

ANALYSIS OF THERMAL LOADS IN AIR COOLED SI ENGINE

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Abstract

In the spark ignition engines with air cooling system the thermal loads of the cylinders and cylinder heads are much bigger than in the water-cooled engines and the loads are not uniform on the whole surface of these elements. Recently most of the cylinders and cylinder heads of the two-wheel vehicles had long ribs that caused a bigger noise and an increase of the mass of the elements. The paper includes the modelling process of the thermal loads and temperature distribution of the cylinder and cylinder head of the two-stroke engine 115 cm³ capacity cooled by air at mean engine load. The results of simulation carried out in ANSYS program with creation of the mesh in CATIA were verified by the experiment on the real engine. The results obtained in the tests and the simulation show, that long ribs have not an effect on the decrease of thermal loads of the cooling elements. The paper includes the colour graphs with deviation of the parts, temperature on the surfaces and thermal stresses.

Main work results allow finding that the cooling heat in the air-cooled two-stroke engine increases with the engine speed and cooling energy, boundary condition enables determine the convection coefficient for heat exchange calculations and FEM enables the prediction of the surface temperatures of the walls, the accuracy of temperature depends on the determination of the outflow temperature of the air.

Keywords: combustion engines, heat transfer

1. Introduction

The air-cooled engines have the worse work condition than the liquid cooled engines. The cooling of the surface of the cylinder and cylinder head is irregular caused by the impossibility of a good distribution of the air around these parts. Particularly in the multi-cylinder engines each of the cooled parts has a different temperature on the same surfaces. The cooling of the one-cylinder engine is much easier, however the front side of the cooling parts is cooler than the rear side. It is important how to distribute the air by applying of shields and ribs and also how to ensure the suitable mass flow rate of the air. The paper describes the cooling of the spark ignition two-stroke engine of small power. These types of engine are popular in different applications and mostly are cooled by the air caused by movement of the engine in the small vehicles (motorcycles, scooters, motorbikes) or by an additional blower. The specific details of heat flows in the reciprocating internal combustion engines are the critical points of the designing of the cooling system. Thermal efficiency is directly affected by the magnitude, time dependence and location of the thermal energy flows. Many parameters influence on the heat flow such as power output, fuel efficiency, knock tolerance, emissions [4]. The heat flow through the walls to the cooling air is depended on the convection on sides, conductivity of the walls (two-layer wall in the cylinder) and the heat stream from the combustion chamber. Big amount of the heat is taken by air during the combustion and expansion strokes. The air-cooled two-stroke engines typically operate at high power outputs, normally greater than 75% of full load rate power. Substantial heat transfer from the exhaust gases to the port occurs during the exhaust port opening. Typically in the computer program the heat transfer rate is estimated from the cylinder pressure, unburned and burned gas

temperatures, combustion chamber surface area and wall temperature, assuming gas velocity inside the cylinder scaled with the mean piston speed. In this work the cooling heat was measured on the working engine by determination of the air mass flow rate and the input and output temperatures. The cooling heat ratio was determined in a function of rotational speed and the temperatures of the surfaces were measured. On the other hand by knowing the cooling heat, the heat exchange between the gas inside the cylinder and the cooling air was modelled by using FEM (ANSYS). The temperatures on the cylinder and cylinder head were calculated and compared with the measurements.

2. Flow of cooling heat

In a two-stroke engine the cooling heat flow contains only two factors: conduction and convection. The radiation presents only small part of the total cooling heat because of absence the soot in the exhaust gases. The heat of the conduction is transferred through the wall of the cylinder sleeve, cylinder body, cylinder head, the piston, the exhaust duct and piston rings. The transfer of the conduction heat \dot{q} per unit area and per time in a steady state condition is given by Fourier's law [8]:

$$\dot{q} = -k\nabla T, \quad (1)$$

where: k is the thermal conductivity. For a steady one-dimensional temperature variation:

$$\dot{q}_x = \frac{\dot{Q}}{A} = -k \frac{dT}{dx}. \quad (2)$$

The convective heat transfer is based on the equation:

$$q_{conv}(t, A) = h_g(t, A)[T_g(t, A) - T_w(t, A)], \quad (3)$$

where: the temperature T and heat transfer coefficient h_g are time and spatially dependent. The model arrives at the convection coefficient through the use of the analogy, which links the heat transfer across the boundary layer to the skin friction C_f and the effective fluid velocity U_{eff} at the edge of the boundary layer [4]:

$$h_g = 0,5C_f \rho U_{eff} C_p \text{Pr}^{0,66}. \quad (3)$$

The skin friction is modelled using a skin friction correlation for a flat plate boundary ($a=0,046$) layer and fully developed pipe flow ($a=0,067$):

$$C_f = a \left(\frac{\rho U_{eff} \delta}{\mu} \right)^{-0,25}, \quad (4)$$

where: μ is the dynamic viscosity coefficient and δ is the thickness of the boundary layer.

Knowing the cooling heat power in a function of the crank angle and its distribution on the cylinder head, piston and cylinder areas one is possible to calculate the temperatures of the walls by using zero-dimensional model at the assumption of the equal temperatures and coefficients for the considered areas.

Determination of the wall's temperatures is carried out in a 3-dimensional real model by using FEM technique in the commercial computer program. The computation is based on the simple unsteady heat transfer equation in the solid materials with density ρ for a small cell with volume dV in the object [3]:

$$\lambda \cdot \nabla^2(t) \cdot dV \cdot d\tau = c \cdot \rho \cdot dV \cdot \frac{\partial t}{\partial \tau} d\tau, \quad (5)$$

where: t is the temperature and τ is time.

After simplification the formula can be presented in other form:

$$\frac{\partial t}{\partial \tau} = \frac{\lambda}{c \cdot \rho} \cdot \nabla^2(t). \quad (6)$$

This formula enables determination of temperatures in the different places of the walls. In the case steady heat flow the equation has the simplest form:

$$\nabla^2(t) = 0. \quad (7)$$

3. Test stand of air-cooled two-stroke engine

The one-cylinder two-stroke engine Robin EC12 is cooled by the centrifugal blower (Fig.1). The experimental stand included the pipe with diameter D on the cooling air inlet. The velocity u_0 and temperature t_0 of the inlet air were measured by the anemometer, which was located in the centre of the pipe. Thus the air mass flow rate could be defined and also the thermal energy of the inflowing air was determined. By measurement of temperature of the hot air on the outflow (rear side of the cylinder) the thermal energy in state 2 also was determined. The difference between the air thermal energy in state 2 and energy in state 0 is the cooling heat stream. Comparing the cooling power and total energy of the fuel consumed by the engine the cooling heat ratio was determined as the function of rotational speed.

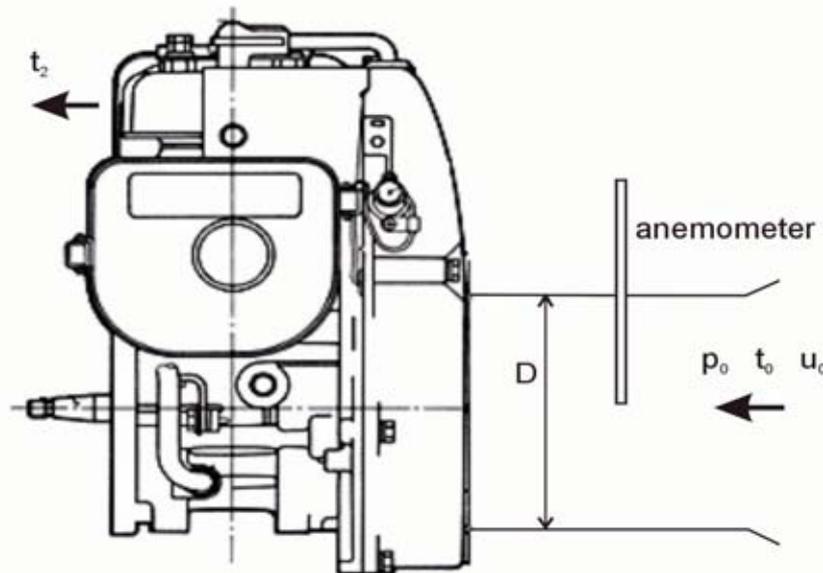


Fig. 1. Diagram of the cooling heat measurement in the two-stroke engine Robin EC12

4. Results of engine tests

The carburetted engine was running at full load with the standard equipment at middle rotational speeds. The speed characteristic of the net power, fuel consumption per hour and specific fuel consumption is shown in Fig. 1.

The cooling power was determined by measurements of the inlet mass flow rate and temperatures on the inlet and outlet sides. In FEM technique one should determine the convection coefficients from the cylinder inner and outer sides. This requires the knowledge of the mean velocity of the air between ribs, which can be found by dividing the mass flow rate by the flow cross section area and density of the air. The cooling power and air velocity between ribs are shown in Fig.3. The increase of the cooling power and velocity of the cooling air is observed.

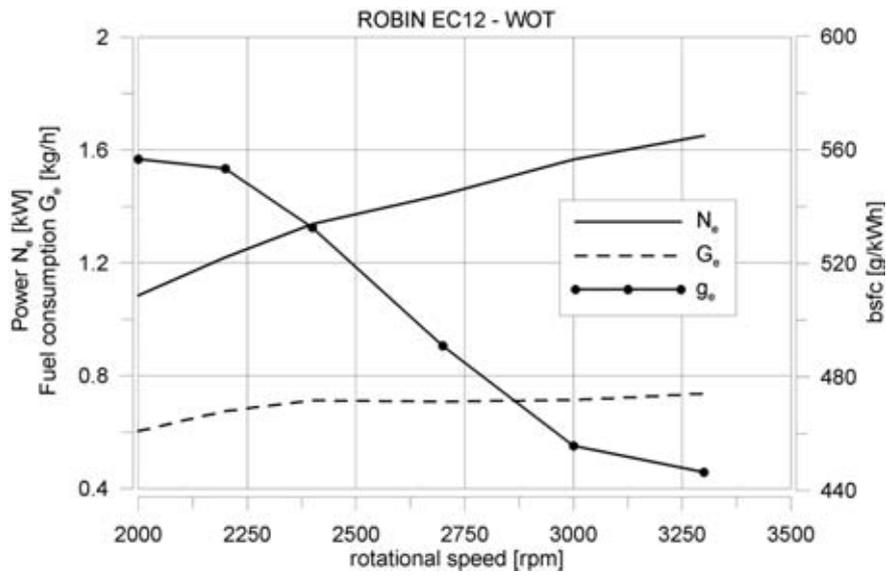


Fig. 2. Engine power and fuel consumption at full throttle opening

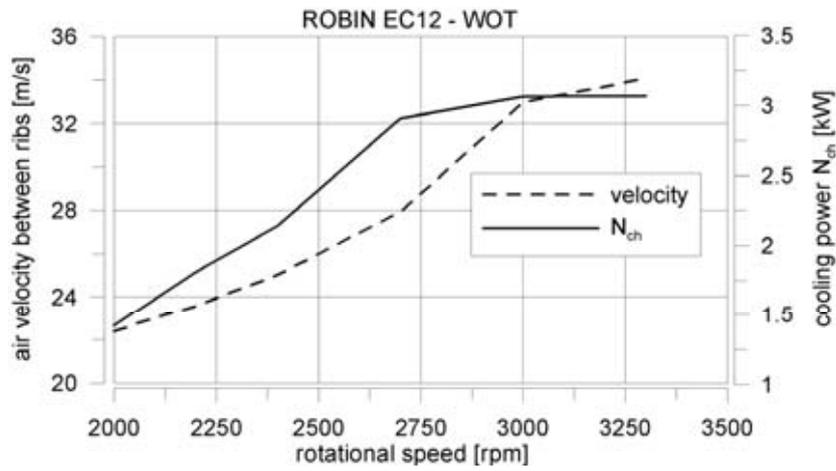


Fig. 3. Cooling power and velocity of the air between the engine ribs

The total efficiency of the engine was very low (below 20%). In comparison to the useful power the cooling heat power amounted from 20% at low speed 2000 rpm to 35% at 3000 rpm. The cooling heat in the engine is almost twice higher than the effective energy.

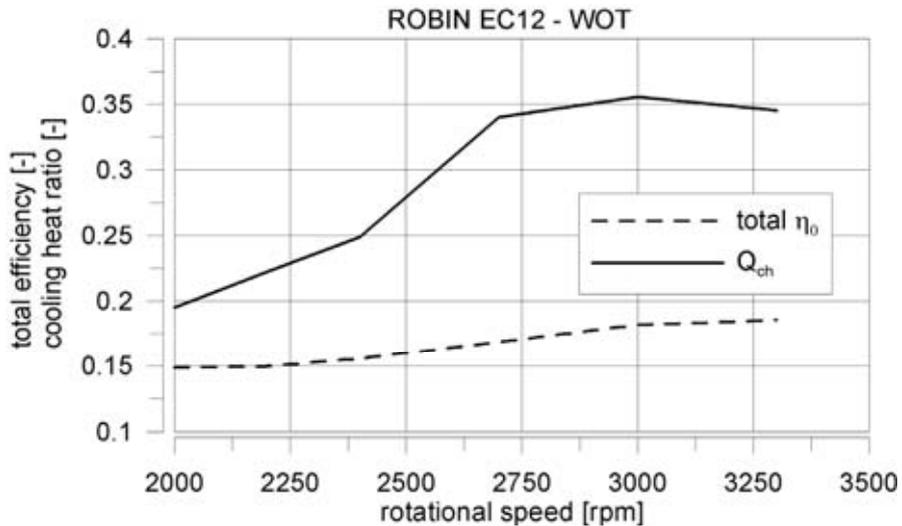


Fig. 4. Total engine efficiency and the cooling heat ratio

5. Fem calculations

On the base of the measurements and calculations the cooling heat power was determined (Fig.3). Determination of the temperatures of the surfaces of the walls was predicted by using FEM technique in the program ANSYS [9] for engine speed $n=2700$ rpm. At first the spatial model of the cylinder and cylinder head were created in CAD system (Fig.5) including also the model of the cylinder iron sleeve, which is alloyed during chill casting. The accuracy of the model was verified by measurements of the modelled and the real parts and the results are shown in table 1. The relative difference of the masses does not exceed 1,5%. After the creation of the models they were translated from the CAD system to ANSYS Workbench program in order to do the meshing and calculations of the thermal loads and thermal deformations of the parts.

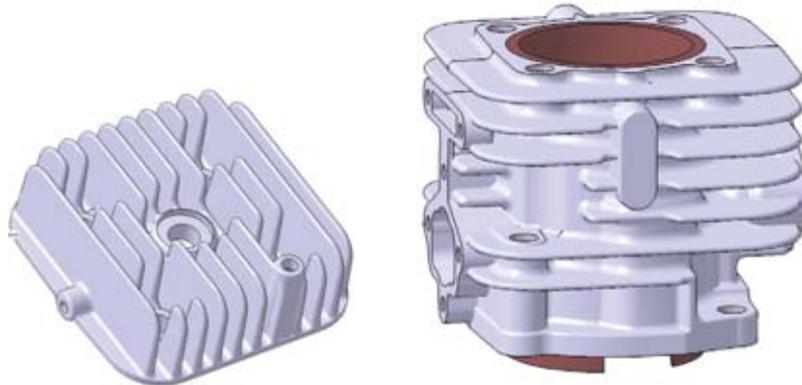


Fig. 5. The CAD models of the cylinder head and the cylinder

Tab. 1. Comparison of the real and modelled parts

	Real	CAD 3D	Relative difference
Cylinder mass	1250 [g]	1241 [g]	0,72 [%]
Cylinder head mass	380 [g]	385 [g]	1,32 [%]

According to the theoretical considerations the heat flow in the engine parts can be treated as quasi-state. In such approach the calculations in FEM are simpler and only single-time solution of the global equations is carried out and a well accuracy of the calculations is obtained. In the transient conditions the calculations in the next step are based on the results obtained from the previous step. The discretization of the models is shown in the Fig.6 basing on the cells with four walls (pyramids cells). The calculations were carried out for rotational speed 2700 rpm with cooling heat power 3kW (Fig.3). It was assumed the percentage distribution of the heat transfer for the parts. Some boundary conditions are shown in table 2.

Tab. 2. Boundary condition in the FEM calculations

Name	Definition	Value	Unit
Cooling power	N_{ch}	3,0	kW
Cooling heat ratio of the cylinder head	Q_{ch}	70	%
Cooling heat ratio of the cylinder	Q_{cyl}	30	%
Air velocity between the ribs	w	28	m/s
Exhaust gas temperature	T_{exh}	720	°C
Temperature of the inlet air	T_{int}	40	°C
Kinematic viscosity of the air	ν_{int}	1,70E-05	m ² /s
Specific heat of the inlet air	$c_{p, int}$	1,005	kJ/kg K
Heat conductivity of the inlet air	λ_{int}	0,0271	W/(m K)
Kinematic viscosity of the exhaust gas	ν_{exh}	1,18E-04	m ² /s
Denisty of the exhaust gas	ρ_{exh}	0,3530	kg/m ³
Specific heat of the exhaust gas	$c_{p, exh}$	1,1411	kJ/kg K
Heat conductivity of the exhaust gas	λ_{exh}	0,06754	W/m K

A big amount of the heat is flown to the piston crown, however this heat is next transferred by the piston rings to the cylinder walls. The results obtained in FEM analysis shown the temperatures distribution in the parts and heat flux.

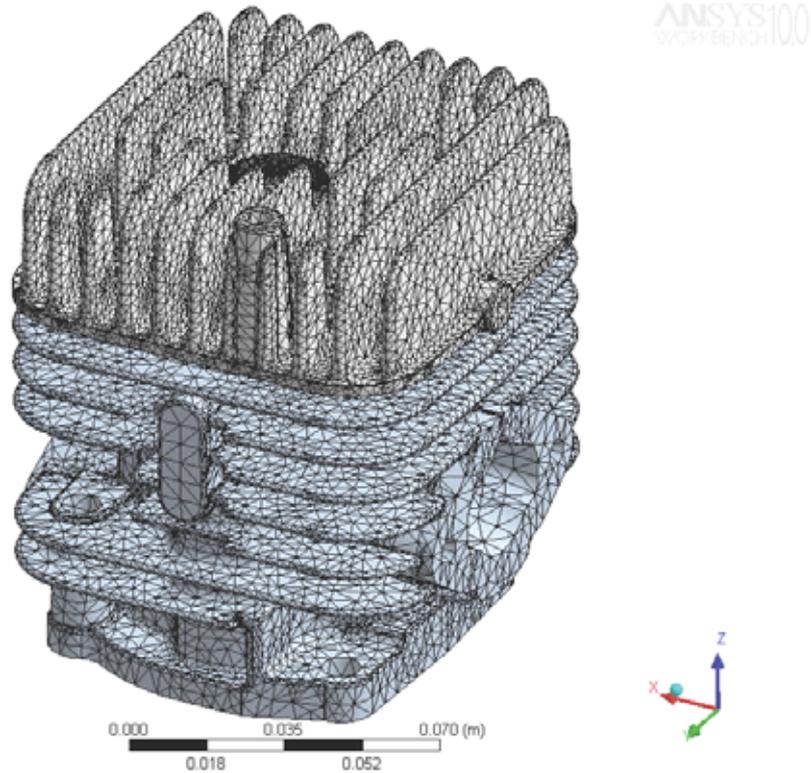


Fig. 6. Mesh of the part for calculations in FEM (ANSYS)

It was assumed that the flow of the air in the spaces between ribs takes place with the same velocity. The biggest thermal load is seen on the cylinder head (Fig.7), particularly near the spark plug. Calculation value of the temperature amounts almost 270 °C. The outer ribs of the cylinder head have the temperature considerably lower 78 °C and can be reduced to lower height.

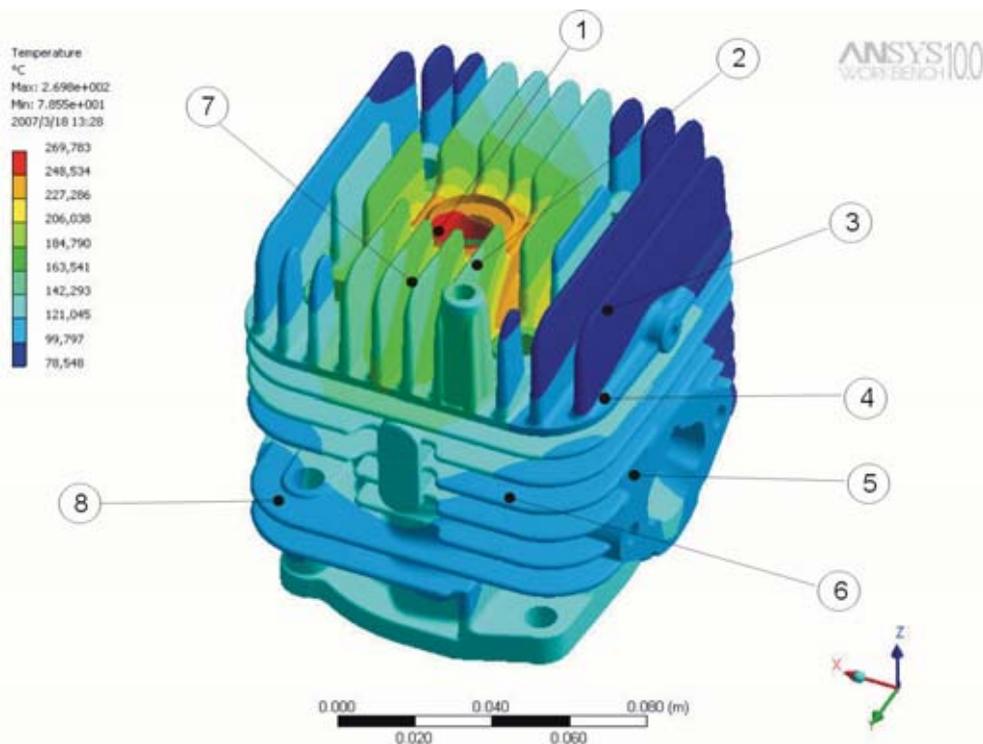


Fig. 7. The calculated and measured (marks with number) temperatures on the model surfaces

The ribs of the cylinder have lower temperature than the ribs of the cylinder head – only 100 - 120°C. In real conditions the flow velocity of the air between ribs has not the same value. In the calculations the thermal loads of the cylinder head is almost symmetrical at assumption of the same convection coefficients (Fig.8). The one of the most important factors is to definition of the radius between the rib and the main plate of the cylinder head or the cylinder wall. It should have the big value, which is possible in order to obtain lower value of the heat flux. The distribution of the heat flux in the cylinder head is shown in Fig.9. The higher heat fluxes are in the bottom of the main plate between the ribs.

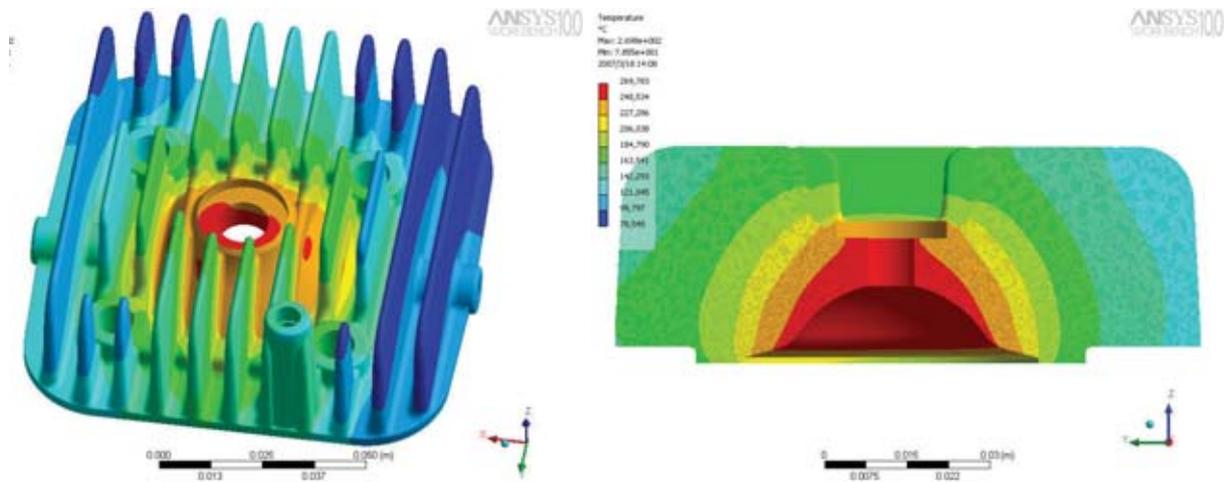


Fig. 8. The temperature distribution on the cylinder head

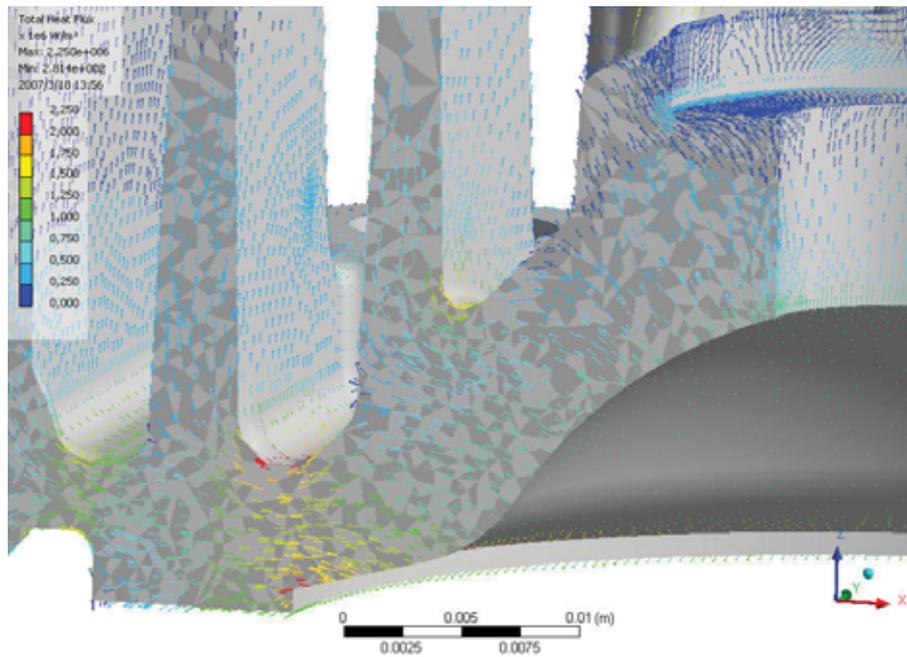


Fig. 9. Total heat flux $[W/m^2]$ in the cylinder head

The distribution of the temperature in the cylinder is shown in Fig.10. The highest temperature amounted 220 °C is only in the narrow area above the piston rings and in the lower part of the cylinder the temperature has lower value than the combustion temperature of the lubricating oil. In Fig.7 the marks with the number are the points of the real temperature measured on the walls by the thermocouples at rotational speed 2700 rpm. One obtained the following temperature values in Celsius degree: 1 - 230; 2 - 150; 3 - 70; 4 - 91; 5 - 130; 6 - 89; 7 - 210 and 8 - 89. One measured much more points on the walls. It was found the difference between measurements and calculation values 10 – 20 °C, because of the unknown distribution of the heat transfer to the wall in the calculations.

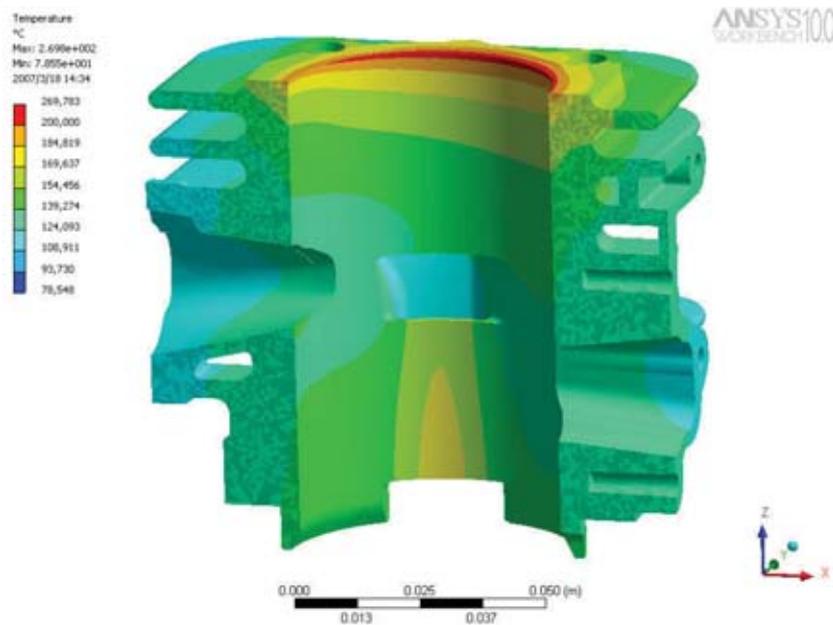


Fig. 10. Distribution of the calculated temperature in the cylinder

6. Conclusions

1. The cooling heat in the air-cooled two-stroke engine increases with the engine speed and cooling energy amounts above 30% of the total energy delivered with fuel.
2. The simple method of the cooled energy was developed in the air-cooled engine.
3. The measurement of the mass flow rate of the cooled air was the boundary condition to determine the convection coefficient for the heat exchange calculations.
4. FEM enables the prediction of the surface temperatures of the walls, however there are still the difficulties with determination of the distribution of the heat flux to the walls.
5. The highest temperature in SI two-stroke engine is in the area of the spark plug, and the outer ribs can be lessened in order to decrease the weight of the parts.
6. The accuracy of the measured and calculated temperature depends on the determination of the outflow temperature of the air, which is not equal in the spaces between the ribs.

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