HEAT TRANSFER IN COMBUSTION CHAMBER OF PISTON ENGINES

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Abstract

Measurements were made during downward propagation of a flat flame in a lean limit methane/air mixture. It was observed that the limit flames are very thick and that heat losses to the walls caused extensive cooling of the product gases within 0.05m behind the propagating flame. Analysis of all data from well synchronized measurements as well as the results of following to them numerical calculations made possible to understand in detail physical picture of heat transfer from the hot gases to the cold walls. It is a unique way for explanation of heat transfer phenomenon. Measurements and calculations were used to prepare heat balance in a flammability tube. In the next step measurements were carried out in a rotating cylindrical combustion chamber. This chamber was treated as a model of piston engine combustion chamber with swirl. Temperature changes on a surface of the rotating combustion chamber were transmitted towards motionless data acquisition system. The temperature measurements were carried out with the use of special thermocouples located at the front wall of the chamber. Influence of rotation velocity on heat transfer was investigated in the range from 1000rpm to 5000 rpm. For a constant mixture concentration maximum temperature rise depended on the rotation velocity. Increased rotation velocity initially intensified heat transfer to the wall, with following to it increase of peak temperature, but further rise of rotation velocity resulted in local flame extinction near the side walls, decreased volume of combustion gases, and as a result of these, decreased value of peak temperature at the wall surface. Presented techniques make possible estimate heat losses to the walls in a combustion chamber of piston engines.

Keywords: *internal combustion engines, heat transfer, heat flux, combustion chamber, heat losses*

1. Introduction

In recent years, many researchers have considered to the rate of heat release in cylinder, rate of heat transfer and characterization of thermal energy transport as key parameters [1-5]. These factors affect engine performance, fuel economy, and emissions, as well as life of components such as piston, rings and valves. Knowledge of transient and local heat transfer characteristics is very important in estimating the temperature distribution in the piston, combustion chamber walls, valves, and valve seats. Heat transfer problems in a cylinder of an internal combustion engine are very important because they affect engine performance, efficiency and emission. Conduction and convection modes of heat transfer are the most important [7-10].

Heat is transferred in engine through fluids in turbulent motion and between the fluid and solid surface in relative motion. Heat is mostly transferred by forced convection between the in-cylinder gases and the cylinder head, valves, walls and piston during induction, compression, combustion, expansion and exhaust processes. Usually values of instantaneous heat flux into the engine cylinder wall elements have been obtained from measurements of the instantaneous surface temperature. The temperature variation at the wall is a result of the time boundary conditions at the gas-wall surface. Measurements have to be made just at a surface by means of very thin thermocouples or resistance thermometers.

In this work, a method of heat flux density evaluation was tested with the use of the standard flammability tube. In the next step measurements were carried out and heat flux density was calculated in experiments with a rotating cylindrical combustion chamber treated as a model of a piston engine combustion chamber with swirling charge.

2. Square tube test stand

The investigations were conducted in a vertical, 51 mm, square flammability tube, 1800 mm long. The tube was equipped with gas temperature and wall surface temperature gauges offered by Medtherm Corporation, to measure gas temperature profiles and heat losses to the wall during the passage of flames for downward propagation of lean limit flames. The gas temperature gauges were Pt-Pt10%Rh thermocouples made from 25 μm wire with a ceramic potted transition. Gauges for wall surface temperature measurements were thermocouples consisting of iron-nickel (Fe-Ni) film layers 0.5 μm thick deposited at the end of a thermally and electrically insulating ceramic rod of mullite. The film layers overlapped in the centre of the end of the rod to form the thermal junction. Sensitivity of the film thermocouples was 10.45 μV/°C and thermal diffusivity of ceramics was equal to 1.058 mm2/s. Signals from gauge were amplified and after that recorded on the screen of Tektronix oscilloscope.

3. Test results of square tube

Typical picture of a flame propagating downward from the open upper end of the tube to its closed bottom end in a limit mixture (6%CH4) is shown in Fig. 1.

Fig. 1. Schlieren picture of flame pr opagating downward in a lean limit mixture (6%CH4) . Dark patch seen at the top part of the tube is made by condensation of water vapour

To understand the mechanism of heat transfer from combustion gases to the walls it was necessary to determine a thermal flame structure and the amount of heat transferred to the walls. Gas temperature measurements showed fluctuations of gas temperature behind flame front generated by natural convection (Fig. 2).

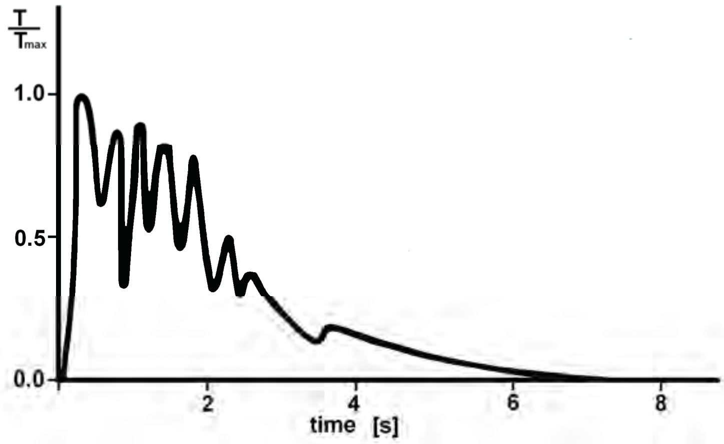

Fig. 2. Typical record of temperature measured behind flame front shown in Fig. 1

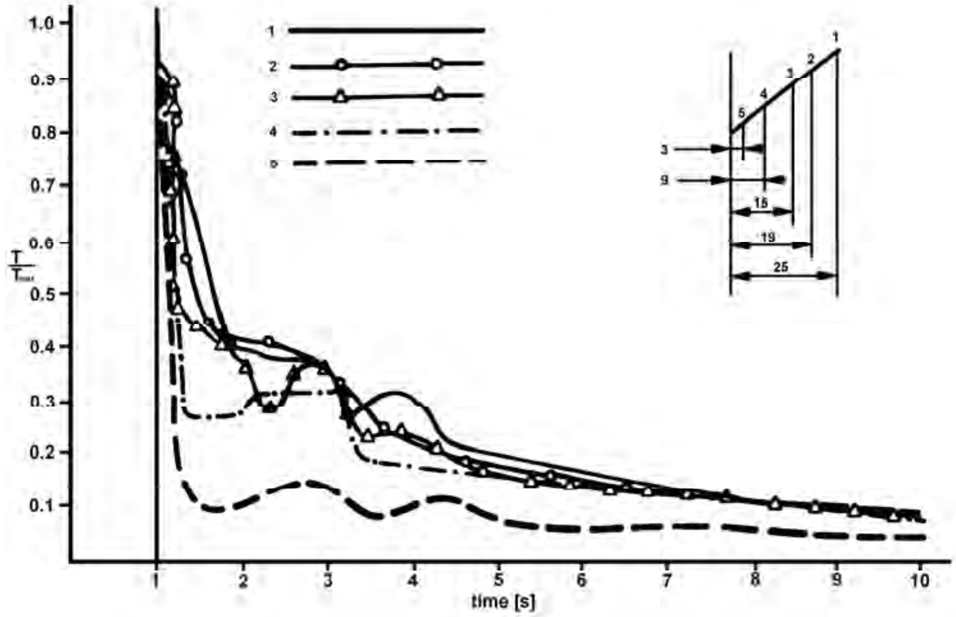

Fig. 3. Temperature curves as the envelopes of the lowest points of measured curves like that shown in Fig. 2

From the point of view of flame extinction there are important the lowest values of recorded temperature. Temperature curves as the envelopes of such values are shown in Fig. 3. Location of the thermocouples (in mm) is shown on the right-hand side of this figure. Thermal flame structure resulting from these curves is indicated in Fig. 4.

In experiments in which the mixture was seeded with small amounts of halocarbon (CF_3Br), the hot region behind flame front was seen in a darkened room as low intensity rose-bright zone, similar in shape to the region surrounded by the isotherm that is marked in Fig. 4 as $T/T_{max}=0.5$.

Records of wall surface temperature made together with Schlieren records of the flame showed that the temperature started to increase just at the moment when the preheat zone of the flame touched the gauge location (Fig. 5). The maximum temperature rises at the wall surface were from 1.3°C to 2.1°C. A long time after passage of the flame the wall temperature remained 0.9°C above its initial temperature.

Instantaneous wall surface temperature was measured by means of gauge shown in Fig. 6. The gauge was constructed in such a manner that the ceramic rod of mullit was thermally insulated and for heat transfer calculations could be considered as one-dimensional rod, 27 mm long, which is thermally attached to a constant temperature wall at its end (see Fig. 6).

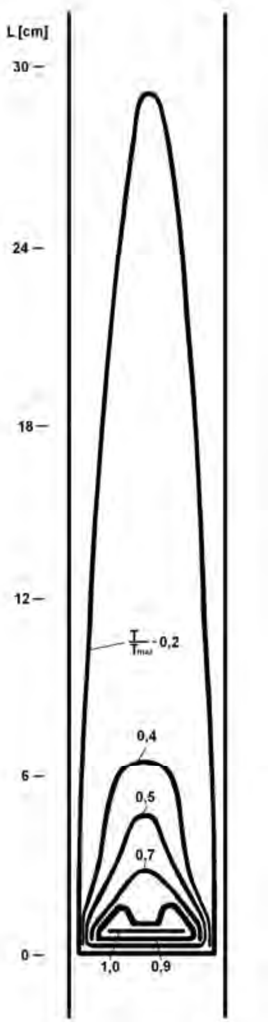

Fig. 4. Flame thermal structure. Isotherms determined based on temperature curves i ndicated in Fig. 3 (with assumption of axial symmetry)

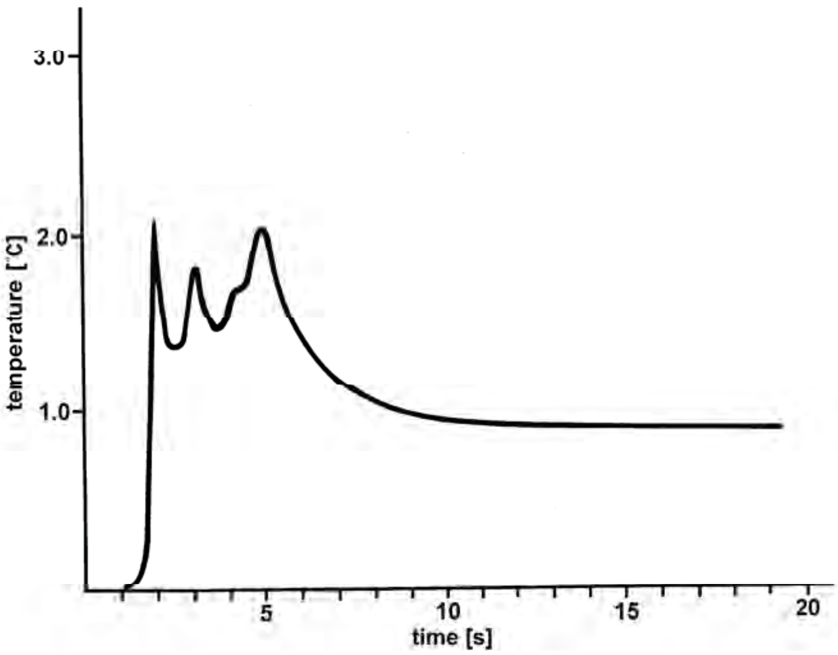

Fig. 5. Example of wall surface temperature record for lean limit methane/air mixture (6%CH4)

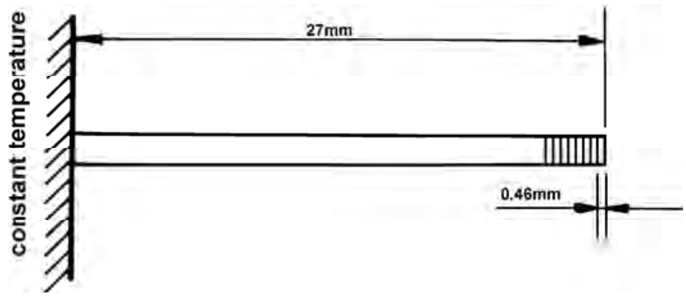

Fig. 6. Mullit rod considered as a one-dimensional system subdivided for numerical solution. Rod is 27 mm long. Calculations were performed for a 46 mm rod. Calculations showed that heat pulse applied at right end did not penetrate 27 mm in 20 second-the test time

4. Calculation for square tube

Heat transfer in a one-dimensional system is described by the equation (1):

$$\lambda \frac{\partial^2 T}{\partial x^2} = c\rho \frac{\partial T}{\partial t}, \tag{1}$$

which, when treated with appropriate boundary and initial conditions, can be solved using a numerical technique in the form of finite difference equation [6]. The numerical procedure becomes particularly simple under conditions when one assumes the distance and time interval are related by the equation (2):

$$\frac{\partial x^2}{\partial t} = 2a, \tag{2}$$

where $a = \frac{\lambda}{c\rho}$ mm2/s.

In this case the expression for the future value of temperature at any location can be calculated from the known temperatures at the previous time step using the relation (3):

$$T_m^{p+1} = \frac{1}{2}\left(T_{m+1}^{p} + T_{m-1}^{p}\right), \tag{3}$$

where *p* is a particular time step and *m* is *x* location along the rod.

Since the value of *a,* for mullite is 1.058 mm2/s, the choice of δt=0.1 s resulted in an incremental length, δx, of 0.46 for mullite. Calculations were made for 200 time steps and 100 length steps. For the calculation the initial conditions were an isothermal rod. Boundary conditions were that the rod was thermally isolated from its surroundings and that the temperature at its front surface was that measured in an experiment. The back boundary condition was an infinite heat sink $T_{100}^{p} = 0.0$. The total time that elapses for one calculation was 20 s and the rod used in this calculation was 46 mm long. The result is shown in Fig. 7.

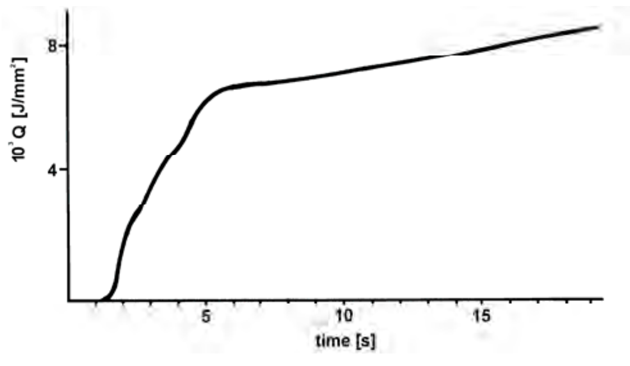

Fig. 7. Calculated heat transfer to the gauge from lean limit downward propagating flame (6%CH4)

The calculated temperature fields inside the rod as a function of time are shown in Fig. 8. As is seen from plots, the increase of temperature is observed no further than 14 mm inside the rod, after 19 s of heat transfer. It can be concluded that the back boundary conditions were such that the rod could be assumed to be of infinite length.

Having the heat of combustion of methane the heat release of the mixture can be determined. Heat of combustion of methane is *Qw*=802320 J/mol. Heat of combustion of methane/air mixture with concentration 6% CH4 is *Qwm*=48139.2 J/mpl. Mole-volume of the mixture (under normal laboratory conditions *T*=293 K, *p*=101325 Pa) is:

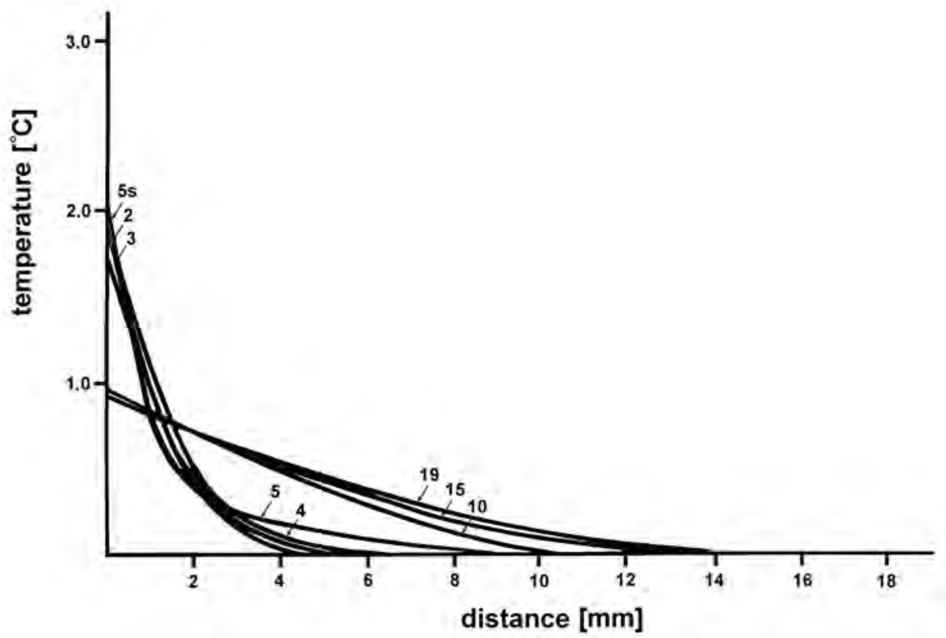

Fig. 8. Calculated temperature profiles inside the rod for lean limit downward propagating flame (6%CH4)

$$\overline{V}_{20} = \frac{\overline{R}}{p}T = \frac{8314.3}{101325}293 = 24.04m^3 = 24040000mm^3. \quad (4)$$

The quantity of heat released per unit volume may be found from:

$$Q = \frac{Q_{wm}}{\overline{V}_{20}} = \frac{48139.2}{24040000}\frac{J}{mm^3}. \quad (5)$$

Each 1mm of height of the tube has a volume $V_1 = 1 \cdot (51)^2 = 2601mm^3$. Thus, the available heat per 1mm of height can be calculated as:

$$q = Q \cdot \overline{V} = \frac{8314.2 \cdot 2601}{24040000} = 5.208\frac{J}{mm}. \quad (6)$$

The calculated heat transfer to the gauge, based on measured temperature curve (see Fig. 5) is 0.0084 $\frac{J}{mm^2}$ (see Fig. 7). Multiplying this value by the surface area of a section of the tube:

$$s = 4 \cdot 51 = 204mm^2 / mm \text{ of the tube height.} \quad (7)$$

The measured heat absorption at the wall can be obtained:

$$q_{calc} = 0.0084 \cdot 204 = 1.714J / mm \text{ of the tube height.} \quad (8)$$

Comparison of available heat per 1mm of the tube height q with the measured heat absorption at the tube wall q_{calc} (for the same 1mm of the tube height) shows that only about 33% of heat is transferred to the walls. The quantity of heat transferred to the mullit gauge is much less than available heat. If the gauge properties are the same as material properties of the walls probably this difference would be smaller.

5. Test stand of model rotating cylindrical combustion chamber

Measurements were carried out in a rotating cylindrical combustion chamber. The chamber was treated as a model of piston engine combustion chamber with swirl. In such chamber flame propagates in a radial direction, outside from its centre, in a shape of cylindrical surface. Flame propagation velocity depends on rotation velocity of a chamber. Temperature changes on a surface of the front walls of the rotating combustion chamber were transmitted towards motionless data acquisition system. The temperature measurements were carried out with the use of special thermocouples located at one of the front wall of the chamber. They measured temperature at the radius r=23 mm and r=28 mm. In these set of experiments methane /air mixture was used with concentrations varied in the range from 6.3% CH4 to 9.5% CH4. Influence of rotation velocity on heat transfer was investigated in the range from 1000 rpm to 5000 rpm. Photo-camera recorded side-view of flame propagation. Such side view of flame propagating in a rotating cylindrical chamber is shown in a form of schematic diagram in Fig. 9. The experiments were conducted in a cylindrical combustion chamber of 90 mm or 140 mm inner diameter and 30 mm height, made of organic glass (see Fig. 9 and 10). The chamber was horizontally mounted on the axis of an electric motor. Rotation rates of up to 6000 rpm were used. The maximum tangential velocity at the edge of the cylinder was 28.3 m/s at 6000 rpm. The central venting orifice at the front vessel wall was open during flow of the mixture. The vessel was filled by displacement. The experimental procedure consisted in supplying the vented vessel for a few minutes with the mixture of the required composition. After shutting off the flow, and closing the venting orifice of the vessel, the tube was rotated at a desired speed for about one minute to establish rigid-body rotation of the mixture. Rigid-body rotation of the gas was confirmed by local velocity measurements. Finally, the mixture was ignited at the centre of the tube by a spark and a process of flame propagation was recorded by a video camera.

Measurement signals were sending to the amplifiers rotating together with the combustion chamber. Amplified signals were transferred to a data acquisition system by means of mercury commutator (Fig. 11).

Schematic view of a location of temperature sensor in the wall is indicated in Fig. 12. Every thermocouple with amplifier was calibrated. During the experiments measurements of temperature as a function of time were carried out. At the same time the side view of the flame was recorded by a video camera.

6. Test results of model rotating cylindrical combustion chamber

Taken by way of example measured wall surface temperature at the radius r=23 mm and r=28 mm are shown as a function of time in Fig. 13. Flame propagating in 8.45%CH4 mixture reached the radius r=23 mm during the time τ=0.028 s and the radius r=28 mm during the time τ=0.029 s at the rotation rate 1000 rpm. At the rotation rate 5000 rpm these radiuses were reached during the time τ=0.039 s and τ=0.056 s, respectively. At the rotation rate 1000 rpm surface temperature rose quickly up to ~13°C during the time ~0.118 s and then slowly decreased. At the rotation rate 5000 rpm the maximum values of temperature are smaller probably as a result close flame extinction. Measured wall surface temperature was used for numerical calculation of heat flux density.

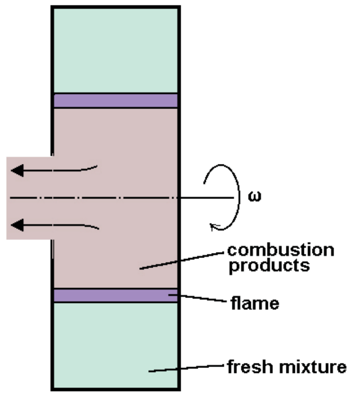

Fig. 9. Side view of flame propagating in a rotating cylindrical combustion chamber shown in a form of schematic diagram

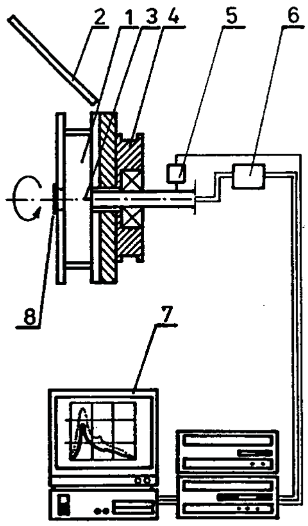

Fig. 10. Experimental set-up: 1-cylindrical chamber (with venting orifice in axis of rotation), 2-mirror, 3-ignitor, 4-V-belt wheel, 5-solenoid valve, 6-mixture supply, 7-PC, 8-tissue paper on the outflow orifice

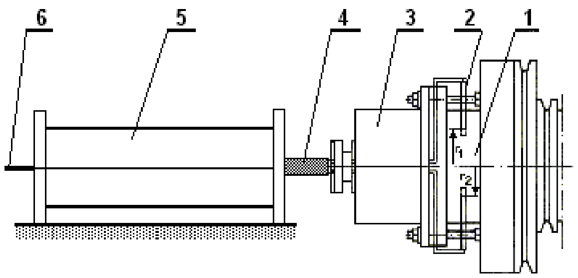

Fig. 11. Schematic diagram of the experimental apparatus for measuring wall temperature: 1-combustion chamber, 2-thermoelectric sensor, 3-system of amplifiers, 4-flexible clutch, 5-mercury commutator, 6-signal wires

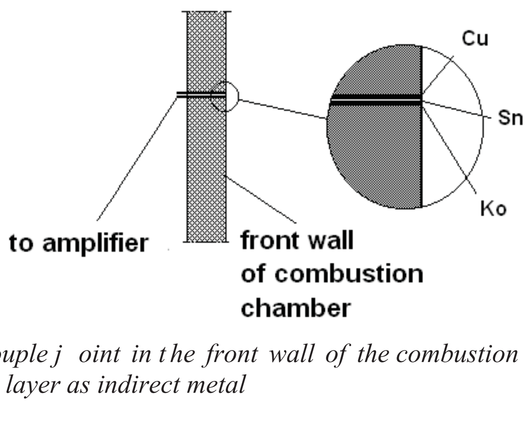

Fig. 12. Schematic view of thermocouple joint in the front wall of the combustion chamber; Cu-copper wire, Ko-constantan wire, Sn-tin thin layer as indirect metal

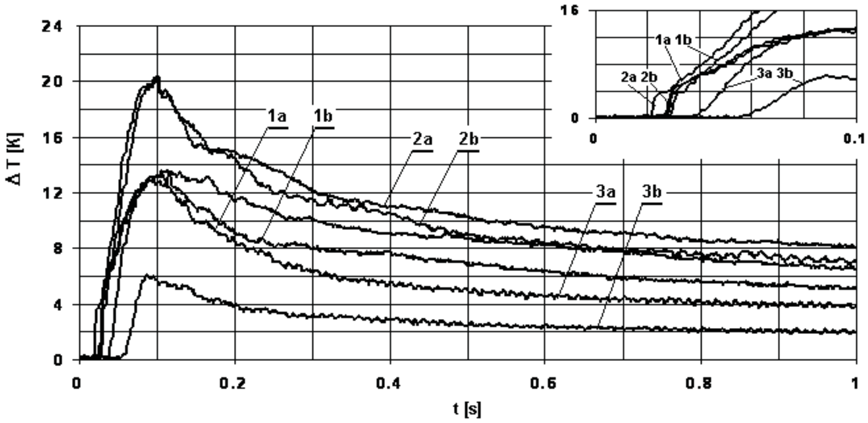

Fig. 13. Measured profiles of temperature changes in the combustion chamber wall surface. Methane/air mixture 8.45%CH4; 1a, 1b - T1, T2, rotational speed 1000rpm, 2a, 2b, - T1, T2, rotational speed 3000rpm, 3a, 3b, - T1, T2, rotational speed 5000rpm (in right top edge of the figure initial part of temperature rise is shown in detail)

Dependence between a surface temperature and a density of heat flux can be obtained by a solution of a problem of one-dimensional heat conduction in a half-infinity body with time-dependant boundary conditions. If surface temperature *T(t)* is a function of time, then heat flux density, *q* at the moment, *t* is expressed by the equation:

$$q(t) = \sqrt{\frac{\lambda \rho c}{\pi}} \int_0^t \frac{1}{\sqrt{t-\tau}} \frac{dT}{d\tau} d\tau \quad \left[\frac{W}{m^2}\right], \tag{9}$$

where particular symbols related to the wall material are:

λ - thermal conductivity,
ρ - density,
c - specific heat.

Equation (9) can be solved when function *T(t)* is known. In the experiments the data acquisition system is used to register given quantities in a form of sequence of discrete values. If derivative $dT/d\tau$ is replaced in a given step „*i*" by its approximate value $\Delta Ti/\Delta \tau i$, heat flux density is as follow:

$$q(t_n) = \sqrt{\frac{\lambda \rho c}{\pi}} \sum_{i=1}^{n} \frac{\Delta T_i}{\Delta \tau_i} \int_{t_{i-1}}^{t_i} \frac{1}{\sqrt{t_n - \tau}} d\tau. \tag{10}$$

In calculations $\Delta\tau = const$, and because of this at every moment *i*: $ti = i\Delta\tau$. Under such assumptions, integral in equation (10) can easily be solved and finally heat flux density is expressed by equation (11):

$$q(t_n) = 2\sqrt{\frac{\lambda \rho c}{\pi \Delta \tau}} \sum_{i=1}^{n} \left(T_i - T_{i-1}\right)\left(\sqrt{n-i-1} - \sqrt{n-i}\right), \tag{11}$$

where *i, n* – numbers of time steps.

High accuracy of calculation might be obtained when the time steps *i* and *n* are sufficiently small.

The exemplary results of calculations are shown in Fig. 14.

4. Conclusions

- At each location the heat flux and heat transfer coefficient rises rapidly as the flame arrives that location. In the combustion chamber at each location where the time of flame arrival coincides with peak cylinder pressure, heat flux has maximum value. Flame velocity is reduced wherever the surface temperature is low; so heat flux at the vicinity of intake valve rises after the other regions. Average surface heat flux on the cylinder wall is the lowest.

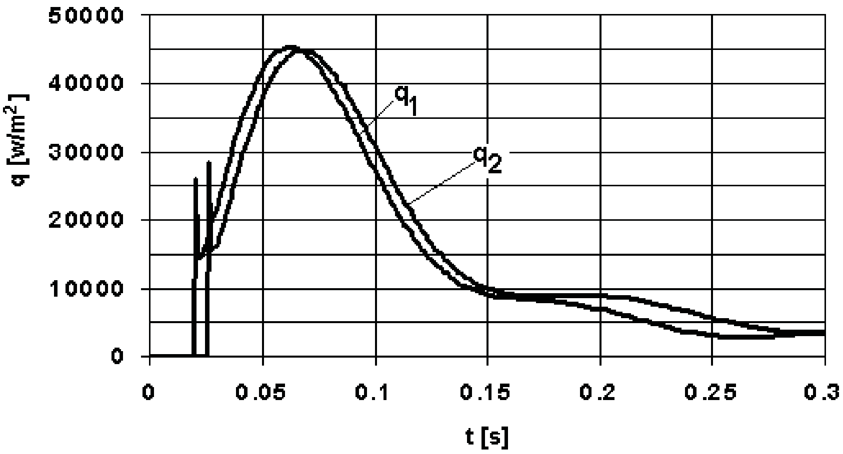

Fig. 14. Calculated heat flux density at the wall for methane/air mixture 8.45%CH4 and rotational speed 3000 rpm: Heat q_1 is for a radius r_1=23 mm and q_2 for r2=28 mm

- The limit flames are very thick and that heat losses to the walls caused extensive cooling of the product gases within 0.05m behind the propagating flame. Numerical calculations made possible to understand in detail physical picture of heat transfer from the hot gases to the cold walls. It is a unique way for explanation of heat transfer phenomenon.
- Flame propagation velocity depends on rotation velocity of a chamber and very high rotation velocities generated flame extinction.
- For a constant mixture concentration maximum temperature rise depended on the rotation velocity. Increased rotation velocity initially intensified heat transfer to the wall, with following to it increase of peak temperature, but further rise of rotation velocity resulted in local flame extinction near the side walls, decreased volume of combustion gases, and as a result of these, decreased value of peak temperature at the wall surface. The research results could be used to draw up chamber heat balance and are useful for internal combustion engine chamber design.
- Presented techniques make possible to estimate heat losses to the walls in a combustion chamber. Knowledge of heat flux density together with contact area of flame and hot combustion gases with the wall as a function of time.

References

[1] Aoki, Y., Emi, M., Kimura, S., Shimano, K., Enomoto, Y., *An Experimental Study on Heat Transfer Coefficient of All Combustion Chamber Wall Surfaces in A Naturally Aspirated D.I. Diesel Engine,* FISITA Proc. F2010-A-151, 2010.

[2] Demuynck, J., De Paepe, M., Sierens, R., Verhelst, S., *Heat Transfer Measurements Inside a Gas Fuelled Spark Ignited Engine for Model Validation*, FISITA Proc. F2010-A-062, 2010.

[3] Finol, C. A., Robinson, K., *Thermal Modelling of Modern Engines: A Review of E mpirical Correlations to Estimate the In-Cylinder Hea t Transfer Coefficient,* Proceedings of the Institution of Mechanical Engineers Part D-Journal of Automobile Engineering, 220 (D12), pp. 1765-1781, 2006.

[4] Han, Z., Reitz, R. D., *A Temperature Wall Function Fo rmulation for Variable-Density Turbulent Flows with Application to Engine Convective Heat Transfer Modelling*, Int. J. Heat Mass Transfer, Vol. 40, No. 3, pp. 613-625, 1997.

[5] Heywood, J. B., *Internal Combustion Engine Fundamentals* , McGraw-Hill Book Company, 1987.

[6] Jankowski, A., Jarosinski, J. Slezak, M., *Evaluation of Heat Transfer from Combustion Gases to Combustion Chamber Walls of Piston Engines*, Proc. EAEC2009 01-049, 2009.

[7] Peters, N., *Turbulent Combustion*, Cambridge University Press, 2004.
[8] Sugihara, T., Shimano, K., Enomoto, Y., Suzuki, Y., Emi, M., *Direct Heat Loss to Combustion Chamber Walls in a DI Diesel Engine, Development of Measurement Technique and Evaluation of Direct Heat Loss to Cylinder Liner Wall*, SAE Technical Papers 2007-24-0006, 2007.
[9] Suzuki, Y., Shimano, K., Enomoto, Y., Emi, M., Yamada, Y., *Direct Heat Loss to Combustion Chamber Walls in A Direct-Injection Diesel Engine, Evaluation of Direct Heat Loss to Piston and Cylinder Head*, International Journal of Engine Research, Vol. 6, pp. 119-135, 2005.
[10] Woschni, G., *A Universally Applicable Equation for the Instantaneous Heat Transfer Coefficient in the Internal Combustion Engine,* SAE Paper No. 670931, 1967.