Study of influence of boundary conditions on deformation and stresses in a cooled piston of a diesel engine – part B - calculations

W Mitianiec
Cracow University of Technology, Al. Jana Pawla II 37, 31-864 Krakow
e-mail: wmitanie@usk.pk.edu.pl

Abstract. The paper is continuation of the part A and presents influence of boundary conditions on mechanical and thermal stresses of an oil cooled piston in diesel engine Scania DC09 074A and its deformation due to high combustion pressure and variable temperature. Calculations of the piston loads were carried out by using 0D mathematical model at assumption of the Vibe combustion model and the Woschni heat transfer model for homogenous mixture. For precise determination of the gas temperature in the combustion chamber the CFD technique was used and for that case the simulation of compression, combustion and expansion processes was carried out by using the program Fluent. The numerical calculations were done for two kinds of fuel: diesel oil and CNG. In order to proper work of CNG engine the compression ratio was decreased. This was done by reducing the volume of the combustion chamber. Calculations done in the program Ansys showed that the modification of the piston do not influence significantly on the total stresses. The paper shows the differences of the total stresses and piston deformation at fastened piston hub and at “floating” piston pin. Most calculations are taken that the piston hubs are fixed, but it is not right assumption. In summary the paper gives indications how to set boundary conditions for “floating” piston pin. Due to different fixing of the pin in the piston hub the deformation of the piston shape and von Misses stresses differ at the same boundary conditions The analysis has shown that FEM calculations of the piston should be done together with the pin for giving the engineers right information how to design the piston shape, usually “barrel-oval”, in order to enable a proper clearance between the piston and cylinder walls at high loads.

1. Introduction

Heat exchange in internal combustion engine was considered by many authors [1, 2, 3, 9]. The piston is an indirect element in heat exchange in combustion engines, because it transfers the heat from the gas to cylinder walls by piston rings and some part of heat is transferred from the bottom surfaces of the piston to the gas inside the crankcase. The part A of this paper presents theoretical considerations of the thermodynamic and mechanical parameters, which should be taken into consideration at calculations of stresses and deformations of the piston and piston pin. Besides of heat transfer from gaseous charge to the piston crown by convection and from the piston to gases being in crevices or crankcase, very important is determination of constraints which set the location of the piston in stable mechanical equilibrium. Determination of the piston shape is very important for assurance of proper clearance between piston and cylinder walls particularly at highest thermal and mechanical loads especially for heavy duty engines mainly diesel engines. Many authors do not take into account a
contact the piston pin with the piston hubs [8, 10] and try to calculate stresses and deformation of the piston at assumption of fixing the piston hubs.

This paper gives the method of strength calculation of the piston and gives information how to set boundary conditions and constraints corresponding to real conditions of the piston work. Determination of convection coefficients in the combustion chamber was predicted on the basis of Woschni hypothesis [4, 5]. Balance of the heat in the piston given in the part A enables determination of temperature in the piston and pin materials at steady state. Stresses in the piston material arise from its thermal expansion and elasticity under acting force (shortening or elongation). Recently strength calculations of pistons are carried out by different computer FEM (finite element method) technique [14]. These programs in final stage solve many linear equations which correspond to number of assumed calculated small volumes obtained during meshing process of the considered object.

The most important is definition of temperature of the gas under the piston, which influences on thermal expansion of the piston crown. On the basis of work of different pistons in combustion engines it should be paid attention on the zone of the piston hubs, where a seizing of the piston often takes place. Correct calculations should show considerable deformations at the upper part of piston and also in the hub zones in respect to the initial cylindrical shape of the piston. The real shape of the piston in “cold conditions” should correspond to the “barrel-oval” shape. Calculated deformations give the engineers how to design the piston shape and calculated stresses give information how to change its inner structure. Strength calculations of the piston are the most important in an engine project.

2. Piston dimensions and calculation mesh

Calculations of stresses and deformation of the piston need fixed values of pressure, temperature and convection coefficients in different parts of piston’s surfaces. It takes into account a contact of the piston pin with the piston’s hub. The piston made with aluminium alloy has another elasticity than the pin made of alloy steel. The main dimensions of the Scania piston are shown in Fig. 1a with the piston pin (external diameter d=64 mm) which slides in the piston hubs. The piston has the internal channel for its cooling by the flowing oil injected at BTDC. Mechanical and thermal calculations are carried out by educational program Ansys ver. 18.2 [12] in module Thermal Stresses which is connected the module Static Structural [ ]. The mesh of the assembly piston-pin contains 154619 nodes and 94048 tetrahedral and brick elements and the mesh view is presented in Fig. 1b.

![Figure 1. Cross section of piston (a) and calculation mesh of piston and piston pin (b)](image)
3. Thermal loads
Due to charge motion under the piston the heat from the gas to the piston crown is transferred by convection. At the same gas temperature but higher value of convection coefficient the temperature of the piston surface is lower. For that case during combustion process when the convection coefficient reaches the highest value temperature of the piston is not so high and for that case the piston material is not a subject to a local damage. The temperature and pressure of the gas in the cylinder was calculated at assumption of Vibe model [7] of combustion. The thermal loads on all piston surfaces are shown in Fig. 2. For highest thermal load the piston crown was divided into two surfaces: the central surface of the combustion chamber with higher external temperature 2500 K obtained from CFD calculation in Ansys Fluent program and external flat surface loaded with temperature 1300 K.

4. Mechanical loads
Boundary conditions of mechanical loads concern to acting of charge pressure on the piston crown with equal value the same as in the cylinder. The author carried out the research for the case of maximum pressure and for mean indicated pressure. The piston is subjected to action of variable pressure during one working cycle. Some authors suggest taking value of pressure at 30° CA ATDC when the tangential force on the crank reaches maximal value. In this work it was assumed the maximal pressure equal 15 MPa which exceeds value at lower loads, but this stationary engine driven electrical generator works almost at high loads. Very important was to determine the constraints of the piston and piston pin. It was assumed that the middle bottom surface of the pin, where the pin is in contact with the small-end of the connecting rod, has no displacement and was fixed in all nodes. It was assumed also the contact of the pin with hubs of the piston. The hubs have possibility of movement in X and Y direction with fastening of rotation according to X-axis. Thus the pressure from...
the piston crown is transferred on the upper surfaces of the hubs which are in the contact with the piston pin. Finally the force acting on the piston is transferred on the bottom surface of the small-end of the connecting rod. This approach guarantees determination of real conditions of the work of the piston pin in almost all solution when the pin is sliding according to the hubs. Figure 3 presents the mechanical loads acting on the assembly piston – pin, where D denotes fixing of the nodes of the pin in the bottom surface.

![Figure 3. Mechanical loads and constraints of Scania piston](image)

Thermal and mechanical loads as boundary conditions and also determination of the constraints in the piston assembly enable to calculate all types of stresses and deformation of the piston shape. Deviations from the round shape of the piston give a possibility to determine the right shape in normal conditions usually it is a barrel-oval shape.

5. Calculation program

Because of the unusual popularity of the commercial program Ansys [12], most of the engineers use it for calculation of stresses and deformation. This computer program but in the educational version was also used for determination mechanical and thermal stresses and the shape deformation. However there are many other computer programs which enable to predict such parameters. There are also some programs licensed on GNU conditions, which are free from any charge, such as Salome-Meca [13] with module Code-Aster for calculation of mechanical stresses. Some other free FEM programs are CalculiX (GNU license), Elmer (GPL license), FreeFEM++, DUNE and others. The Ansys program with module Thermal Stresses is initiated in Workbench, where geometry is imported from different CAD systems, in this case from Salome-Meca program, where the geometry of the piston was created and next the geometry was exported to the file in *.stp format. The geometry in such format is easy transferred to the program Ansys, where in the module Mesh the real geometry can be divided into several small volumes in tetrahedral or brick forms. This module requires definition of contacts and named selections of the surfaces or volumes needed to assign the required boundary conditions, such as pressure, temperature convection, constraints, properties of material and others. Calculations usually are carried out for steady state conditions, but it is possible calculations in transient thermal conditions. The first step is calculation of temperature of the piston assembly and heat flux (total and directional). By knowing temperature in the next step the static structural analysis is carried out which finally determines total and directional deformation, equivalent von Mises stresses, equivalent total
strain, thermal strain, contact tool and safety factors in the whole piston and pin materials. The program enables presentation results in a graphic form with exporting the screen view to graphic files.

6. Results of calculation
Calculations were carried out for different boundary conditions and constraints of piston hubs and piston pin. The Scania engine fuelled with CNG requires lower compression ratio. For that case the piston height was reduced and volume of the combustion chamber was increased. Temperature distribution in the combustion chamber was determined by simulation of compression, combustion and expansion processes in Fluent program and additionally for comparison also in 0-D computer program with simulation the working cycle in the whole engine system.

6.1. Contact of piston and piston pin
The first case of analysis was to perform an influence of the contact of the pin with hubs at high loads (pressure 15 MPa and temperature in the central combustion chamber 2500 K). At given above assumptions and boundary conditions the program calculated the contact surfaces in the scale from lower values to highest values. Particularly exposed to high pressure are upper surfaces of the hub bore near the small-end of the connecting rod. Figure 4 presents the places where is sticking of the pin and hub (dry friction), sliding surfaces with possible lubrication, near surfaces outside from the contact and far, where is a clearance and is any contact between the pin and the hub. This picture presents the contact of two parts and concerns to the case of sliding piston pin in the hub bore and traditionally is called as “floating pin”. In some heavy duty engines the pin is fixed in the piston hubs (a press a pin into the hub), but also in this case the pin is bent and causes deformation of the hubs. The contact between the pin and hub is not constant on surfaces as a result of bending of the pin.

![Figure 4. Contact surfaces of piston and piston pin](image)

6.2. Deformation of the piston shape
Change of the piston shape is caused mainly by thermal expansion of material and is very large for the aluminium alloy from which the piston was made. However a deformation of the piston hubs occurs due to vertical force being a result of pressure action on the piston crown. The first analyzed case with a “floating pin”, where the contact between the piston and the pin is considered. Total deformation of the piston shape and piston pin for that case is shown in Fig.5. The biggest deformation occurs on the piston crown, where the outer surface expanded 0.4 mm in all directions counting from the z-axis. Deformation of the piston pin made from steel alloy also occurs but is only 0.1 mm.
Figure 5. Piston and pin deformations at free hub surfaces (X and Y axis)

Due to a possibility of lengthening of the region with piston hubs and heat flux from the piston crown the diameter of the piston in the plane crossing the pin axis increases about 0.5 mm. The outer surface on the bottom part of the piston shows a large deformation, which is caused by bending of the hubs by the piston pin. For more accurate determination of deformation values the Ansys program enables presentation of directional deformation in order to do the right shape of the piston when the drawing is prepared by the engineers. The deformation of the piston is significantly reduced by cooled oil which flows through the oil channel around the ring zone. In the second case, which is mostly considered by calculation engineers, with a hub support (no movement and rotation and no influence of the pin on deformation of the hubs) results of the diameter change in the middle part of the piston are quite different in comparison to the first case (Fig. 6).

Figure 6. Deformation of piston at fixed hubs without piston pin
Total deformation of the piston shape with fixed hubs shows that the piston crown increases radially with value 0.4 mm and in the pin axis there is any change of dimensions. Deformation of the shape occurs only in the perpendicular plane to the piston pin axis. Generally in this case deformations of the piston shape are smaller than in the first case.

The third analysed case considered to the piston without the pin but with possible movement of the hubs in X and Y axis, but with a support in X axis and Z axis. For the same mechanical and thermal loads (boundary conditions) but with other constraints deformation of the piston differs from the previous cases and is shown in Fig. 7. The change of the piston diameter in the upper part amounts 0.8 mm and in the plane of the pin axis increases value about 0.7 mm. Like in the first case (Fig. 5) there is big deformation of the surface in the bottom part of the piston. The upper surface of the hub bore shows a low deformation. This case shows the change of dimensions approximately similar to the first case. It means that on deformation of the piston the biggest impact has temperature, not mechanical loads, but in calculations very important is determination of all constraints.

![Deformation of piston at assumption of fixed rotations of hubs and free XY constraints](image)

**Figure 7.** Deformation of piston at assumption of fixed rotations of hubs and free XY constraints

### 6.3. Temperature distribution

In combustion process of diesel engine temperature of the charge in the toroidal chamber reaches value 2500 K, but due to the strong swirl convective coefficient is of great value, so the surface temperature of the piston crown shows lower temperature in comparison to the gas temperature. Figure 8 presents distribution of temperature in the piston material for two cases: a) engine fuelled with diesel oil and b) engine fuelled with CNG as SI engine.

In the diesel engine highest temperature occurs in the central part of the toroidal shape of the combustion chamber with temperature 285 °C. The outer surface of the upper part of the piston shows lower temperature about 200 °C. The influence of cooling oil is visible by decreasing of temperature in the ring zone and temperature on outer surface of this zone does not exceed 200 °C. Quite different is temperature distribution in the modified piston with the cylindrical chamber in the second case (CNG engine), where highest temperature occurs at outer surface of the piston crown, where it reaches value 300 °C, because combustion process of the homogenous mixture is very fast. In both cases temperature of the piston parts connected with the pin have low temperature about 150 °C. Only upper parts of both pistons transfer the heat into the piston rings.
6.4. Equivalent von Misses stresses

In theory of strength materials equivalent stresses are determinant of material durability of machine elements and the hypothesis of von Mises (or Huber) was accepted for definition of equivalent stresses. Thermal loads cause compressive stresses on the side of temperature action and lower thermal stresses occur at thinner walls, but lower mechanical stresses at thicker walls. The change of mechanical, thermal and total stresses with thickness of the wall presents Fig. 9. For that reason there is possible to optimize thickness of the wall at lowest value of stresses. In the assembly piston-pin the most loaded is the pin. Figure 10 presents calculated total equivalent von Mises stresses in the piston and pin for the case with “floating pin”. The biggest stresses occur in the piston pin and reaches value 700 MPa at the lower surface in the place of contact with small-end of connecting rod. It should be mention that such stresses take place at highest pressure and temperature for short period. The stresses in the piston material do not exceed 75 MPa in whole volume, only in the piston hubs they are greater and amount almost 150 MPa. The allowable stresses for aluminium-silicon alloy according to literature [6] amounts 75 MPa.

---

**Figure 8.** Temperature distribution in the piston body: a) diesel engine, b) CNG engine

**Figure 9.** Variation of stresses with thickness of wall
The thermal and mechanical stresses in the piston are much smaller than in the piston pin which is also ovalized by the force acting in a half its length. If in calculations the contact of the pin is omitted then the equivalent stresses are different. The second considered case without taking into consideration the piston pin but with the possible movement of the piston hubs in X and Y axis and with no possibility of movement in Z axis gives another results of calculated stresses (Fig. 11). In this case it is seen higher equivalent von Mises stresses in the side surfaces of the hub bores and surprisingly lower stresses in the central surfaces of the combustion chamber. The maximal values of equivalent stresses in the piston occur in the part of the hubs and amount about 150 MPa, but the greater part of the piston is under lower stresses less than 75 MPa at assumption of highest value of pressure 14 MPa and local temperature in the combustion chamber 2500 K.

**Figure 10.** Equivalent von Mises stresses at assumption of sliding hub in relation to the piston pin

**Figure 11.** Equivalent von Misses stresses at assumption free constraints in X and Y axis without piston pin
The third case with other constraints of the piston hubs (at fixing their position – no movement in any direction) shows quite different total stresses (Fig. 12). Assumption of such boundary conditions and constraints without taking into account the piston pin causes a change of equivalent stresses in comparison to the previous cases and greatest stresses occur in the hubs reaching value 400 MPa, which is too high for the assumed piston material. Locally they reach value 600 MPa. This case with limitation of movement of the piston hub shows that it is the worst solution of design of connection of the pin and piston.

Figure 12. Equivalent von Misses stresses at assumption fixed constraints of piston hubs

The attached figures show only distribution of stresses in the piston cross sections along the piston axis. Calculations carried out in Ansys program for stationary conditions show that the best solution of assembly piston – pin is “floating pin” in the piston hubs enabling to obtain lower stresses in the piston, but higher deformation of the piston in hub zone. The cooling channel, through which the oil flows, decreases significantly equivalent stresses.

6.5. Safety factor

The program Ansys also calculates safety factors in the piston and pin volumes which are obtained on the basis of the stresses and allowable stresses for a given material. Distribution of safety factors for three analysed cases is shown in Fig. 13 a), b) and c).

The highest values of safety factors are obtained at the first case, where whole volume of the piston demonstrates safety factors above 5, only the piston pin made from steel alloy has the safety factor below 1.0 in the half of its length. The other two cases demonstrate lower safety coefficients, particularly at fixed piston hubs the safety coefficient on the surfaces of the hub bore is lower than 1.0. Fixing of the piston hub is not a good way for the assembly piston – pin.
Figure 13. Safety factors for three different types of constraints: a) sliding piston pin, b) free constraints of hubs in X-Y axis, c) fixed piston hubs (without piston pin)

7. Discussion of results

Calculations of the piston stresses and deformation due to thermal and mechanical loads, is one of the main process during making a project of a new engine. It is difficult to measure these values in the real working engine, however, there have been carried out some experimental works with measurement of stresses in piston [11], but it is very expensive and they give only approximate values.

This part of the paper presents simulation results of thermal and mechanical loads of the piston in the diesel engine Scania DC09 074A at different constraints carried out in the module Thermal Stresses of Ansys program ver. 18.2. FEM technique allows determining equivalent stresses, deformation of the piston shape and shows the material strength by giving safety factors. Accuracy of calculations depends partly on mesh density and type of calculation volume. The strength calculations were carried out for the piston made from silicon aluminium alloy and for the piston pin made from the alloy steel at three cases of constraints. Generally it can be said that the piston pin is most exposed to destruction. On the other hand the piston material at highest thermal and mechanical load show local stresses near an allowable value 75 MPa and they appear in the piston hubs.

The carried out calculations give certain remarks:

1. The connection between the piston and piston pin in most of engines is a type of “floating pin”. For that case the calculation of the piston stresses and deformation should be carried out
at assumption of a contact of these two elements, because deformation of the piston pin influences of the pressure in the piston hubs.

2. The assumed constraints should enable the movement of the piston hubs in the axis of the piston pin and also in the transverse direction.

3. Heat transfer from the piston material to the cylinder walls and to gas in the crankcase given by assumed values of convection coefficients and temperature of the gas in crevices enables to obtain a real values of stresses and deformation of the piston.

4. Fixing of the piston hubs increases equivalent stresses in the piston hubs and does not take into account an influence of the piston pin on deformation of the piston hubs.

5. The most heat-laden surface of the central surface of combustion chamber, but the upper part of the piston has lower values of thermal stresses than the other parts.

6. Deformation of the considered piston in diameter amounts almost 0.8 mm in the zone of piston crown and also is very high in the zone of the piston pin and amounts in diameter 0.6 mm.

7. Calculations showed that piston should have and “barrel-oval” shape for obtaining a right clearance during high engine loads.

The other conclusions can be drawn on the observations of the deformations, stresses and safety factors presented in the enclosed figures.

References

[1] Heywood J B 1988 Internal Combustion Engines Fundamentals (New York: Mc Graw-Hill)
[2] Szargut J 1975 Termodynamika (Warszawa: PWN)
[3] Wisniewski S 1972 Thermal loads of piston engines, WKiL Warsaw
[4] Woschni G 1978 Experimentelle Untersuchung des Warmeflusses in Kolben und Zylinderbuchse eines schnelllaufenden Dieselmotors MTZ 42 No 2
[5] Woschni G Fieger J 1981 Experimentelle Bestimmung des ortlich gemittelten Warmeutgangskeffizienten in Ottomotor MTZ 42 No 6
[6] Wajand J A Wajand J T 2005 Medium- and high-speed combustion piston engines WNT Warsaw
[7] Vibe I I 1970 Brennlauf und Kreisprozess von Verbrennungs-motoren VEB Technik Berlin
[8] Wisniewski T 2004 Research of heat exchange processes in chosen elements of piston engines Scientific Works Mechanics Issue 203 Printing House of Warsaw University of Technology
[9] Mitianiec W Buczek K 2007 Analysis of thermal loads in air cooled SI engine Journal of KONES Powertrain and Transport Vol.14 No. 3
[10] Gonera M Sandin O 2015 Thermal Analysis of a Diesel Piston and Cylinder Liner using the Inverse Heat Conduction Method Master’s Thesis Chalmers University of Technology Goteborg
[11] Ward D M 2004 Engine piston temperature measurements for thermal loading using a fiber bragg grating (FBG) embedded into the piston surface Master’s Thesis Madison University
[12] www.ansys.com
[13] https://www.salome-platform.org
[14] Liu G R 2010 Smoothed Finite Element Methods CRC Press ISBN 978-1-4398-2027-8.