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THERMAL LOADING OF STATIONARY ENGINE CYLINDER HEAD

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Summary: *Cylinder head is a complex assembly with both thermal and mechanical loading. Analysis of steady state temperatures including effects of local boiling in cooling passages partially verified by experimentally measured temperatures is the first part of presented work. The second one deals with mechanical loading of head including assembly loads, pressure loads in cylinder and temperature loading. Heat transfer analysis has acknowledged importance of including local boiling assumption into analysis. Structural analysis results were not fully evaluated yet. Partially, influence of valve seats deformations due to assembly, pressure and thermal loading onto contact pressure between valves and seats distribution is significant.*

1. Introduction

Numerical heat/stress analysis of engine head was carried out by *Josef Božek Research Center of Engine and Automotive Engineering*. C/28 series engine head was released for this purposes by manufacturer (ČKD Motory, a.s.) in the form of full CAD model. Developed FE model is assigned to analyze both thermal and mechanical loads.

Cylinder heads of diesel engines need to house intake and exhaust valve ports, fuel injector and complex cooling passages. In addition, as they are directly exposed to high combustion pressures and temperatures, they experience severe thermal and mechanical loading under engine operation. Compliance of all these requirements leads to many compromises in design. To avoid risk of fail in operation (deformations, cracks) due to overheating in regions of limited cooling, resulting from the compromises, is the target of engines designers. Usually, the design of head must be adapted accordingly to the operational experience. The costs of such approach are usually large. Cylinder head operational conditions FE modeling is appropriate complementary alternative in comparison with operational testing only. Mechanical engineers require both numerical evaluation/simulation and laboratory/operational testing, to produce high-quality designs. Computational engineers‘ drive for the most appropriate model depends on actual hardware and software power, costs and possible benefits following from the analyses. Getting experience with such tasks, verification of software ABAQUS applicability in it was the main motivation of our work. *ČKD Motory* - producer of stationary diesel engines in our country – have released design documentation of C 28 series engine for this research purposes.

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2. Assembly and FE model

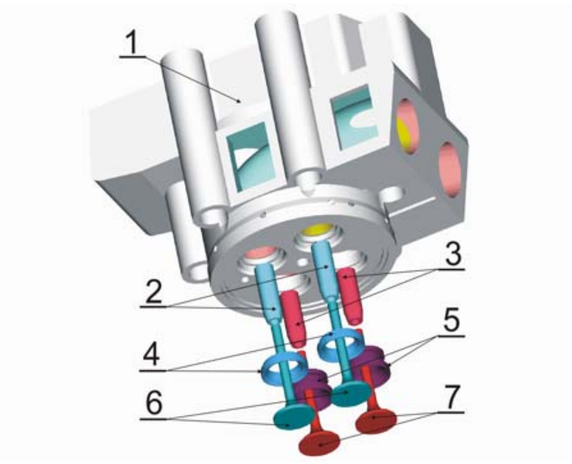


Fig. 1: Assembly scheme

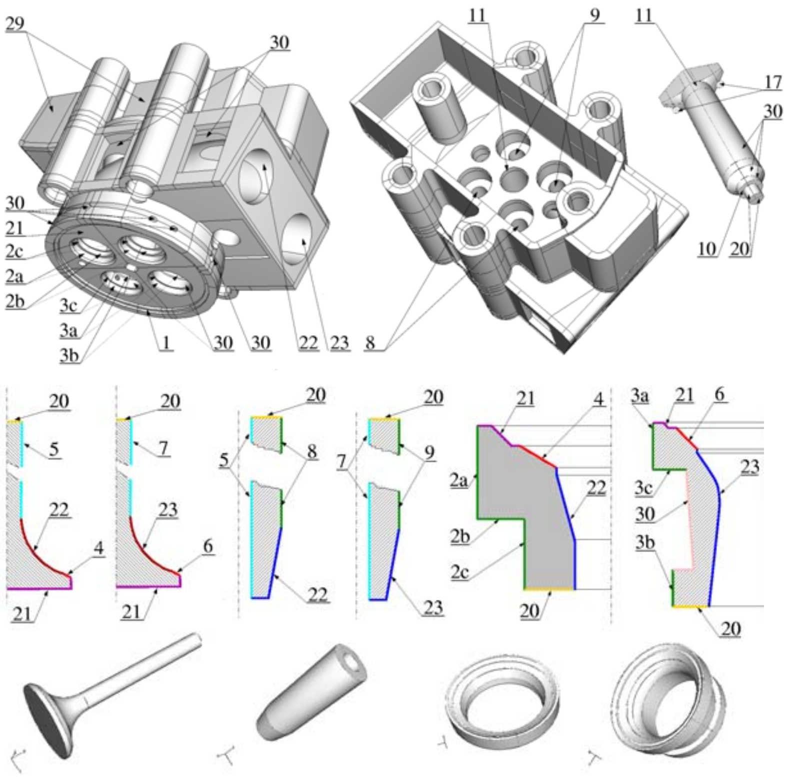
The main parameters of C 28 direct-injection diesel engine commonly used in power generation units-are: bore 275 mm, stroke 330 mm, mean effective pressure 1.96 MPa and nominal speed 750 rpm. The cylinder head (Fig 1, link 1), made of cast iron, contains two intake (Fig 1, link 6) and two exhaust (Fig 1, link 7) valves made from forged alloy steel. The valve guides (2,3) and seats (4,5) are pressed into the head. The exhaust valves seats are directly water-cooled. The fuel-injector is situated into the axis of the cylinder.

Combustion pressure and heat are the most important operational loads. Heat transfer analysis, resulting into temperature field over all assembly, is typically the first step of cylinder head. Following mechanical analysis consists of assembly loads step, averaged and extreme pressure step, and heating.

All components mentioned above are included in developed FE model. Some parts of valves and fuel-injector were significantly simplified or completely left out. Thermal contact interactions are described by heat flux ρ_{AB} from surface A to B. It depends on temperatures θ_A, θ_B accordingly to

$\rho = k(\theta_A - \theta_B)$,

where k is contact heat transfer coefficient



Boundary condition description	Link	Heat transfer coefficient [$Wm^{-2}K^{-1}$]	Bulk temp. [K]
Insulated surfaces	20	0 (adiabatic)	–
Free surfaces (contact with ambient air)	29	5	320
Cooling passages	30	3 000*	350
In-cylinder surfaces	21	450	1120
Intake-port surfaces	22	800	330
Exhaust-port surfaces	23	800	700

Fig. 2: The interactions and boundary conditions on parts of the cylinder-head assembly. The boundary conditions considered in the FE

Boundary conditions are described by heat flux ρ from free surface (with temperature θ) to surrounding environment (with bulk temperature θ_0) by

$$\rho = h(\theta - \theta_0),$$

using heat transfer (film) coefficient h. Both k and h may depend on surface temperature- $k = k(\theta)$, $h = h(\theta)$.

Thermal loading involves the heat fluxes from burning and the convection from the burned gases exhaust. All these parameters are periodically varying in time. However, considering the speed of the periodical changes with comparison of thermal inertia of all the components of cylinder head, the computation is performed assuming steady-state loading using the averaged values. Boundary conditions - the in-cylinder as well as intake and exhaust port heat transfer coefficients and bulk gas temperatures - were obtained from the detailed thermodynamic analysis of the engine using the 0-D thermodynamic model OBEH [1] based on Eichelberg's empiric equation.

Possibility of exceeding cooling water boiling point was expected. Therefore jump of film coefficient $h(\theta)$ near the boiling point of cooling water is considered (see Fig. 3).

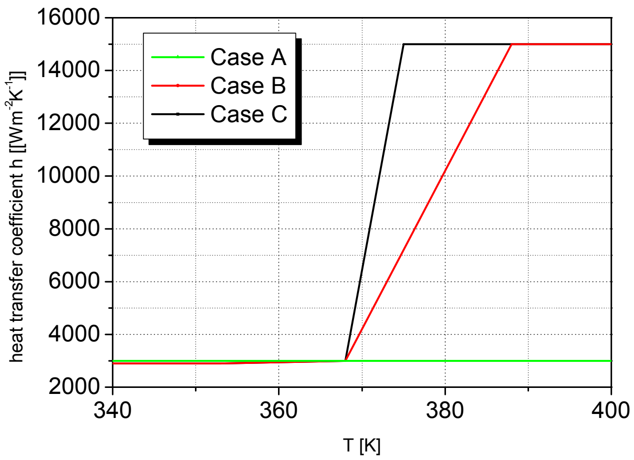
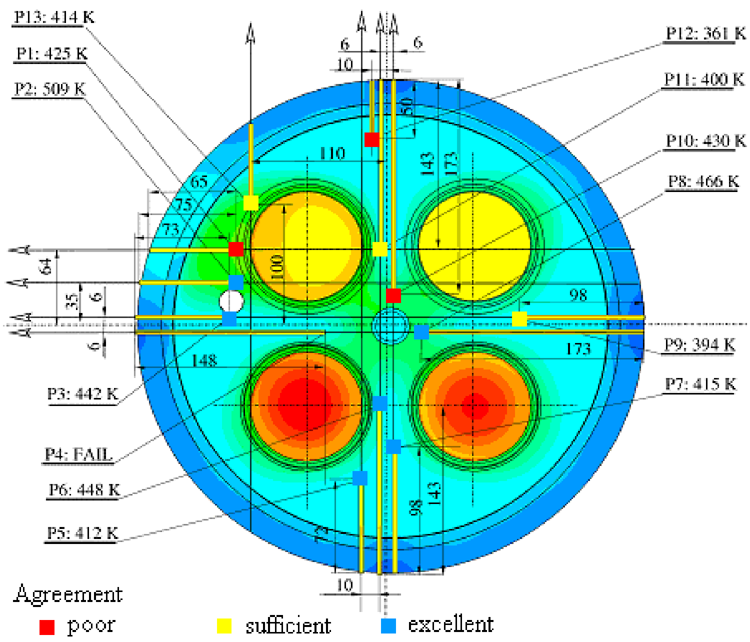


Fig. 3: The cooling passages heat-transfer coefficient dependency on the surface temperature. 3 approximations were used when tuning the model: case A (local boiling ignored), case B (weak jump), case C (severe jump). Case B was found sufficient and it was used in final analysis.

Contact interactions between head and valve guides/seats, valves and guides/seats, head and gasket ring, pre-stressed bolts and valve springs are included in structural analysis of the head. Five basic states were solved. 1) **assembly**: gasket ring is constrained in cylinder side; head is bolted on cylinder gasket ring by six pre-stressed bolts fully constrained in cylinder side; valve seats/guides are pressed into head using contact constrains; valves interact with guides by special MPC constrains, with seats by contact constrains; pre-stressed valve springs are inserted between valve and head; fuel injector is constrained on head bottom inner surface by contact and pressed onto it by 2 pre-stressed bolts. 2) **average pressure**: all assembly loads; head bottom outer surface and valves bottoms are loaded by average in-cylinder pressure $p = 1.96$ MPa. 3) **maximum pressure**: all assembly loads; head bottom outer surface and valves bottoms are loaded by maximum in-cylinder pressure $p = 12$ MPa. 4) **maximum pressure and temperature**: all assembly loads; all maximum pressure loads; temperature field from previous steady state heat transfer analysis. 5) **average pressure and temperature**: all assembly loads; all average pressure loads; temperature field from previous steady state heat transfer analysis.

3. Results

Experimentally determined temperatures provided by the engine manufacturer, were compared with computed results. The thermocouples were placed in special bores. All the bores were situated at the distance of 18 mm from the bottom margin of the cylinder head (Fig 4). Despite of a lack of further detail information on conditions of the experiment (errors caused by the measuring equipment, influence of the location and fixation of the thermocouples in the bores, etc.), the authors found the data provided as useable and useful resource for the verification of the presented model.



Measured point	Temperature [K]	
	Computed	Measured
1	**571**	**425**
2	**551**	**509**
3	**441**	**442**
4		**-**
5	**440**	**412**
6	**459**	**448**
7	**446**	**415**
8	**511**	**468**
9	**446**	**394**
10	**523**	**430**
11	**453**	**400**
12	**446**	**361**
13	**481**	**414**

Fig. 4: Comparison of computed and measured temperatures.

The direct comparison is disabled by the fact that the experimental data and the computed values do not correspond exactly to the same load of the engine. Thermal boundary conditions for the FE analysis were computed for about 5% higher BMEP (brake mean effective pressure) than that prevailing during experiment. Accordingly to [2], the experimental temperatures for the same BMEP as that used in FE analysis might cause an increase in the observed temperatures of about 15 K. Fig. 4 provides the comparison of the computed results with those from measurement. Regarding some uncertainty about the precision of the placement of quite lengthy bores, the sensitivity of the calculated values on the positioning of measuring points was tested. Therefore, all the

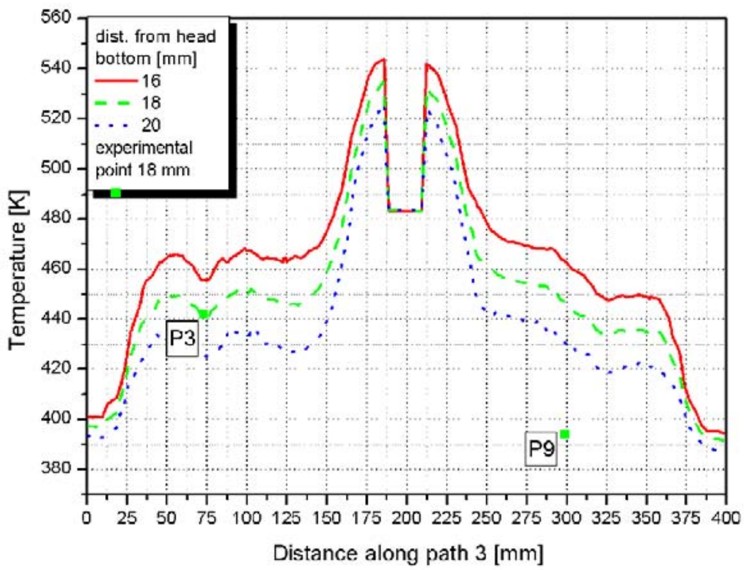


Fig. 5: Temperature along path P3-P9

temperatures were observed at the distances of 16, 18 and 20 mm from the bottom margin of the cylinder head. The comparison displayed in Fig. 5 affords a rough estimation of the possible errors arising from the inaccurate fit of the points where the temperatures observed in model to the real measuring points.

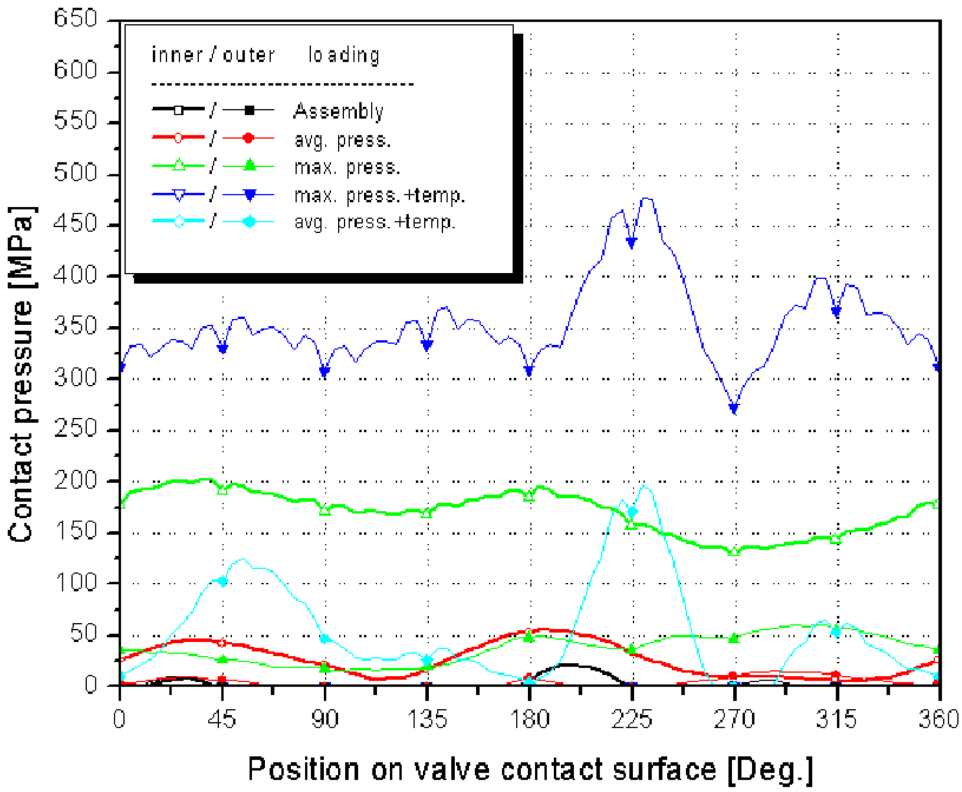


Fig. 6: Contact pressure between intake valve and port

As an example of structural analysis contact pressure distribution between intake valve and seat is plotted on figure 6. Idealized contact surface is conical. Two edge circles of this surface establish inner / outer path the contact pressure is mapped in. Position on both inner and outer circles is measured as angle coordinate in cylindrical system associated to valve axis. Curves on figure 6 document strong dependency of valve/seat contact on temperature loading. While in cold state inner edge transfers more loads, in hot one the outer edge is simply overloaded.

5. Conclusion

Experimental data provided by the engine manufacturer, were compared with computed results. The thermocouples were placed in special bores, at the distance of 18 mm from the bottom margin of cylinder head. Heat transfer analysis has acknowledged importance of including local boiling assumption into analysis. Structural analysis results were not fully evaluated yet. Partially, influence of valve seats deformations due to assembly, pressure and thermal loading onto contact pressure between valves and seats distribution is significant.

Acknowledgement

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[2] Macek, J. – Vítek, O. – Vávra, J.: *Kogenerační jednotka s plynovým motorem o výkonu větším než 3 MW –II.* ČVUT v Praze, Fakulta strojní, 2000. (in Czech)