THE STRESS ANALYSIS OF HIGH LOADED DIESEL ENGINE PISTON

Karol Cupiał

Czestochowa University of Technology
Department of Mechanical Engineering and Informatics
Institute of Internal Combustion Engines and Control Engineering
Armii Krajowej Av. 21, 42-201 Czestochowa, Poland
tel.: +48 34 3250501
e-mail: cupial@imtits.pcz.czest.pl

Michał Sobiepański

Czestochowa University of Technology
Department of Mechanical Engineering and Informatics
Institute of Technology and Manufacturing Automatization
Armii Krajowej Av. 21, 42-201 Czestochowa, Poland
tel.: +48 34 3250509
e-mail: sobiepan@itm.pcz.czest.pl

Abstract

Modern diesel engines reach high power output per volume unit. This is the main reason, why modern passenger cars and other medium size vehicles are powered by those engines. It is very important to keep reliability, characteristic for past generations of diesel engines. It is a difficult task, because of higher thermal loading of engine parts, especially pistons. Typical fuel injection systems are now, almost all, high pressure, common-rail systems. Modern common-rail injectors are situated centrally between valves, and are manufactured as minimum six nozzle injectors. Combustion chamber is made in piston crown. This is the highest thermal loaded part of whole engine. Injected fuel starts to burn when piston is near to TDC position. Flame acts piston surfaces, what can be proven by inspection of piston with help of endoscope, or after removing engine head. The thermal load of piston crown is not uniform, due to not uniform conditions of heat transfer at piston crown surface. Part of piston crown reaches higher temperature due to direct flame acting. Authors present results of numerical simulations, which were done to check how stresses in piston depend on injector position and temperature distribution in piston material. The results of calculations indicate, that injector should not to be situated randomly. If nozzles of injector are situated in such way, that burning fuel sprays acts piston in the piston pin plane, stresses in hub of piston pin and piston crown increase significantly. That can decrease reliability of engine due to piston failure. Authors indicate, that thermal load is the most dangerous for piston of modern, common-rail diesel engines. Load caused by mass forces and pressure are less important when temperature of some regions of piston is above 320°C.

Keywords: *pistons, internal combustion engines, stresses, numerical analysis*

1. Introduction

Common feature of modern diesel engines is the high value of volumetric power factor. Modern diesel engines are almost all turbocharged, direct injection engines. Increased exertion of these engines entails difficulties with obtaining the expected reliability and durability. Durability and reliability of highly heat loaded engine components (pistons, exhaust valves, turbocharger rotors) determines the durability of the whole engine. High thermal and mechanical loads contribute to the erosion of the edge of the combustion chamber and piston cracks. Examples of such defects are shown in Fig. 1.

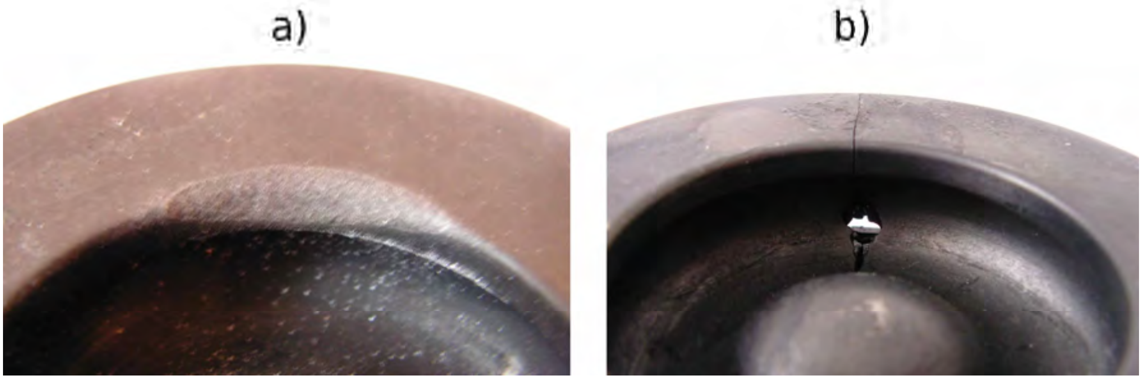

Fig. 1. Examples of failures typical for pistons of modern diesel engines: a) erosion of piston crown acted directly by burning spray of fuel , b) perforation and rupture of piston crown

To ensure the expected sustainability of piston it is important that the temperature did not exceed about 66% of the melting point temperature of the alloy [1]. In addition, stress should not exceed the limit value of stress fatigue. For the contemporary used aluminium alloys with high silicon content the maximum temperature of piston crown is limited to about 370°C [3]. The use of combined pistons or pistons made of steel can increase the limit of their acceptable heat load . The main drawback of such solutions is the piston cost. Such pistons are too expensive to be used in passenger cars engines. Other fact is the mass of pistons made of steel, which exceeds considerably mass of pistons made of light alloys.

Piston load is unsteady processes. It consists of instantaneous load (mass forces and pressure forces) and low frequency loads - thermal loads depending on engine operating parameters. A numerical analysis was performed to examining the reasons for cracking of pistons, found in cars with mileage over 200,000km. Numerical analysis included the calculation of stresses in the piston with a special emphasis on those areas, where stresses reach the highest values.

2. The object of research

The examined piston came from the turbocharged, four-stroke diesel engine, equipped with common-rail fuel injection system. The maximum mean effective pressure for this engine (p_e) is 1.772 MPa. Volumetric power factor reaches 44 kW/dm^3. The piston is equipped with two sealing rings and one oil ring. The top sealing ring is situated in a carrier, which protects ring groove against accelerated wear. This carrier is made of austenitic cast iron alloy from the Ni-Resist alloys family. Its presence has a minimal impact on the reduction of radial deformation of the piston caused by temperature. It is result of small difference between thermal expansion coefficient values for Ni-Resist and aluminium alloy used to form the piston. All piston rings are situated above piston pin. Near the edge of toroidal combustion chamber made in the piston crown, at the level of the second sealing ring, is placed the channel to which the lubricating oil is injected, providing internal cooling of the piston crown – Fig. 2.

3. Computational model of the piston

The first step in preparing of the calculations was to create three-dimensional models of components whose presence is taken into account during calculations. These models were created in CAD software – CATIA [9] . Separate models of the piston, first ring carrier, connecting rod, pin and bushings were created as three-dimensional discrete models. Such models were then imported into the main analysis software – ABAQUS [10]. The real system has one symmetric plane, which is perpendicular to piston pin axis. Thanks to this calculations were performed for simplified model, consisting of a half of the model made in the CAD program. Taking into account the symmetry of the model allows concentrating the finite elements mesh maintaining unchanged the calculation time. In such way the influence of mesh asymmetry on computation results was eliminated.

Fig. 2. Cross-section of piston being the analysis object. The cooling channel and carrier of the top sealing ring are shown

4. Stress computations by finite element method

Literature is rich in examples of analysis of the temperature distribution in the piston material. Boundary conditions, used during such analysis, were determined on the basis of measurements performed on the test bench equipped with real engine. The knowledge of in cylinder pressure trace is fundamental for determining of heat exchange conditions describing heat exchange between working medium and combustion chamber walls, including top surface of engine piston. Since the authors were unable to carry out the measurements on the working engine, the boundary conditions for heat exchange for piston surfaces, excluding top piston surface, were assumed on the basis of data available in literature [1]] . To determine heat flux flowing through the top piston surface the trace of in the cylinder pressure was obtained by modelling engine cycle with computer program SILNIK [2]. In case of steady engine operating conditions (p_e=const, n=const) the heat flux flowing through piston surfaces is periodical, with frequency equal to engine cycle frequency. The temperature of external layer of the piston surface changes with the same frequency, but the fluctuations are small due to thermal inertia of piston material. At a depth of several tenths of millimeter the temperature fluctuations disappear, and the temperature field can be regarded like in steady state. This allows to significantly simplifying all calculations by replacement of instantaneous values of convection coefficient and working medium temperature by averaged values. Additionally averaged value of heat flux for top surface of the piston can be determined by simulations of engine cycle done with SILNK program. During computational modelling of engine cycle, considering 0-dimensional model of combustion chamber, averaged values of heat flux for piston were calculated, taking into account the Vibe [7] model of heat release and Woschni [6] model for heat loses . To take into account the impact of a flame on the piston surface this surface was divided into two areas. Higher value of heat flux was assigned for the area considered as acted directly by burning fuel. The total heat flux for piston surface was preserved by decreasing heat flux for area not acted by the burning fuel spray. Two considered computation cases, determined by injector position against piston are illustrated in Fig. 3. The analysis of stresses and deformations was conducted in ABAQUS. It was necessary to perform two special models. In first the mesh created by mesh generator consists of elements dedicated for heat transfer analysis. The temperature distribution calculated in this model was imported as field load into second model, which consists of elements appropriate for stress analysis. Stress analysis consists of three static steps. The load in first step was mass force load increased in second step by pressure load. Finally temperature field, computed previously during thermal analysis, was loaded into model to introduce thermal loading. Properties of materials important for computations are summarized in Tab. 1. Additionally computations were conducted for two cases of piston load, adequate for maximum torque (p_e=1.772 MPa, n=1600 1/min) and maximum effective power (p_e=1.497 MPa, n = 3500 1/min). Calculated maximum values of acceleration for piston are respectively 1726 m/s^2, and 8260 m/s^2.

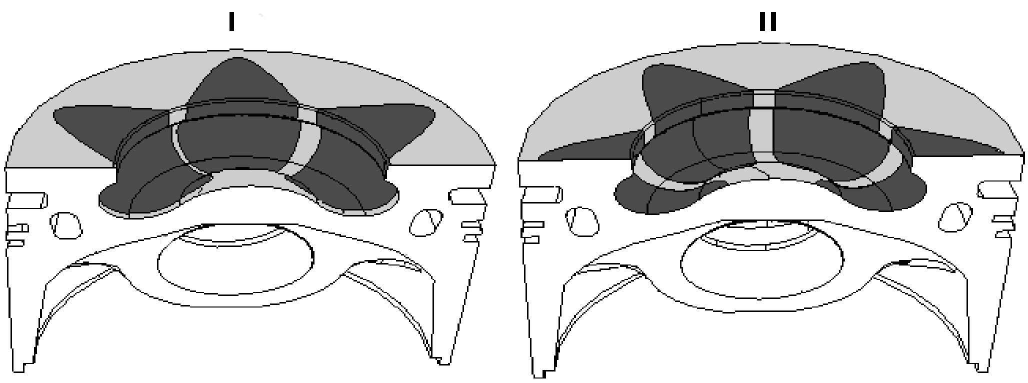

Fig. 3. Two computation cases determined by injector position against piston crown: I- burning fuel injected by injector acts piston area above the hub, II- piston hub area located between two fuel sprays

Tab. 1. Material properties assumed during computations [3]

Material	Thermal conductivity [W/mK]	Thermal expansion 10^{-6} [1/K]	Density [kg/m^3]	Specific heat [J/kgK]	Poisson's ratio	Young's modulus [GPa]
AlSi	155	21	2700	960	0.3	90
Steel	79	12.2	7870	500	0.3	200
Ni-Resist	17	18	7730	600	0.3	150

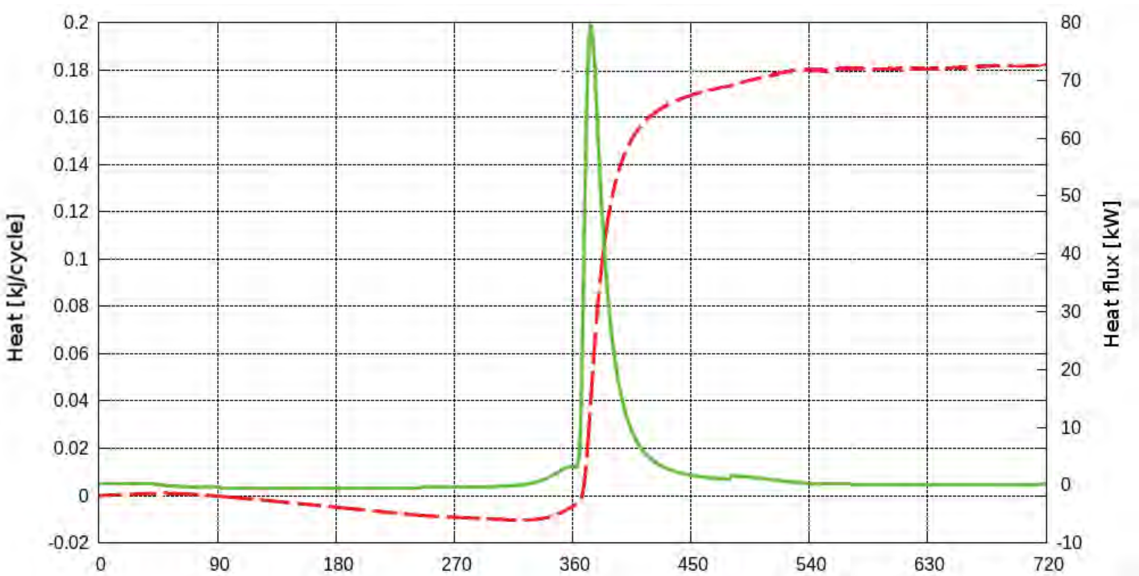

Fig. 4. Heat loses for piston (dashed line) and instantaneous heat flux for piston surface being with contact with working medium (solid line) as example results of engine cycle modelling done in SILNIK program (pe =1.64MPa, n=3500 1/min)

Tab. 2. Parameters describing heat exchange process between working medium and piston surface obtained as results of engine cycle modelling in program SILNIK

Engine operation parameters	p_e =1.772 MPa n=1600 1/min	p_e = 1.497 MPa n=3500 1/min
Mean medium in cylinder pressure [MPa]	1.88	1.65
Peak cylinder pressure [MPa]	17.04	16.18
Effective piston top surface area [cm^2]	109.5	109.5
Engine cycle duration [s]	0.75	0.03
Piston heat flux per one engine cycle [kJ/cycle]	0.82	0.09

Tab. 3. Summarized boundary conditions used during temperature field calculations

Surface name	α [W/m^2 K]	T [K]
Top surface of piston above the first piston ring (it was assumed, that 40% of this area is acted directly by burning fuel sprays)	Boundary conditions were adopted in the form of steady heat flux calculated in program SILNIK	
Side surface of piston crown	85	479
Surface between sealing rings	110	457
Surface between oil and sealing ring	110	426
Piston coat	110	400
Internal piston surface	75	350
Channels of cooling oil	970	360
Top ring groove surface	upper 3400 lower 8900	465 465
Middle ring groove surface	upper 2700 lower 5900	455 455
Oil ring groove surface	upper 2100 lower 3900	409 409
Pin hub	80	350

5. Computations results

First step in both analysed cases was computation of temperature distribution in system consists of parts models. Fig. 5 shows example temperature distribution in system cross-section obtained for one of analysed cases. Maximum temperature region includes upper edge of combustion chamber situated in the piston. The mesh node located in cross-section, for which temperature reaches the highest value is indicated as point *B*. Calculated temperature is lower than the limit temperature described as about 370°C [3]. Calculated deformations (example shown in Fig. 7, 8) are in the limits of piston clearance. This confirms the correctness of the adopted boundary conditions. Stresses, main object of analysis, are strongly depended on temperature distribution in piston material. Calculated von Mises stress distribution in piston pin symmetry plane indicates the region in piston hub, in which stresses can be high - above 200 MPa, what is illustrated in Fig. 6. This indicates, that the yield of piston alloy should be higher than 200 MPa in temperature about 500 K. Analysis have shown, that the most important for stress distribution is temperature distribution. It is well illustrated in table where are summarised stresses in points A and B for three load steps of both analysis cases.

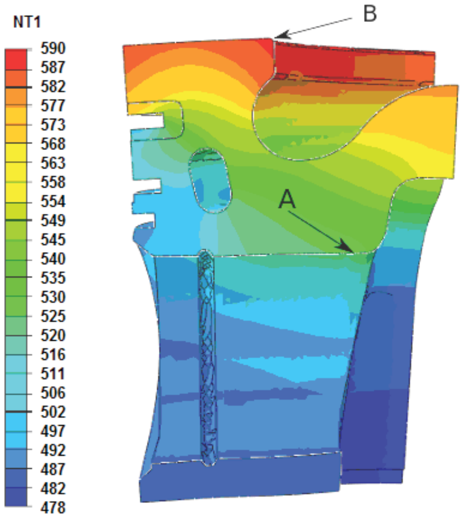

Fig. 5. Computed temperature distribution (pe=1.772 MPa, n=1600 1/min). Point B indicates the mesh node in which the temperature reaches the highest value in presented cross-section

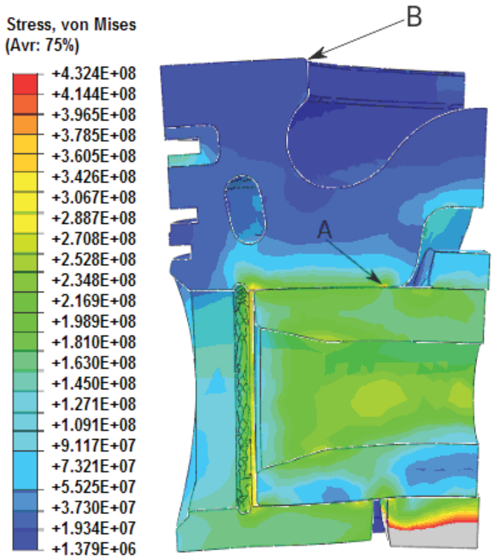

Fig. 6. Von Mises stress and their accumulation in mesh node market with the letter A (p_e=1.772MPa, n=1600 1/min)

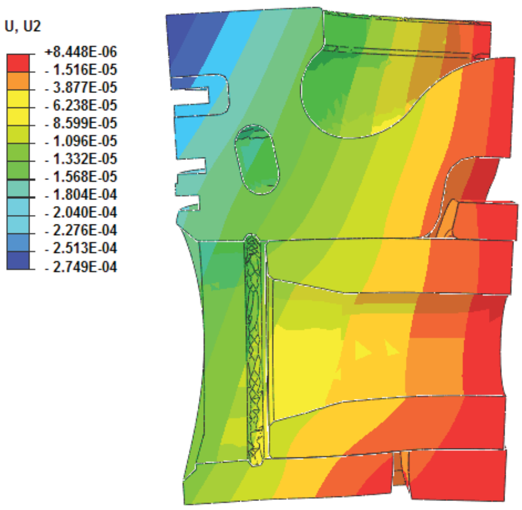

Fig. 7. Radial deformation of the hot piston decrease piston clearance to 0.05mm at the external edge of piston crown

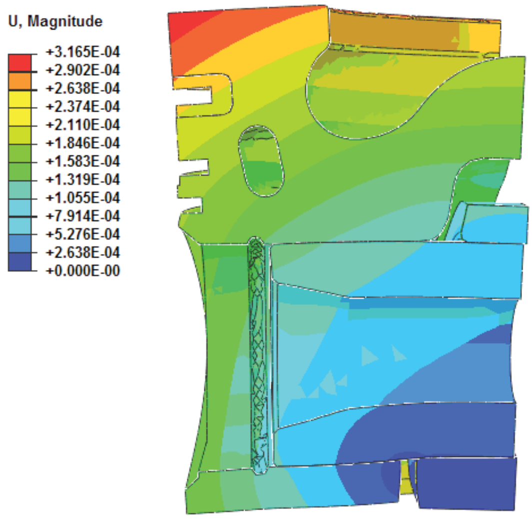

Fig. 8. Total deformation of hot piston loaded by mass forces, pressure and non uniform temperature field

Tab. 4. Summary of von Mises stresses (σ) stress component (σ_{11}) normal to symmetry plane inclusive pin axis and temperature calculated for points A and B during analysis

Analysis case (Fig. 3)	Engine operating parameters	Point (Fig. 6)	σ [MPa]	σ_{11} [MPa]			T [K]
				G*	G+p	G+p+T	
I	p_e =1.772 MPa n=1600 1/min	A	350	1	-120	-88	525
		B	10	0.2	5	-9	588
	p_e = 1.497 MPa n=3500 1/min	A	382	5	-109	-75	538
		B	8	0.4	4	-12	607
II	p_e =1.772 MPa n=1600 1/min	A	253	0.5	-57	-89	528
		B	21	1.2	5	-2	589
	p_e = 1.497 MPa n=3500 1/min	A	282	2.25	-53	-86	540
		B	18	0.6	4.1	-4	603

** Letter "G" means load caused only by mass forces, "G+p" is load caused by mass and pressure forces, "G+p+T" is complete load including non uniform temperature distribution in piston material*

6. Final conclusions

Obtained analysis results allow the following conclusions:

1. The large value of temperature gradient in piston crown is the main source of piston deformations and stresses. Load caused by mass forces and pressure forces is not dominant, but because of its periodicity it is the source of fatigue stresses characterised by high amplitude.
2. The influence of injector position against piston crown on the highest temperature of piston is not as important as the influence of this positioning on the stress distribution. The stresses in piston, especially in piston hub, are higher when burning fuel acts the piston crown directly over pin hub. Such positioning of the injector is adverse and should be avoided.
3. Analysis of only von Mises stress is not enough for precise evaluating of true loading of the piston. Additional analysis of all stress tensor components is necessary due to nature of fatigue stresses.
4. Further studies in discussed area seem to be needed due to rising popularity of modern diesel engines equipped with high pressure fuel injection systems.

Acknowledgements

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