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Considerations about the Thermal Requirements of the Cylinder Heads of Air-Cooled Compression Ignition Engines Used in the Construction Industry

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Abstract – **When designing a cylinder head for two families of stationary, air-cooled, direct-injection diesel engines to drive electric current generators or water pumps in the construction industry, we must take into account the following aspects: BMEP, which is limited in these engines at values of 3.8-5.2 bar. The paper presents the optimal solution found in the functional analysis performed for a FLUENT program with a single-cylinder engine and a family of two-cylinder engines, considering each cooling air flow case. Another aspect to consider when designing cylinder heads is to determine the thermal regimes of the cylinder head, piston, cylinder, valves and nozzle, to define the development of heat per engine work cycle. The objectives of this paper are to define optimal construction solutions for a cylinder head for a direct injection air-cooled Diesel engine having a stroke of 65 mm and a bore of 82 mm.**

Keywords* – *analysis, construction industry, cylinder head, direct injection, heat release

1. INTRODUCTION

The cylinder head is designed for direct injection Diesel engines, which include mono and poly-cylinder, we proposed the solution inlet and outlet channel placement on the same side of the cylinder head. For this reason, the injector should be placed between the rocker arms, which brings the advantage of shortening the injection pipe. Also, this solution provides a large surface design of the cooling air reception, and the thermal regime of the injector is much better controlled than in other types of air-cooled cylinder heads, because the injector is not in contact with a wall of the hot channel.

2. EXPERIMENT DESCRIPTION

In Table 1 there are presented the main features of the engine cylinder head. For the 5.5 bar BMEP wish we got in our combustion code simulations the evolution of pressure, temperature and apparent heat release, as shown in figure 1. In the figure 2 we showed, în the same conditions, the evolution of apparent heat release and cumulative heat release.

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Table.1 Engine specification

Bore / *Alezaj* [mm]	82
Stroke / *Cursa* [mm]	65
Compression ratio / *Raport de comprimare*	19,2
Displacement volume / *Cilindree* [cm^3]	343
Engine speed / *Turație* [min^{-1}]	3000
BMEP / *Presiune medie efectivă* [bar]	5,5

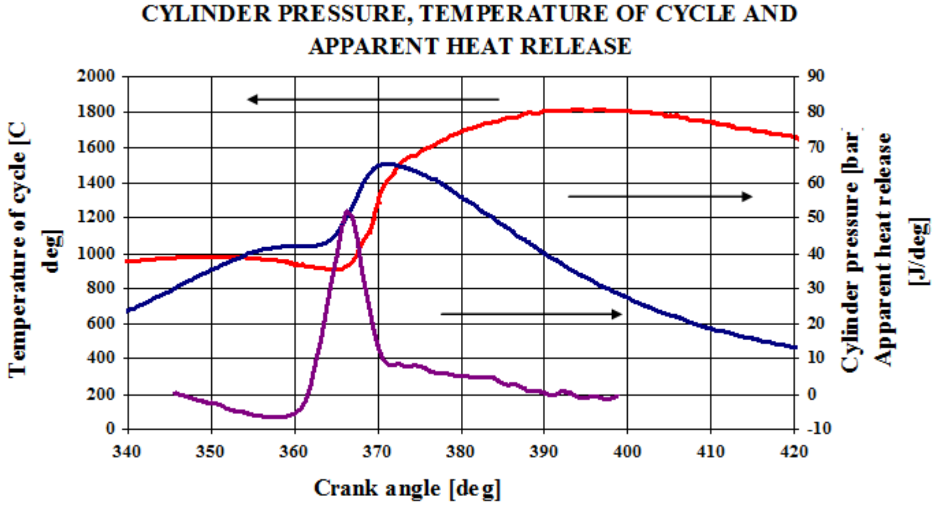


Fig. 1 In cylinder evolutions of pressure, temperature and aparent heat relase simulated for a BMEP of 5.5 bar

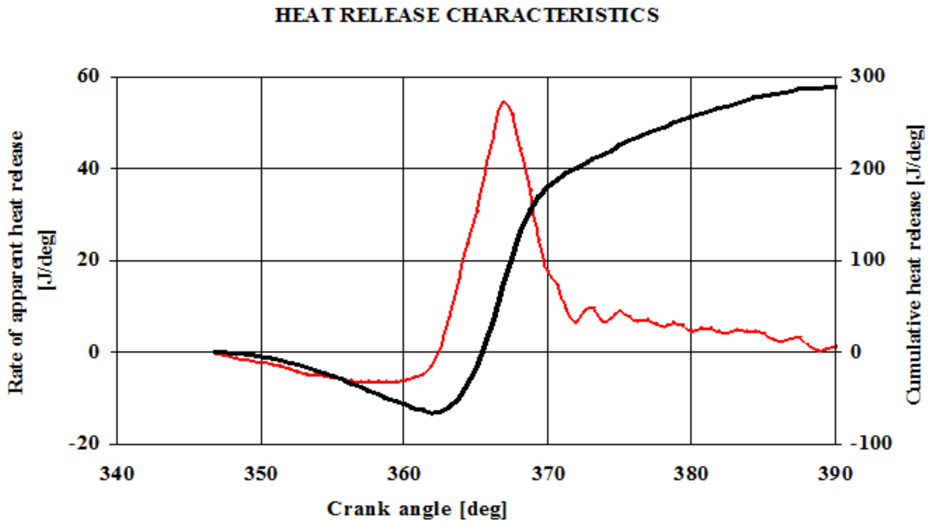


Fig. 2 Evolutions of aparent heat relase and cumulative heat release

The momentary above piston cylinder volume is given by ecuation (2) where B is bore and Vc is combustion chamber volume.

To characterize the evolution of in cylinder available volumes we established the movement of the piston from top dead center (TDC), as can be seen in equation (1):

$$S(\alpha) = r\left[1 + l_b^1 - \cos(\alpha) - \sqrt{\left(l_b^1\right)^2 - \sin^2(\alpha)}\right] \tag{1}$$

$$V_{cyl}(\alpha) = \frac{\pi B^2}{4} S(\alpha) + V_c \tag{2}$$

Where.

S(α) – distance crossed of biston TDC;
r – cranck radius;
$l_b^1 = \frac{l_b}{r}$
l_b - conecting rod length;
α – cranck angle.

In figure 3 is showed the momentary above piston cylinder volume evolution.

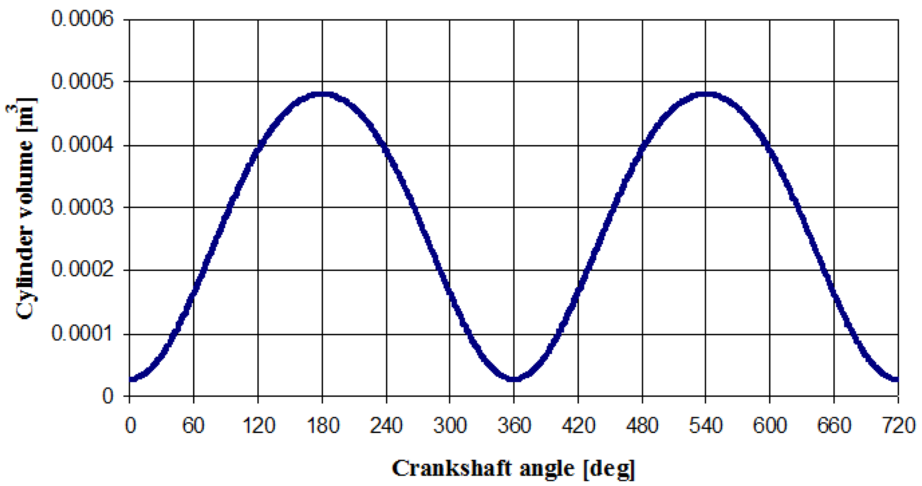


Fig. 3 Evolution of the momentary above piston cylinder volume

In order to determine the gasses temperature in the compression step we used equation (3), where:

μ_g - residual gas fraction;

γ - adiabatic exponent of real gas= $\frac{c_p}{c_v}$;

p_e - exhaust absolute pressure;

p_i - intake absolute pressure;

T_i - intake absolute tamperature.

$$T(\alpha)=\frac{1-\mu_g}{1-\frac{1}{\gamma\varepsilon}\left[\frac{p_e}{p_i}+(\gamma-1)\right]}\cdot T_i \tag{3}$$

Instantaneous gas temperature in combustion phase is estimated by formulae (4):

$$T(\alpha)=\frac{p(\alpha)\cdot V(\alpha)}{\frac{8315}{M}(m_{air}+m_{fuel}+m_{res})} \tag{4}$$

To validate the heat transfer coefficient of the combustion chamber we applied Woschni correlation (4)[4], where:

B –cylinder bore;
P – average gasses pressure;
T – gasses temperature;
W- average of gasses velocity.

In figure 4 it can be seen a section through the cylinder head that is the object of our study. In the drawing you can see the section for tracing the cooling air between the intake and exhaust channels. This section is dangerous because the thermal stresses, given by the temperature differences, generate high mechanical stresses of thermal origin.

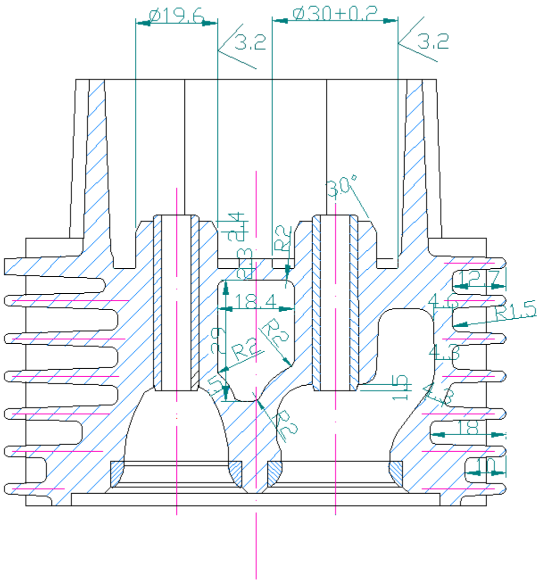


Fig.4 Section through the cylinder head

Woschni's correlation for average heat-transfer coefficient can be seen in equation (5)

$$h_c(W/m^2\cdot K)=3.26B(m)^{-0.2}\,p(kPa)^{0.8}\,T(K)^{-0.55}W(m/s)^{0.8} \tag{5}$$

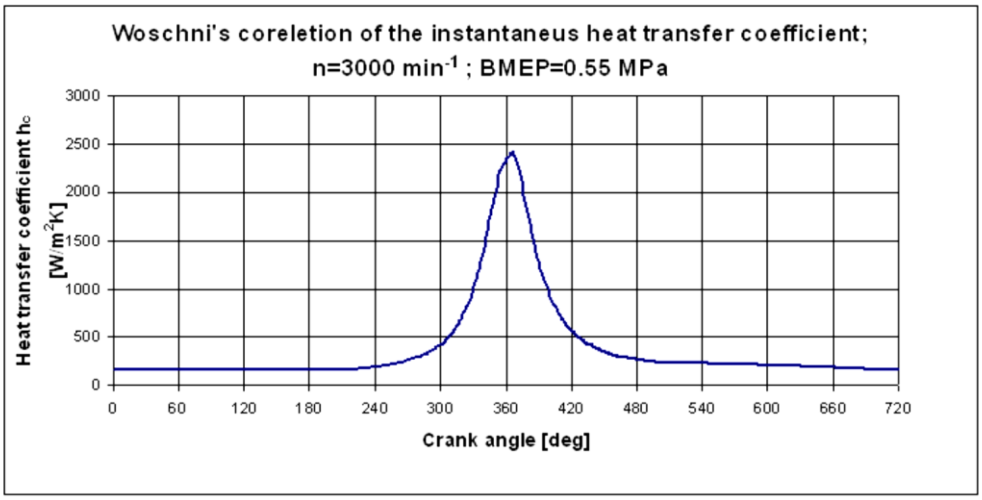


Fig. 5 The average heat-transfer coefficient

To analyze the gas flow channels and the determination of temperature and thermal stress we were use the cylinder head designing in CATIA V5, as shown in figure 6.

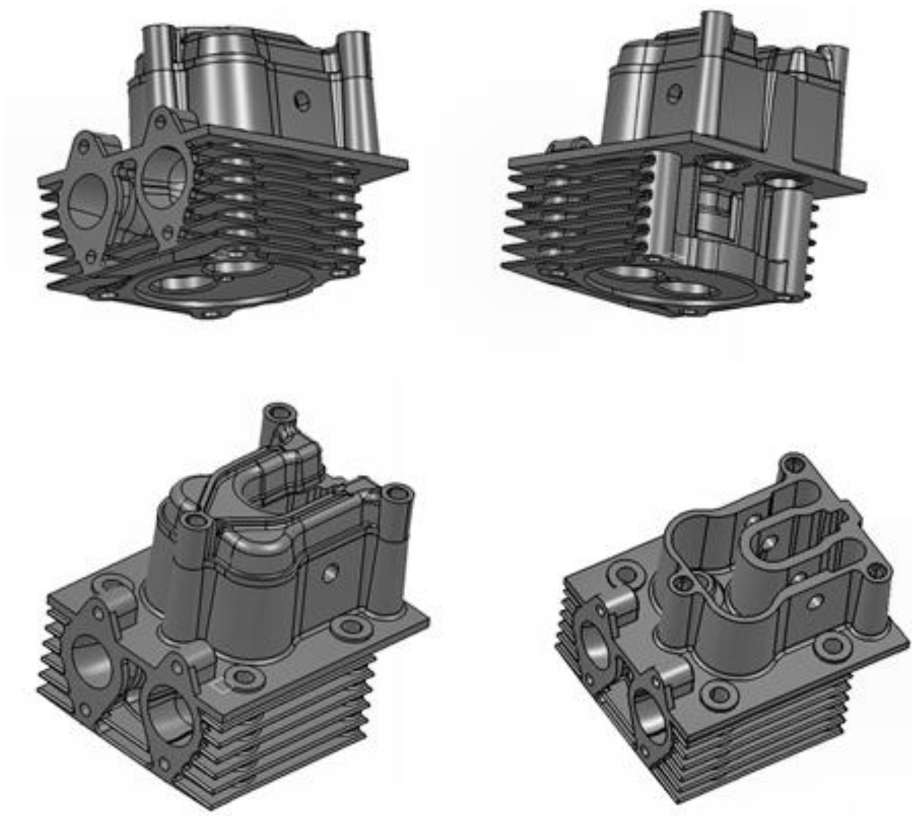


Fig. 6 Cylinder head designing in CATIA V5

The duct was placed above the cylinder just like the cylinder head geometry. Inlet valve could be moved like in table 2, where: Hv – lift of valve, Dv – diameter of the intake channel. Dv=28 mm.

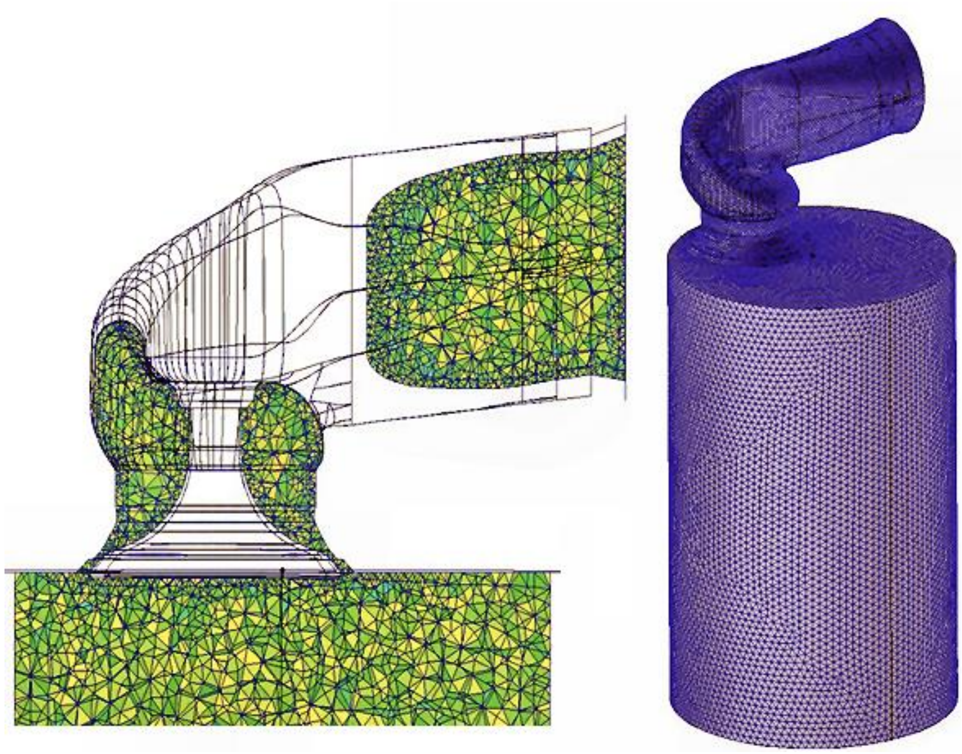


Fig. 7 Finite element mesh for inlet geometry generated for FLUENT analysis

Table 2 Inlet valve characteristics

No.	H_v/D_v	$H_{v\,[mm]}$
1	0	0
2	0,04	1,12
3	0,08	2,24
4	0,12	3,36
5	0,16	4,48
6	0,2	5,6
7	0,24	6,72
8	0,28	7,84

In the figure 8 is show the flow for Hv=1.12 mm and in figure 9 is show Hv=4.48 mm. In Figure 10 it can see the pressure field for described flow conditions.

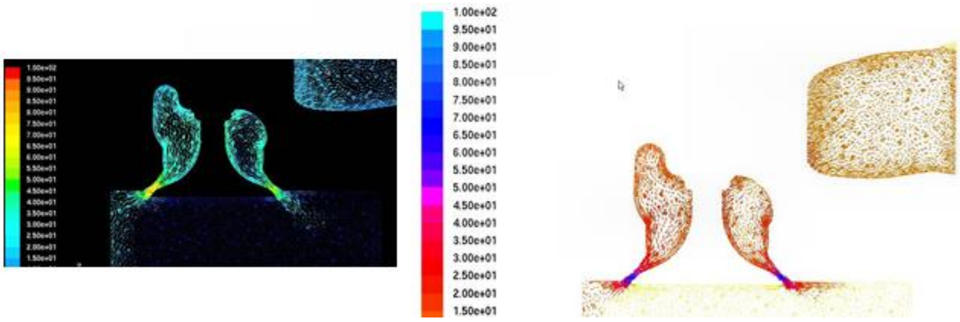


Fig. 8 Flow simulation for Hv=1,12 mm

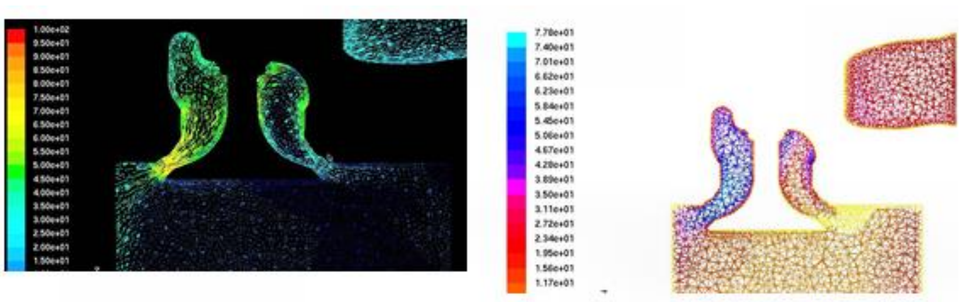


Fig. 9 Flow simulation for Hv=4,48 mm

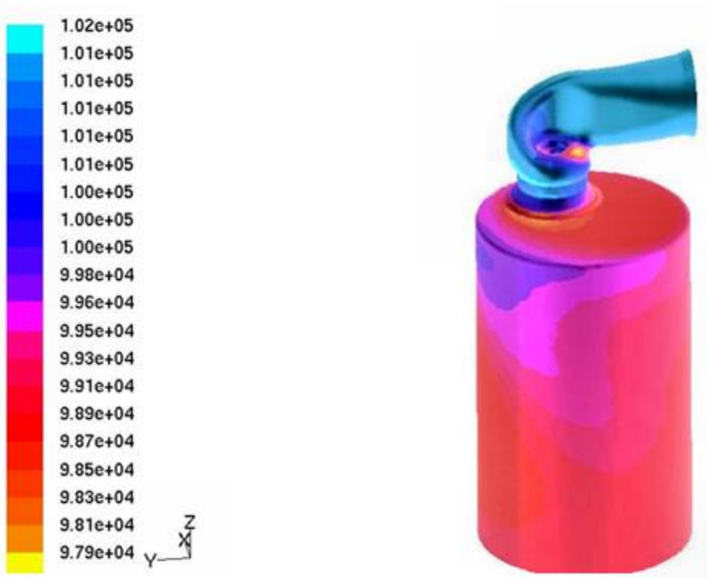


Fig.10 Pressure field in the described flow condition

Stress analysis of the thermo-mechanics origin fields

The item type is used for thermal analysis DC3D10 coding. This is a typical thermal analysis. It is an element of order 2 with 10 nodes tetrahedral type. Thermal analysis was run with the idea of obtaining the temperature field structure analysis. The thermal field was loaded over static analysis. For static analysis we used two-order tetrahedral elements with 10 nodes. Coding element in ABAQUS is C3D10. The analysis of static stress field was obtained on the structure analyzed.

Number of elements in the analysis: 262 509

Number of nodes in the analysis: 451 281.

Based on heat transfer coefficient we obtained the plot temperature and stress distribution, which can be seen in figure 11.

Plot stress and temperature distribution for inlet can be seen in figure 11, and for exhaust duct is presented in figure12.

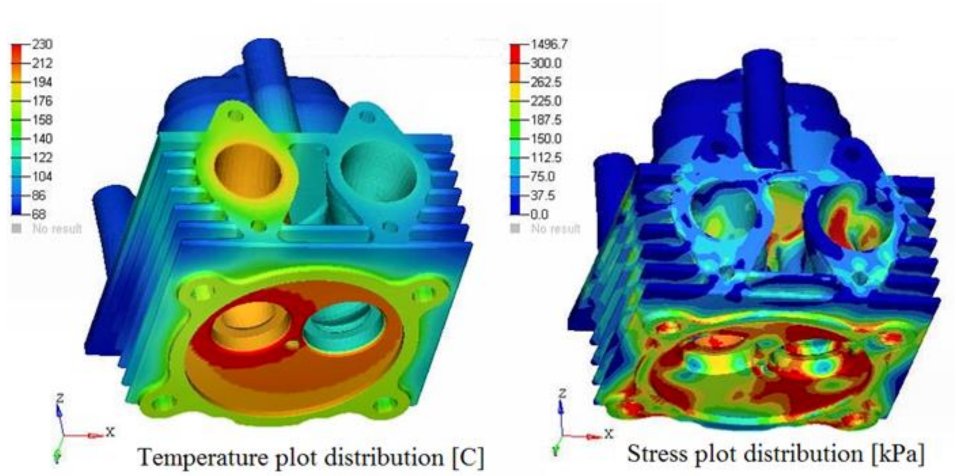


Fig. 11 The plot temperature and stress distribution

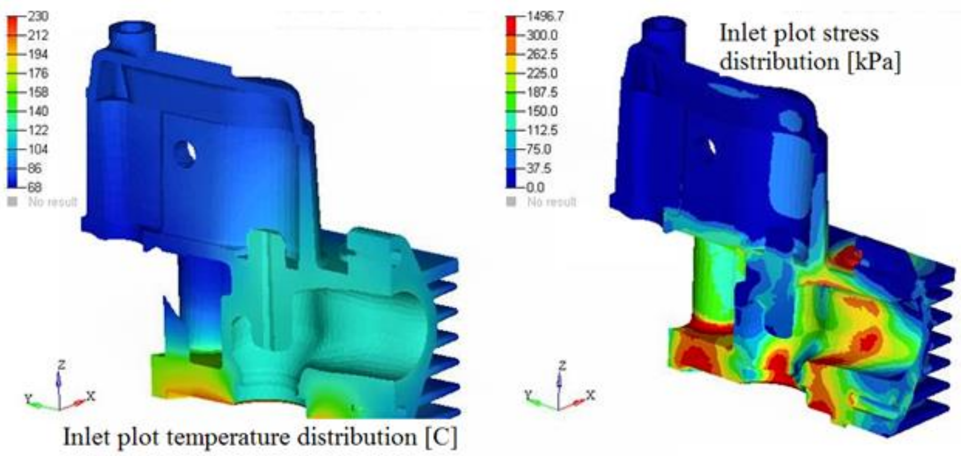
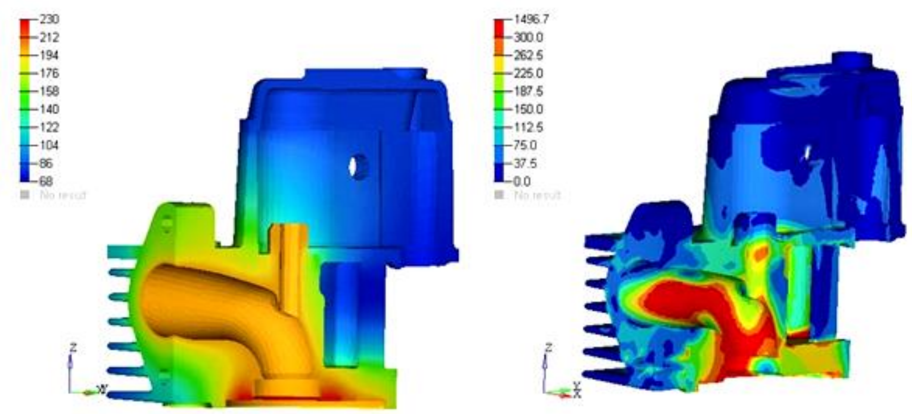


Fig. 12 Inlet temperature and stress distribution



Exhaust plot temperature distribution [C] Exhaust plot stress distribution [kPa]

Fig. 13 Exhaust temperature and stress distribution

3. CONCLUSION

Results presented in this work could be used as a preliminary step in a development of two families of Diesel engines, just in terms of the main parameters of the cylinder head and engine performances.

At a cylinder capacity of 0.343 liters for the breake mean effective pressure of 0.55 MPa, the engine's effective power at 3000 rpm will be 4.7 kW. For the operation of electric current generators and motor pumps in the construction industry, the power absorbed must not be greater than 75% of the maximum power of the thermal engine. In the presented simulations, the cylinder head was dimensioned for maximum power, so it will cope without problems in operation.

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