Integrated approach for stress analysis of high performance diesel engine cylinder head

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Abstract. Growing thermal and mechanical loads due to development of engines with high level of a mean effective pressure determine requirements to cylinder head durability. In this paper, computational schemes for thermal and mechanical stress analysis of a high performance diesel engine cylinder head were described. The most important aspects in this approach are the account of temperature fields of conjugated details (valves and saddles), heat transfer modeling in a cooling jacket of a cylinder head and topology optimization of the detail force scheme. Simulation results are shown and analyzed.

1. Introduction
The development of modern diesel engines is characterized by increasing engine performance primarily due to intensification of the combustion process. Some companies are actively developing engines with the highest brake mean effective pressure (bMEP) in the range of 30 to 40 bar. For example, in MTZ journal paper [1], the bMEP of the engine was increased up to 40 bar resulting in a peak firing pressure of 365 bar. Therefore, projections of increasing engine performance given in [2] should be recognized as justified. Consequently, the problem of ensuring the reliability of engine components considers that the main thermal and mechanical loads remain important.

A cylinder head is one of the most complicated engine components due to a lot of functions it performs. It closes the cylinder from the top forming the combustion chamber. It contains the key components for control of the gas exchange including intake and exhaust ducts and valves. It also transfers heat away from the combustion chamber to the engine cooling system. Therefore, improvement of calculation methods and performance criteria for cylinder heads operating under high specific loads is of considerable interest.

The specification of boundary conditions is a major challenge in modeling the stress-strain state of a cylinder head. In order to obtain them, it is necessary to estimate the temperature field of the cylinder head taking into account the hydrodynamics of the liquid in the cooling jacket. This procedure is a set of interrelated calculations schematically presented in Figure 1.

The purpose of this study is to assess the possibility of forcing a four-stroke diesel locomotive engine to the bMEP of 35 bar. Cylinder head temperatures and stresses, calculated by the procedure mentioned above (Figure 1), are the evaluation criteria.

Performance and combustion characteristics of the engine were derived from thermodynamic full-cycle engine simulation using Diesel-RK software [3]. Maximum pressure of predetermined mechanical tension of the engine was 300 bar.
2. Mathematical model

The governing equations of the considered calculation procedure are the continuum mechanics conservation laws.

a) To simulate the hydrodynamics of an incompressible fluid flow, equations for pressure and averaged flow velocity are as follows:

\[
\frac{\partial \bar{u}_i}{\partial x_j} = 0. \quad (1)
\]

Reynolds-averaged Navier–Stokes equations:

\[
\frac{1}{\rho} \frac{\partial \bar{p}}{\partial x_j} + \frac{1}{\rho} \frac{\partial}{\partial x_j} (\tau_{ji} - \tau_i), \quad (2)
\]

where \(\tau_{ji}\) - the components of the viscous stress tensor, \(\tau_i\) – additional viscous shear stresses predicted by the SST turbulence model [4].

The non-isothermal flow problem is of particular interest (for calculating the convective heat transfer coefficient). In this case, energy conservation equation is also considered:

\[
\frac{\partial T}{\partial t} + \bar{u}_j \frac{\partial T}{\partial x_j} = \frac{1}{\rho C_p} \left( \frac{\partial}{\partial x_j} (q_{\text{conv}}) \right) + \nu \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \frac{\partial \bar{u}_i}{\partial x_j}, \quad (3)
\]

where the second term on the right-hand side is the work done against viscous dissipative forces; \(q_e\) - heat transfer rate associated with the local temperature gradient. In the case of a turbulent flow, the latter is derived from the following equation:

\[
q_{\text{conv}} = \left( \frac{\lambda + C_p \mu_t}{\text{Pr}_t} \right) \frac{\partial T}{\partial x_j}, \quad (4)
\]

where \(\lambda\) and \(C_p\) are the heat conductivity and the specific heat capacity of the cooling liquid, respectively; \(\mu_t\) and \(\text{Pr}_t\) are the turbulent viscosity and the turbulent Prandtl number, respectively.

High thermal loads on the cylinder head can lead to the transient heat transfer. In this case, convective and boiling heat transfer takes place in the cooling cavities. This has a significant effect on the intensity of heat transfer and requires additional relations for the boiling heat transfer. In most published papers, the additive approach is proposed to determine the resultant thermal loads. In this study, the approach described in [5, 6] is adopted:

\[
q_w = q_{\text{conv}} + q_{\text{evap}} + q_q, \quad (5)
\]

where \(q_{\text{conv}}\), \(q_{\text{evap}}\), \(q_q\) denote the heat flux components due to single-phase turbulent convection, evaporation and quenching, respectively. The boiling heat transfer strongly depends on the cooled surface temperature. This makes it possible to use an iterative solution algorithm where the convergence criterion is the temperature increment magnitude. Such algorithms applied to internal combustion engines were implemented in [7 – 9].

b) The problem of determining the stress-strain state is solved using displacement formulation. Components of the stress tensor must satisfy the equilibrium conditions:
\[
\frac{\partial \sigma_{ji}}{\partial x_j} + \rho \cdot F_i = 0 .
\]  

At the same time, six Cauchy’s equations relate strain tensor components to the displacement vector components:

\[
\varepsilon_{ij} = \frac{1}{2} \left( \frac{\partial v_i}{\partial x_j} + \frac{\partial v_j}{\partial x_i} \right).
\]

Hooke’s law for isotropic materials closes the system of equations:

\[
\begin{align*}
\varepsilon_x &= \frac{1}{E} \left[ (\sigma_x - \mu (\sigma_y + \sigma_z)) + \alpha_T \cdot T \right], \quad \gamma_{xy} = \frac{\tau_{xy}}{G}, \\
\varepsilon_y &= \frac{1}{E} \left[ (\sigma_y - \mu (\sigma_x + \sigma_z)) + \alpha_T \cdot T \right], \quad \gamma_{yz} = \frac{\tau_{yz}}{G}, \\
\varepsilon_z &= \frac{1}{E} \left[ (\sigma_z - \mu (\sigma_x + \sigma_y)) + \alpha_T \cdot T \right], \quad \gamma_{zx} = \frac{\tau_{zx}}{G}.
\end{align*}
\]

The finite-element formulation for modeling the stress-strain state of the cylinder head can be obtained based on variational principles.

3. Simulation of fluid hydrodynamics in the cooling system

The liquid distribution in each cylinder head can be considered uniform since the individual coolant supply is organized. Based on the liquid motion in the cooling water jacket of the cylinder head, the following regularities can be noted.

The fluid flow colliding with the intermediate bottom loses its speed and produces swirling motion. As a result, the coolant inflow to the intervave region decreases that leads to formation of stagnation zones.

The bypass channel in the intermediate bottom of the cylinder head determines the flow intensity in the regions of the valve bridges and the fuel injector sleeve. The highest flow speed is in a region (between the two exhaust ports) of cooling channels of valve seats. The lowest flow speed is in a region between intake and exhaust ports as well as in the periphery of the cylinder head next to the exhaust valve (Figure 2a). Based on the obtained data, it is possible to compare the resultant thermal loads on the cylinder head with the flow intensity in cooling cavities. High heat transfer coefficients (7000…9000 W/ K·m²) are obtained at the intermediate bottom. Low heat transfer coefficients (1500…2000 W/ K·m²) correspond to zones with a low flow speed (Figure 2b).

**Figure 2.** a - liquid velocity field in the cylinder head at 15 mm from the heat-release surface; b - heat transfer coefficient on the heat-release surface; 1 – intake port; 2 – exhaust port.

**Cylinder head thermal state simulation.** The use of the flow boiling model and taking into account the local distribution of the heat transfer coefficient over the cooled surfaces allow one to improve the method for determining the thermal state of the cylinder head. At the same time, it is important to correctly define the heat transfer conditions over the remaining surfaces of the cylinder.
head. Mutual influence of temperature fields of the related parts (valves and valve seats) also plays an important role. The latter should be taken into account by means of the resulting temperatures of inlet and exhaust valve facets that allow avoiding direct contact modeling of the parts. The temperatures are determined iteratively. Therefore, determination of the thermal state of the cylinder head requires two sequential iterative algorithms.

The main steps in determining the boundary conditions for ports and the combustion chamber have been presented in several papers (e.g. [8]). Finite element models of parts consisting of approximately 16 million elements were used for thermal state modeling. The simulation results are the temperature distributions of the related parts (Figure 3).

The maximum temperature of the intake valve seats (431 °C) is higher than that of the exhaust valves seats. The reason for this is that the latter were cooled by the liquid. The maximum temperature of intake valves is 529 °C and the same for exhaust valves is 631 °C. The maximum temperature of the cylinder head is 425 °C in the region between the air start valve and the intake port (Figure 3a). Some reasons of this are as follows: high intensity of the thermal load from the combustion chamber, low thermal conductivity of the material of uncooled valve seats, presence of stagnant zones in the cooling cavities (Figure 2b).

![Figure 3. Parts temperature distribution: a – cylinder head; b – valve seats](image)

Therefore, the temperatures for a given engine performance reach critical values. However, the authors of [10] have noted that profiling the cooling cavities reduces the maximum temperature of the cylinder head by 25-30 °C. Therefore, such design optimization of the cylinder head is also a crucial task [11]. For example, in [1] the presence of drilled channels in the cylinder head of the engine with the MEP of 35 bar led to a reduction the maximum temperature to 380 °C.

**Thermal stress state simulation of a cylinder head.** In modeling, both mechanical loads and thermal stresses induced by uneven temperature distributions were taken into account. The major purpose the study was the estimation of maximum stresses. In addition, the level of stress intensity in elements of a cylinder head determined during the simulation provides the information to evaluate tight sealing and thermo-mechanical fatigue. However, these are beyond the scope of this study.

The finite-element model of assembly consisting of a cylinder head, a cylinder liner, a cylinder block and a gasket was used in the simulation. The cylinder head is held against the cylinder liner top collar by four studs screwed into the cylinder block.

The lack of symmetry and the presence of nonlinear contacts require the use of submodeling procedure to ensure an acceptable accuracy of the solution. Submodeling procedure consists of the following steps: setup and run the global model; submodel creation with refined mesh; global displacements interpolation onto the submodel at the cut boundary locations; evaluation of the submodel results.

In this study, submodeling is performed on the portion of cylinder head adjacent to the combustion chamber. Both the global and submodel finite-element models consist of approximately 800 thousand elements. The loading conditions were as follows: the cylinder head temperature distribution, the
maximum pressure of 300 bar, the stud preload of 600 kN. The simulations results are the stress and strain distributions of the assembly elements (Figure 4).

**Figure 4.** Thermal stress state of the cylinder head assembly: a – stress intensity distribution; b – total deformation (x100)

There are compressive stresses in the cylinder head bottom on the side of the combustion chamber. The values of these stresses in the region between intake valves reach 554 MPa, and in the region between exhaust valves - 495 MPa (Figure 5a). The tensile stresses in the region of water jacket openings are approximately 250 MPa. Maximum compressive stresses in the cylinder head bottom on the side of the cooling cavities are located close to the cylinder head outer wall (Figure 5b). The maximum value of compressive stresses in the region of ports entries is 705 MPa.

**Figure 5.** Minimum principal stresses: a - on the side of the combustion chamber; b - on the side of the cooling cavities; 1 – intake port; 2 – exhaust port

Maximum stresses in the cylinder head are located in the periphery of the cylinder head bottom on the side of the cooling cavities rather than in a region between valves on the side of the combustion chamber (as it usually is [12]). This indicates the predominant role of stud preloads caused by the increase engine performance and the necessity to ensure gas-tight sealing between cylinder head and cylinder liner.

To confirm the above-mentioned assumption, the stress-strain state of the cylinder head subjected only to the stud preloads was calculated. The calculation procedure and finite-element models remained unchanged. Compressive stresses in the zones previously identified in Figure 5b slightly increased: Point 4 - 733 MPa, Point 5 – 700 MPa.

One way to improve the design is the topology optimization, for example “Element Birth and Death Method”. It excludes elements from consideration when predefined conditions are met (e.g. \( \sigma_{\text{eqv}} < \sigma \)). Afterwards, the stress-strain state of the modified structure is evaluated.

For further structural optimization of the cylinder head, initial design was modified by removing unloaded elements (Figure 6). The primary mechanical loads were the stud preloads. The allowable equivalent stress was assumed 150 MPa. During the calculation, there were 24 iterative cycles of elements birth / death.

Based on the numerical results the cylinder head design was supplemented with reinforcing elements (Figure 6a) as well as with elements transferring stud preloads to the cylinder liner top collar through the gasket. It should be noted that the increase in stresses on the periphery of the cylinder head
is determined by the bending moment in the bottom caused by the stud preloads. Therefore, it is advisable to change the design of the cylinder head as shown in Figure 7.

![Figure 6](image1.png) ![Figure 7](image2.png)

**Figure 6.** A force diagram of the cylinder head: a – internal contours; b – cylinder head skeleton of the remaining elements

**Figure 7.** Cylinder head: a – initial design; b – modified design;

The stress-strain state of the modified cylinder head (Figure 7b) subjected only to the stud preloads was also performed. Maximum compressive stresses (Points 4-6 in Figure 5) did not exceed 230 MPa.

Used construction modifications allow the horizontal supply of coolant to the cylinder head cavities that is more preferable in accordance with [11]. Such coolant supply configuration can be realized by means of an intermediate bushing. Similar technologies are used for marine engines of well-known manufacturers (e.g. MAN, Sultzer).

4. **Conclusion**

1. The proposed calculation procedure allows estimating the influence of wide range of factors on working capacity of investigated cylinder head.

2. Thermal stress analysis of the cylinder head under high performance engine conditions showed that the initial design does not provide a permissible level of temperature and stresses and requires modifications.

3. The proposed cylinder head design enabled obtaining a significantly lower level of stresses and ability to implement the alternative coolant supply configuration.

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