by CAD and CAE software. The results are compared with those obtained by the analytical method and conclusions have been drawn.

1. Introduction

The piston is an important component of the internal combustion engines, due to the role it fulfils, but also due to the complex loads which act upon it. Upon the piston act both thermal and mechanical loads. The piston’s resistance to the thermal and mechanical loads limits the power of the internal combustion engine.

For this reason, during the design stage of the piston, there is a great emphasis on determining the stresses and deformations due to the mechanical and thermal loads, an aspect which is revealed in numerous scientific papers dealing with this issues. Studying these papers [1 – 6], show that the authors have used CAD software for 3D modeling of the piston, and CAE software to determine the stresses and deformations in the piston, which allow finite element analysis and the study of the product behavior in terms of mechanical-thermal loads.

By using the finite element method, this paper determines the stresses and deformations of the piston subjected to mechanical loads (flue gas pressure) and thermal loads (flue gas temperature), considering that they act independently and simultaneously (mechanical – thermal load).

The results are compared with those obtained by the analytical method and conclusions are drawn.

Autodesk Inventor Professional is used to obtain the 3D model of the piston and Ansys Workbench V.15 is used for the finite element analysis.

2. Initial data

This study considers a flat head piston, with two compression segments and a lubrication segment. In order to prevent the heat flow orientation from the piston head to the first segment, the canal of the fire segment is placed below the bottom of the piston. To the same end, the port-segment area and the bottom of the piston are broadly joined up. The length of the piston’s mantle must be enough in order to provide a good guidance, low side pressure and to limit the tilting.
Based on the initial data and recommendations from literature [7], [8], we established the main dimensions of the piston, after which we built the 3D model (Figure 1, a and b).

The material used for the piston is a silicon based aluminum alloy. Eutectic alloy type ATC12CuMgNi, having the properties on table 1, will be used.

Table 1. Properties of the material

| Properties of the material       | Value               |
|----------------------------------|---------------------|
| Thermal conductivity            | Figure 2            |
| Thermal Expansion               | $2.3 \times 10^{-5}$ $\degree/C$ |
| Specific Heat                   | 875 J/kg$\degree/C$|
| Density                         | 2770 kg/m$^3$       |
| Young’s Modulus                 | $7.1 \times 10^4$ MPa |
| Poisson’s Ratio                 | 0.34                |
| Tensile Yield Strength          | 280 MPa             |
| Tensile Ultimate Strength       | 310 a               |

3. Analytical calculation of the piston

The piston head is the most used area, in thermal and mechanical terms, as it comes into contact with the flue gases and takes over pressure forces at the same time. For this reason, the analytical calculation of the stresses from the piston head, which arise due to this loads, is used.

3.1. Analytical calculation of the mechanical stresses in the piston head

Mechanical stresses ($\sigma_{mec}$) were calculated analytically, using the formulas and terms described in [7], [8]. The piston head was considered as a circular plate embedded to the contour given by the inner diameter of the piston head, having a constant thickness and a uniformly distributed load, given by the maximum pressure of the gas.

In such plate, radial stresses ($\sigma_r^m$) and tangential stresses ($\sigma_t^m$) appear, whose extreme values are obtained in the center of the plate (c) and in the embedment (m), on the bottom surfaces (b) and the top surfaces (t).
3.2. Analytical calculation of the thermal stresses in the piston head
Thermal stresses ($\sigma_{\text{ter}}$) are generated by the uneven distribution of temperatures in the piston head and by the way it expands free or restricted. In the case of a flat piston, the maximum temperature is reached in the center of the piston and the temperature decreases towards the extremities.

The thermal stress of the head piston was established as follows: the stresses ($\sigma_{r3}^f$) and ($\sigma_{t3}^f$) were determined, considering the piston head as a circular plate, free on the contour, with constant thickness, subjected to an axially symmetric temperature field. It was taken into account that the side wall of the piston prevents the free expansion of the head, creating a pressure on the surface of the piston, which causes radial stresses ($\sigma_{r2}^f$) and tangential stresses ($\sigma_{t2}^f$). Considering a linear temperature variation on the thickness of the piston head, there are bending stresses on the radial ($\sigma_{r3}^f$) and tangential ($\sigma_{t3}^f$) direction, equal between them. Tangential and radial thermal stresses are determined by superposition of effects, obtaining $\sigma_f^t$ and $\sigma_f^r$.

3.3. Analytical calculation of mechanical-thermal stresses in the piston head
Mechanical-thermal stresses ($\sigma_{\text{mec–ter}}$) that appear in the piston head $\sigma_r$ and $\sigma_t$ are also determined by applying the principle of superposition effects.

| Table 2. Results of the analytically calculus, in MPa |
|-------------------------------------------------|
| Type of Stress | Position | Value 1 | Value 2 |
|----------------|----------|---------|---------|
| Mechanical Stress $\sigma_{\text{mec}}$ | Radial $\sigma_r^m$ | (t) | 63.95 | -42.84 |
| | (b) | -63.95 | 42.84 |
| | Tangential $\sigma_t^m$ | (t) | 21.74 | -42.84 |
| | (b) | -21.74 | 42.84 |
| Thermal Stress $\sigma_{\text{ter}}$ | Radial $\sigma_r^t$ | (t) | -15.63 | -25.21 |
| | (b) | -7.11 | -16.69 |
| | Tangential $\sigma_t^t$ | (t) | 17.11 | -25.21 |
| | (b) | 26.93 | -16.69 |
| Mechanical – Thermal Stress $\sigma_{\text{mec–ter}}$ | Radial $\sigma_r$ | (t) | 48.32 | -68.05 |
| | (b) | -71.06 | 26.15 |
| | Tangential $\sigma_t$ | (t) | 38.85 | 68.05 |
| | (b) | 5.25 | 33.94 |
| Equivalent Stress | $\sigma_{\text{equiv M2S}}$ (Tresca) | (t) | 48.32 | 68.05 |
| | (b) | 76.31 | 33.94 |
| | $\sigma_{\text{equiv Mises}}$ (Mises) | (t) | 44.34 | 68.05 |
| | (b) | 73.82 | 30.79 |
In the design calculation, it is recommended to use equivalent stress, determined using a theory of resistance. Due to the fact that in the center of the plate, as well as in the marginal section, there is a state of two dimensional stress space, the maximum shear stress theory can be used. If tensions $\sigma_p$ si $\sigma_q$ have the same sign, the maximum stress is considered in an absolute value and it is compared to the allowable stress. If the two stresses have different signs, the sum of the absolute values will be calculated and compared to the allowable stress. At the same time, the equivalent stress calculated using von Mises theory customized for plane stress state, can be used.

![Figure 3. Notations used in paper](image)

The results obtained through the calculation are shown in Table 2, using the following notations: (t) – top surface of the piston head, (b) – bottom surface of the piston head, (m) – marginal, (c) – center of the piston head (Figure 3), $\sigma_{echiv \, MSS}$ - the maximum shear stress and $\sigma_{echiv \, Mises}$ - von Mises stress.

4. Finite element analysis of the piston

4.1. Finite element analysis of the piston subjected to mechanical loads

To perform the finite element analysis of the piston when upon it acts the pressure of the gasses, a structural analysis using Ansys Workbench V.15.1 takes place.

At this stage, the analysis of the piston is a linear static one, when small changes in rigidity happen, there are no changes in the direction of the loading, the materials remain within linear elastic area and small deformations and stresses are produced.

The model of the piston is made in Autodesk Inventor and saved in this program as *.sat, and then imported in Ansys Workbench.

![Figure 4. Radial stress due to mechanical loads – in section](image)

![Figure 5. Radial stress due to mechanical loads – in 3D model](image)
The pressure of the gasses is the main mechanical stress which the piston is being subjected to, which in this paper is considered to be equal to 6.5 MPa, normally applied on the top surface of the piston head. Also, all the degrees of freedom of the bores for the bolt are bound. The results are obtained as stresses fields and deformations fields. For the present situation, due to the fact that the radial and tangential normal stresses in the piston head were determined analytically, the value fields of these dimensions were displayed in the areas of interest.

Figure 6. Tangential stress due to mechanical loads – in section

Figure 7. Tangential stress due to mechanical loads – in 3D model

Figure 8. Total deformation due to mechanical loads – in section

Figure 9. Total deformation due to mechanical loads – in 3D model

The field of the normal radial stress is shown in Figures 4 and 5, as follows: in section through the axis of the piston, with a perpendicular plane on the axis of the bore for the bolt (Figure 4) and in ¼ section (Figure 5). The normal tangential stresses are show in the same manner in Figures 6 and 7. The total directional deformations are shown in section in Figure 8 and in ¼ section in Figure 9.

4.2. Finite element analysis of the piston subjected to thermal loads

The designed piston operates in high temperatures conditions, amounted to 200…250°C. Hereinafter, we present the results obtained from the thermal analysis of the piston, in order to determine the piston temperature field in the piston body and thermal stresses due to different temperatures in the piston body.
The analysis is a steady-state thermal one. Upon the piston acts a heat flux (291×10³ W/m²), which is applied to the upper surface of the piston head. The heat transfer between the piston and the oil film between the piston and the cylinder is made by convection, case which requires a convection heat transfer coefficient equal to 2330 W/m²·°C, at a temperature of 90°C, and between the inner surfaces of the piston and oil, a coefficient equal to 750 W/m²·°C, having the same temperature as the oil. The results are shown in the following figures. Thus, the temperature field in the piston body is shown in Figures 11 and 12, noticing that the thermal flow determines a maximum temperature on the top surface of the piston head, which decreases towards its edge.

Thermal load, as well as mechanical load, determine thermal stresses and deformations inside the piston body. These can be determined by performing a static analysis using the thermal conditions of the performed steady analysis.

As a constraint, the binding of the degrees of freedom of the bores for bolt of the piston is required, and the results are obtained as thermal stresses and deformations.

The results are shown in the following figure, as follows: the ones related to normal radial stresses are shown in Figures 13 and 14, the ones related to tangential normal stresses are shown in Figures 15 and 16, and the ones related to total deformations in the set of Figures 17 and 18.
Figure 13. Radial stress due to thermal loads – in section

Figure 14. Radial stress due to thermal loads – in 3D model

Figure 15. Tangential stress due to thermal loads – in section

Figure 16. Tangential stress due to thermal loads – in 3D model

Figure 17. Total deformation due to thermal loads – in section

Figure 18. Total deformation due to thermal loads – in 3D model
Figure 19. Radial stress due to mechanical-thermal loads – in section

Figure 20. Radial stress due to mechanical-thermal loads – in 3D model

Figure 21. Tangential stress due to mechanical-thermal loads – in section

Figure 22. Tangential stress due to mechanical-thermal loads – in 3D model

Figure 23. Total deformation due to mechanical-thermal loads – in section

Figure 24. Total deformation due to mechanical-thermal loads – in 3D model
4.3. **Finite element analysis of the piston subjected to combined loads: mechanical and thermal**

For this analysis, in addition to the thermal analysis conditions, it is considered that a pressure equal to the maximum gas pressure (6.5 MPa) acts on the top surface of the piston head acts.

The results obtained emphasize the influence of both types of loads on the stresses and deformations which appear in the considered piston body. The radial stresses are shown in Figures 19 and 20, tangential stresses are shown in Figures 21 and 22, and Figures 23 and 24 display the total deformations of the analyzed piston.

![Figure 25. Maximum Shear Stress - in section](image1)

![Figure 26. Maximum Shear Stress - in 3D model](image2)

![Figure 27. Von Mises Stress - in section](image3)

![Figure 28. Von Mises Stress - in 3D model](image4)

Equivalent stresses are also determined (Figures 25, 26), calculated using the third theory of resistance (Maximum Shear Stress theory), recommended theory in the case of a plane state of stress, respectively using the 4th theory (von Mises), customized for the plane state of stress (Figures 27, 28).

5. **Analysis of the results**

Analyzing Figures 4…7, we notice that radial and tangential stresses caused by mechanical loads in the interest areas (at the edge of the piston head and in the center of the piston head, on the top and bottom surfaces), obtained through finite element analysis, have similar values as the ones obtained through analytical calculation method. Simultaneously, we can note that the obtained stresses are similar in terms of signs.

Figures 8 and 9 show that the maximum total deformation of the piston head caused by mechanical load is 0,09 mm and it is positioned in the center of the piston head.
The field of temperature in the piston is shown in Figure 11. It is noticed that the maximum value of the temperature is in the center of the piston head and it decreases towards the edge of the piston head, the difference between the center and the edge of the piston head being of $45\ldots50^\circ\text{C}$, and the difference between the top surface of the piston and the bottom surface being around $8\ldots10^\circ\text{C}$, which confirms both the analytical calculation and data from the literature [7], [8].

In the Figures 13…16, it is noticed that both radial and tangential stresses have, on the top and bottom surfaces, values and signs corresponding to the results obtained through the analytical calculation. The total deformations caused by thermal loads (Figures 17 and 18) are maximum on the top surface of the piston head ($0.164 \text{ mm}$), on the edge, and have the opposite deformations sense due to mechanical load.

Regarding the mechanical-thermal load, the radial and tangential stresses (Figures 19 – 22) correspond in terms of values and signs with those determined analytically.

In the case of this complex load, the equivalent stresses are of interest, because in the design stage these stresses are compared to the allowable stress of the material the piston is made of.

It is noted that the values of the equivalent stresses calculated with the theory of the maximum shear stresses (Figures 25 and 26), in the piston head, do not exceed 55 MPa, a value much lower than the allowable stress of the material (80…120 MPa).

For the same purpose it is important to determine the von Mises equivalent stresses (Figures 27 and 28). Also, it is noticed that in the piston head, the von Mises equivalent stresses do not exceed the allowable stress of the material.

Analyzing Figures 25…26 and table no. 3, we can see that the values of the equivalent stresses calculated using the finite element method are very similar to the ones calculated using the analytical method. Also, it can be stated that both types of equivalent stresses can be used for the designing of the piston, their values being sensibly equal, and the distribution is identical.

The maximum total deformation of the piston due to mechanical-thermal load is lower than the maximum total deformation due to thermal load, it is found at the edge of the top surface of the piston head and has the same sense as the deformation due to thermal load (Figure 29). It can be concluded that the deformation due to mechanical load cancels in part the deformation due to thermal load.

![Figure 29](image)

**Figure 29.** Total deformation: (a) – due to mechanical loads, (b) – due to thermal loads, (c) – due to mechanical - thermal loads

6. Conclusions

Due to the constructive complexity and complex mechanical and thermal loads the piston is subjected to, the accurate determination of the unitary stresses and deformations is required. The classical methods of calculation of the resistance of the piston (determination by calculation of the unitary stresses) are based on simplified representations of the geometric shape and loads.

Using the finite element analysis for the calculation of the unitary stresses and deformations, enables the detailed analysis of every area of the analyzed structure, under the conditions of obtaining...
an adequate precision of the results, results which can be successfully used in the stage of the optimization of the piston’s shape.

By using CAD and CAE software in designing parts/assemblies of parts, the designer has the opportunity to analyze various constructive variants in a short time, thereby optimizing the designing work.

Regarding the results obtained using the two designing methods, it is noted that the obtained values are similar in the case of stresses, which indicates the fact that the three dimensional model, the loads and the constraints have been properly defined, and the finite element analysis is recommended for use.

By using the finite element method, stresses and deformations are obtained relatively quickly and important and complex information can be achieved in terms of the behavior of the piston during operation.

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