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ANALYSIS OF THERMAL AND MECHANICAL STRESSES OF RENAULT PREMIUM DXI11 460 EEV FOUR-STROKE PISTON

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Abstract

In this article the engine piston Renault Premium DXi11 430 460 EEV has been analysed using the Finite Element Method. Analysis consider as well heat transfer phenomenon as the thermal and mechanical strains of the piston. Simulations were performed for the point of engine maximum power. Piston material was assumed to be 40HM (1.7225) steel and its properties are delivered basing on available scientific papers. The simulation assumed mean values of heat transfer coefficient, reference temperature and cycle pressure based on engine data, maximum power engine work simulation in AVL Boost software and literature. Part of boundary condition (e.g. cylinder wall temperature) was assumed basing on authors' engineering intuition and experience. The resulting temperature distribution in the piston was implemented for geometrically nonlinear mechanical FEM analysis. Both the analysis of thermal stresses and stresses of the hot piston in the top dead centre were performed.

Keywords: *Heat transfer, temperature distribution, four-stroke engine piston, Finite Element Method, 1D combustion*

1. Introduction

Renault Premium DXi11 460 EEV is a long distance truck characterised by high torque and elasticity. Its power and torque in a function of rotational speed is presented in a Fig. 1. Characteristic dimension and properties are appended in a Tab. 1 (all provided values were taken from [3] except of connecting rod length, which was approximately measured from an online picture). Material of a piston, wrist pin, and connecting rod (Fig. 2) was assumed to be 40HM steel of properties presented in Tab. 2 [1].

Tab. 1. Characteristic properties of Renault Premium DXi11 460 EEV engine

Bore diameter	123 mm
Stroke	152 mm
Connecting rod	245.5 mm
Displacement volume	10.836 dm^3 (all cylinders)
Number of cylinders	6
Compression ratio	18.3:1
Weight	1010 kg
Power rating	338 kW at 1,800 rpm
Torque	2,200 Nm from 950 to 1,400 rpm

Tab. 2. Properties of 40 HM steel

Temperature [ºC]	Young modulus [GPa]	Poisson ratio	Thermal conductivity [W/m^2K]	CTE [µm/m]	Proof strength 0.2% [MPa]
20	220	0.3	33.5	12	415
200	195	0.3	33.9	12.7	370
400	175	0.3	34.1	13.6	340
600	155	0.3	34.5	14.4	180

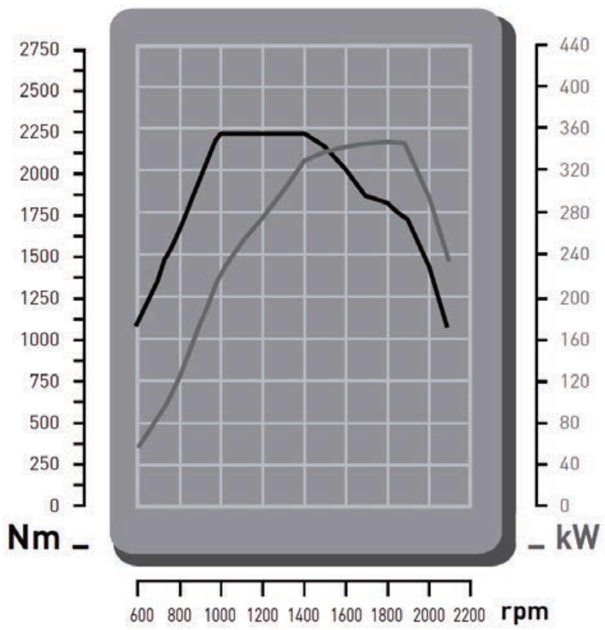

Fig. 1. Characteristic chart of Renault Premium DXi11 460 EEV engine

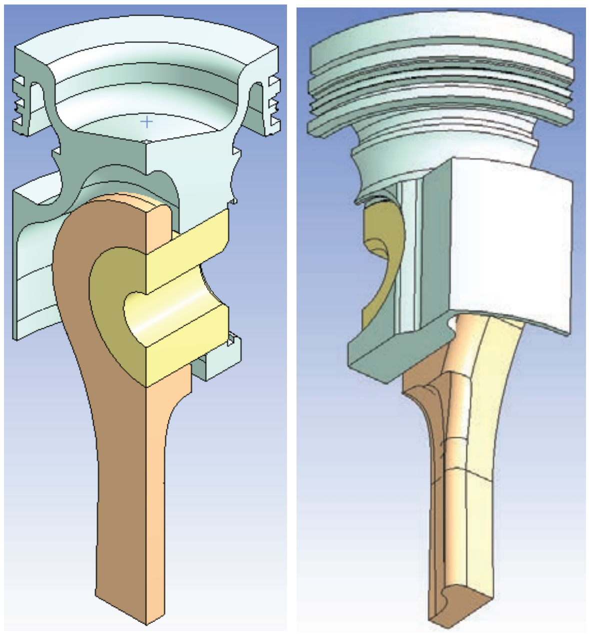

Fig. 2. Piston geometry

2. In-cylinder conditions

In-cylinder conditions were obtained by analysis of engine dimensions, power, and torque at its maximal power point (1800 rpm). Analysis was performed by AVL Boost software using system presented in a Fig. 3, which is provided in AVL Boost manual. One-dimensional model of combustion was included. The mass of injected fuel where calculated iteratively trying to obtain power of 343 kW. Values from Tab. 1 were entered to the system while the other variables (open and close valves timing, valves clearance, pipes and collectors dimensions, intercooler and turbocharger properties) were assumed.

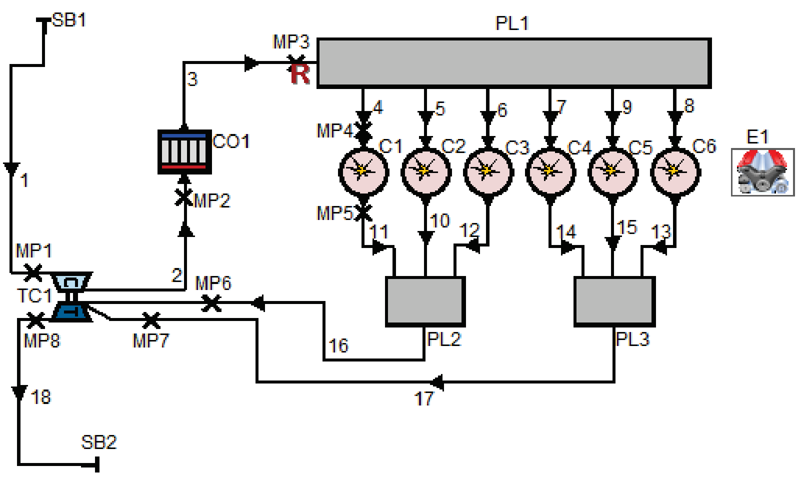

Fig. 3. AVL Boost scheme of Diesel Engine.

By specified procedure, *p-V* (Fig. 4) diagram in each cylinder was determined as well as gas temperature and heat transfer coefficient (Fig. 5). Heat transfer coefficient was calculated by Hohenberg formula [2] and include both convection and radiation heat transfer. Those functions will be used later in order to determine piston boundary conditions. Hohenberg formula was considered as providing most reliable results in this case based on author's intuition.

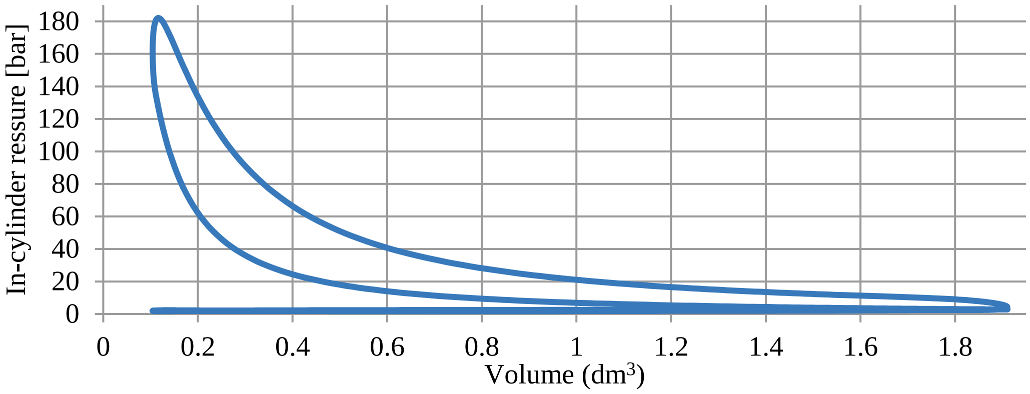

Fig. 4. p-V diagram

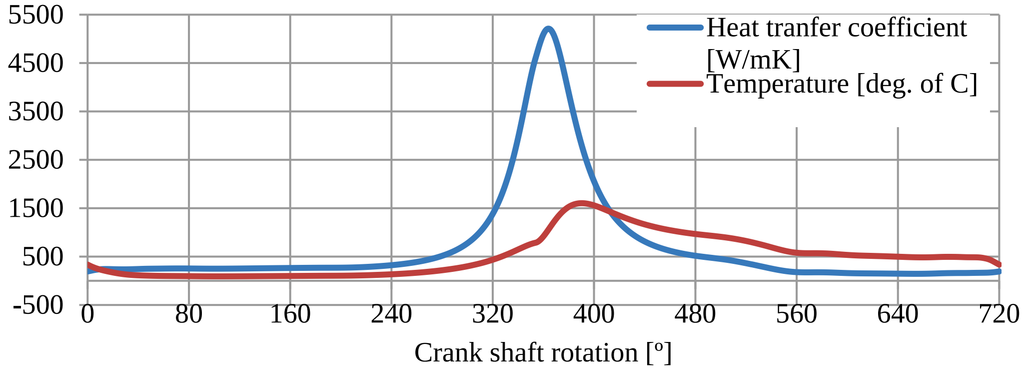

Fig. 5. Temperature and heat transfer coefficient

3. Stationary thermal and mechanical boundary conditions

Piston head is constantly in contact with exhaust gases or air pressed inside a cylinder, which is a most important heat source in a cylinder–piston–connecting rod system. Heat transfer boundary conditions were calculated by mean value and weighted mean values presented in Eq. (1-2) where α is heat transfer coefficient while T is temperature [4]:

$$\alpha_{mean} = \frac{1}{\tau}\int_{\tau} \alpha(t)\,\mathrm{d}t = 720.9\ \mathrm{W/m^2K}\,, \tag{1}$$

$$T_{mean} = \frac{1}{\tau\,\alpha_{mean}}\int_{\tau} \alpha(t)\cdot T(t)\,\mathrm{d}t = 869.7°\mathrm{C}\,. \tag{2}$$

Piston wall is moving along a cylinder therefor its contact temperature is changing in time. Also mean contact temperature is different for each place in a piston and is provided by Eq. (3) (where y is a position on piston measured from its head, while x is position of piston head measured from cylinder head). Piston wall nearest to a piston head is in contact with gases in combustion chamber; therefore, it was assumed that reference temperature in this place drops linearly from mean temperature in combustion chamber to temperature determined previously.

It was assumed that cylinder temperature is stationary (Fig. 6) and is constant along a cylinder circumference. Mean contact reference temperature calculated for the piston is presented in Fig. 7. Heat transfer coefficient from piston walls to a cylinder wall is assumed to be 750 W/m^{2}K referred to piston walls between piston rings and 500 W/m^{2}K referred to reaming walls:

$$T_{mean}(y) = \frac{1}{\tau\,\alpha}\int_{\tau} \alpha\cdot T(y + x(t))\,\mathrm{d}t\,. \tag{3}$$

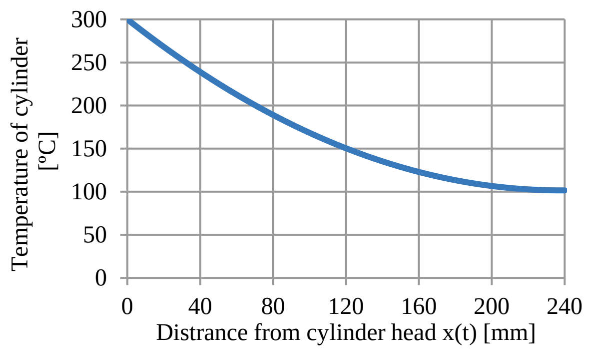

Fig. 6. Cylinder temperature

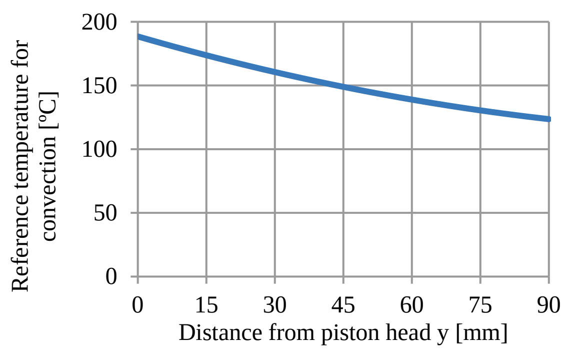

Fig. 7. Reference temperature for piston convection

Piston rings are continually in contact with cylinder wall and repeatedly in contact with ring groove walls. Mean values of heat transfer coefficient between ring and upper or lower ring groove face is estimated as 10-20 kW/m^{2}K or 20-30 kW/m^{2}K, respectively. Nonetheless, only first of them was applied in the FEM model (estimated as 15 kW/m^{2}K) because only stationary heat flow is considered in a paper. Applying heat transfer to both upper and lower groove wall would imply heat transfer between those surfaces which do not appear in piston heat transfer due repeatability of contact and low heat capacity of piston ring. Heat transfer coefficient between cylinder wall and piston ring is estimated to be 24 kW/m^{2}K. Thermal conductance in a piston – wrist pin – connecting rod system is assumed to be 2500 W/m^{2}K.

Moreover, piston rings are constantly pressed against the cylinder wall. Friction coefficient teamed with a pressure and velocity generates heat, which may be described by the formula (4). It is assumed that heat is generated both cylinder and ring surfaces half by half:

$$q = \frac{\mu S n p}{30} = 91200 \text{ W/m}^2. \tag{4}$$

Piston is assumed to be in top dead centre when maximum pressure occurs; therefor the crank rod is aligned to a cylinder symmetry axis. The maximum pressure read from *p-V* diagram shown in Fig. 5 is 183 bars. Piston rings are not modelled; however, it is assumed that pressure drops 3 times at each ring as it is presented in Fig. 8. There is also a displacement along cylinder axis set to a 0 mm at surface denoted as A in the figure.

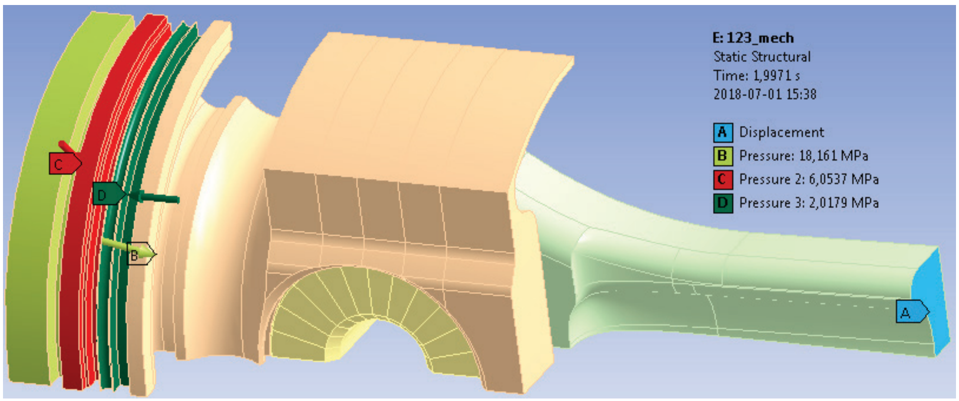

Fig. 8. Mechanical boundary conditions

4. Finite Element Analysis results

Stationary thermal analysis resulted in a temperature distribution and heat transfer presented in Fig. 9 and 10. Maximum temperature obtained is 486ºC and is placed in a convex blended surface of a piston head. Resulting temperature is very high for assumed steel.

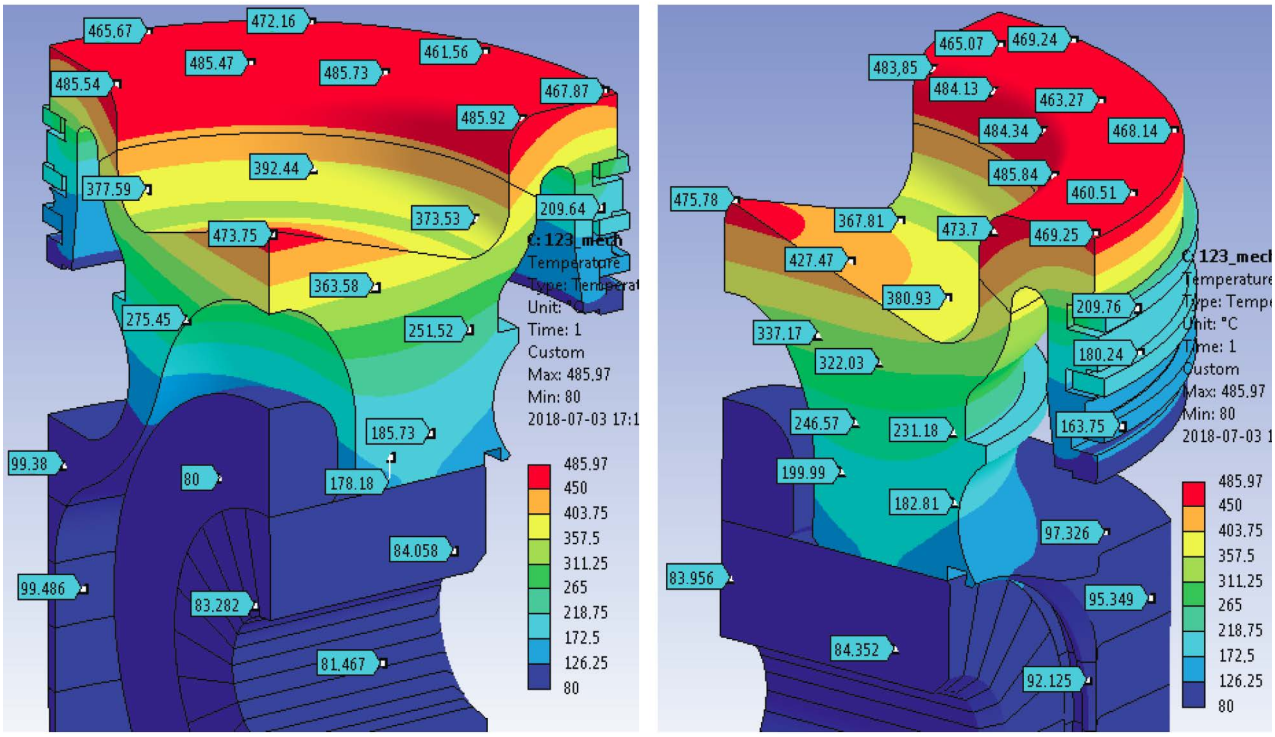

Fig. 9. Temperature distribution in a piston

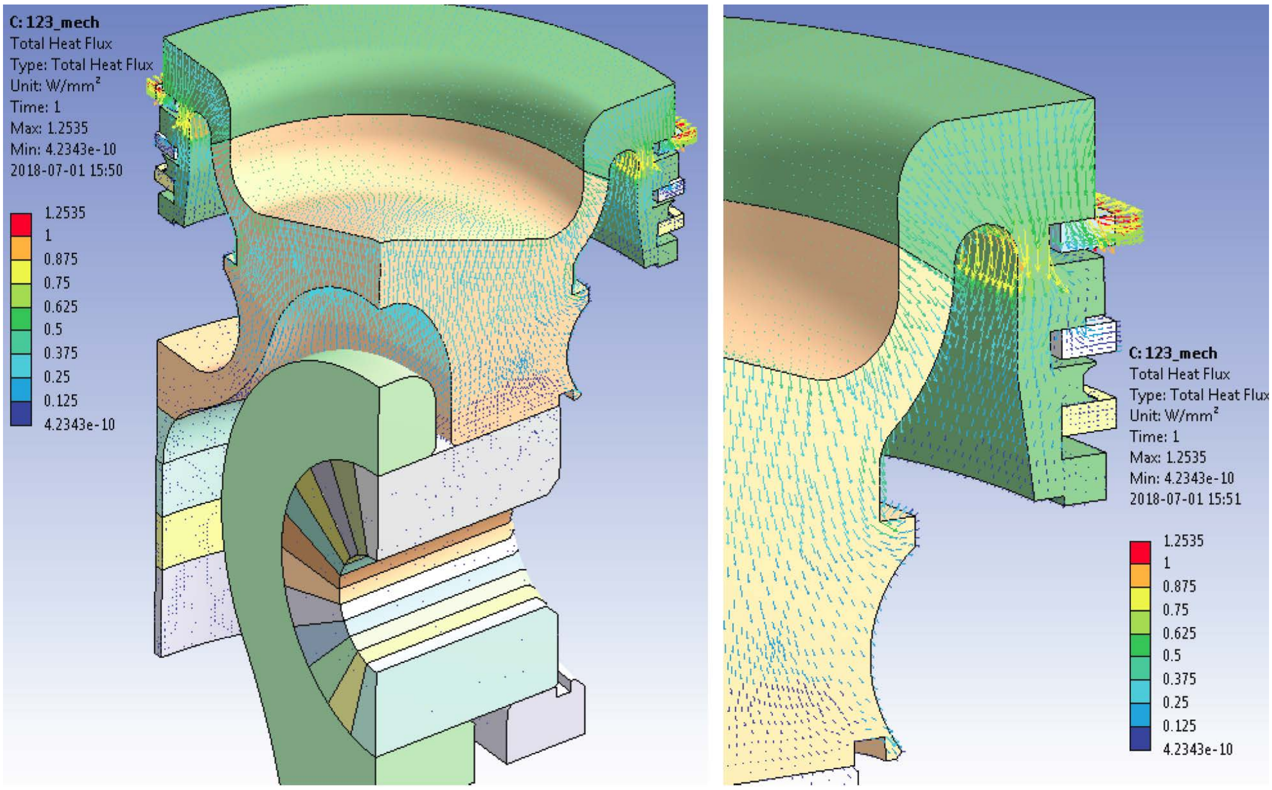

Fig. 10. Heat transfer in a piston

The most loaded part is concave blend surface near to the fire ring groove. Equivalent stresses in these places vary from 230 to 580 MPa (Fig. 11 and 12). The second most loaded place is concave blend in top piston surface. Its equivalent stress varies from 250 to 330 MPa, which is, much less than in previously mentioned place. In both of mentioned regions, the maximum stress is a result of compression of elements. Wrist pin is loaded to 389 MPa of equivalent stress.

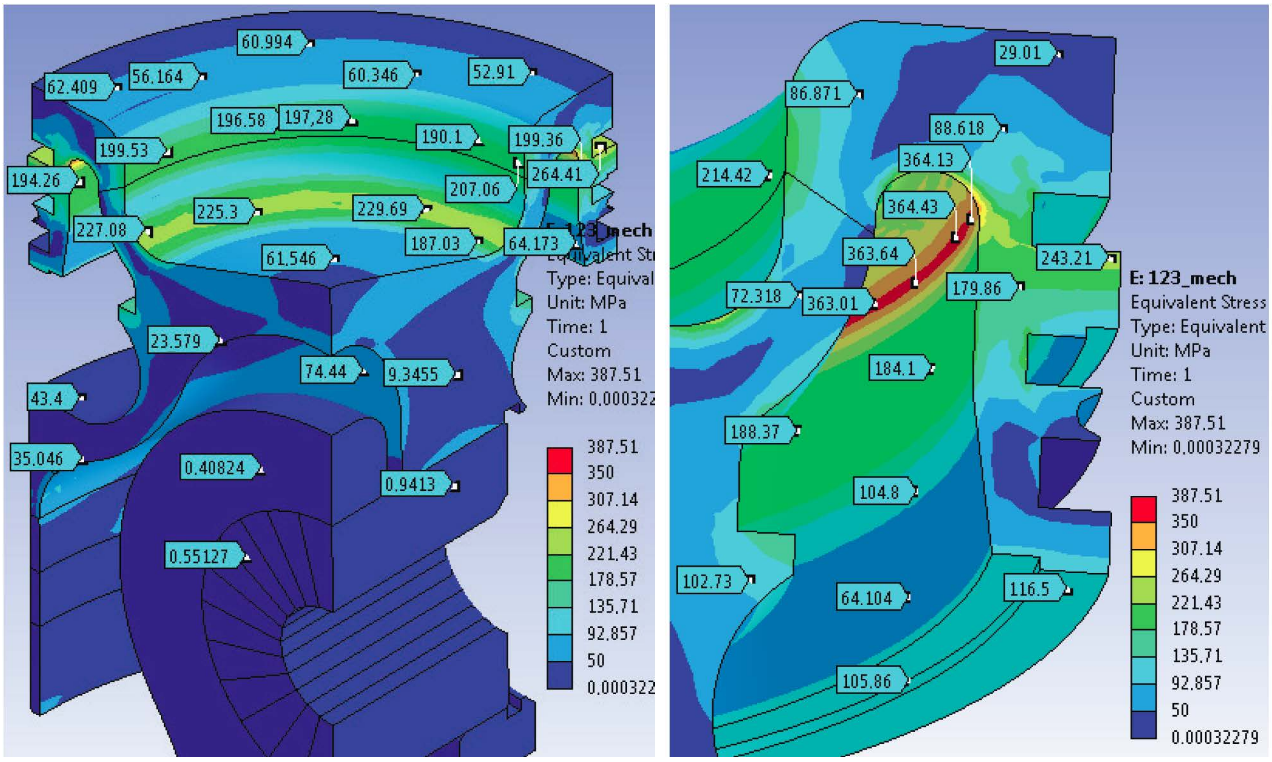

Fig. 11. Thermal stresses in a piston

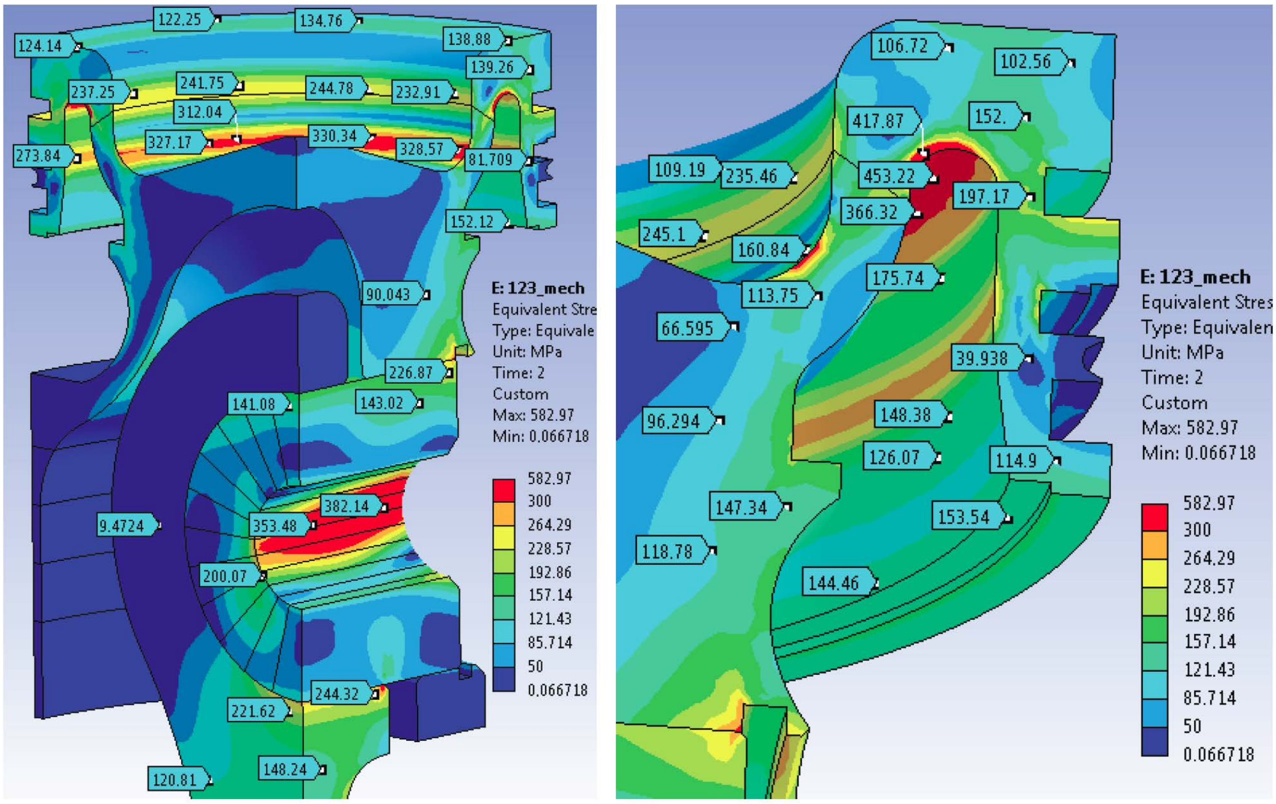

Fig. 12. Thermal and mechanical stresses in a piston

5. Conclusions

Maximum temperature region is exposed to highly oxidation conditions since is in contact with hot exhaust gases. Fortunately, stresses in places of highest temperature are not very high so oxidation is the main life-limiting factor in this place. Equivalent von Mises stress at piston head surface is only about 130 MPa and – what is even more important – only a little of it varies during engine work.

Places of maximum equivalent stresses are overloaded since both proof strength and fatigue strength (which is approximately 0.5 of Re) are exceeded. It is probably a result of implied pressure, heat transfer coefficient and temperature presented in Fig. 4 and 5. Combustion process as well as open and close valve timing differs from Renault engines, therefor p, α and T are also different. Due to authors it is the main reason of such a situation since Renault Premium DXi11 460 EEV is well tested and proven over the year's construction.

Wrist pin is loaded to 389 MPa of equivalent stress and fatigue phenomenon is a zero based shear stress. This value is probably exceeded too until stronger steel than assumed is applied.

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