Analysis of Heat Transfer in the Combustion Chamber of an Internal Combustion Engine using Thermal Networks

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Abstract: - The heat transfer processes in an internal combustion engine can be modeled with a variety of methods. These methods range from simple thermal networks to multidimensional differential equation modeling. Thermal network models, using resistors and capacitors, are very useful for rapid and efficient estimation of conduction, convection and radiation heat transfer processes in engines. Using a thermal network, the significant resistances to heat flow, and the effects of changing material thermal conductivity, thickness, and coolant properties can be easily determined.

Key-Words: - cylinder wall, piston, cylinder head, heat transfer, thermal stresses

1 Introduction

Thermal stresses analysis in combustion chamber of spark ignition engine is an important field which automobile engineers study in order to increase life of mechanical parts.

Some of studies have been presented about thermal stresses and analysis for various engines. Local convective heat transfer in a four-stroke single cylinder engine using a computational fluid dynamics code was analyzed by Mohammadi and Yaghoubi [11]. The results were compared with experimental measurement in the literature. Besides, new correlations were proposed to predict maximum and minimum convective heat transfer coefficient in a SI engine's combustion chamber. Lee and Assanis [8] analyzed thermo-mechanical of optically accessible quartz cylinder under fired engine operation. Using FEM analysis, temperature and stress distributions were estimated. Three types of outside cooling for reducing thermal stress level (natural, moderate forced and intensive forced convection), were considered. Thermal analysis of cylinder head carbon deposits from single cylinder diesel engine was researched by Husnawan et al. [7].

The thermal stresses in the cylinder head and in the piston head of an internal combustion engine are determined as a function of the level of temperatures, the temperature difference between different parts and the heat transfer between parts. These further depend on the speed and the way the heat is transferred to the parts, on the part shape, their thermal conductivity and cooling.

The temperature of the parts has an effect on operating temperature of the lubricating oil and hence, on its viscosity, oil – film thickness which separates of the rubbing pair and the nature of friction [4].

The latter together with wear characteristics of materials, which also depend on the temperature of parts, determine the wear rate.

Temperature stresses appear because of uneven distribution of temperature in the parts and also because the majority of parts do not enable the most heated portions to expand freely.

By the thermal load it is meant the value of specific heat flux transferred from the working fluid to the surface of a part. Transfer of heat from the working fluid to the surface of parts is affected in two ways: by convection and by radiation. Convection has a major importance for engines because combustion is accompanied by formation of soot which burns out subsequently.

The soot content in the flame is the cause of its degree of blackness, and therefore, of high emissive power of flame. High flame – temperatures and degrees of blackness of flame are the cause of high fraction of heat transferred by radiation.

The thermal stress level of separate portions of parts depends mainly on the disposition of the portion relative to the flame and is therefore not the same.

In the piston – bowl combustion chamber engines, some zones of the parts like cylinder head, cylinder liner and piston head are shielded by the piston body from the flame in the period of the most intense radiation [6].

It follows that the thermal stress level depends on the distribution of temperature in the parts. It is a function of the heat load, design of the parts and cooling conditions. The distribution of local resistances depends on the design of parts. The cylinder head and the piston head are the most thermally stressed parts. The thermal state of the cylinder liner is also of importance, because it has an appreciable effect on the thermal state of the piston.

From engines operating experience the maximum permissible temperatures of different parts are known. They should not exceed 350^0C. The minimum temperature (about 160 - 180^0C) of the cylinder liner is restricted because of its effect on the water vapour condensation conditions, depending also of the materials of the parts.

The given values may also be regarded as tentative, because the design and definite working conditions of a part affect the limiting possible temperature.

Measuring the temperatures in different parts of the cylinder head and the piston head, the cooling can be adjusted or the materials can be improved, or even the properties of the fuels can be improved. We can take into consideration the mixture of a small quantity of alcohol with gasoline in order to reduce the level of temperature [15].

2 Thermal Network Models

Thermal network models, using resistors and capacitors, are very useful for rapid and efficient estimation of conduction, convection and radiation heat transfer processes in engines [4].

Using a thermal network, the significant resistances to heat flow, and the effects of changing material thermal conductivity, thickness, and coolant properties can be easily determined. A simple four node series network, which includes convection and conductivity is shown in figure 1.

This is an illustration of steady state heat transfer from engine cylinder gas to the coolant. This series path is composed of convection through the coolant liquid boundary layer.

The cylinder gas boundary layer insulates the cylinder wall from high temperature cylinder gases.

Thermal networks are primarily used for convection and conduction heat transfer, as the radiation heat transfer equation needs to be linearized to conform to resistance model [3].

Using Fourier's equation, the conduction resistance is:

$$R_{cond} = \frac{\Delta T}{Q/A} = \frac{L}{\lambda} \tag{1}$$

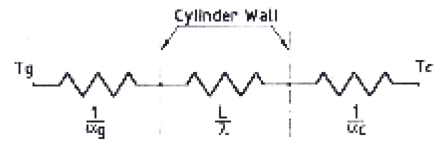

Fig. 1

And using Newton's equation, the convection resistance is:

$$R_{conv} = \frac{\Delta T}{Q/A} = \frac{1}{\alpha} \tag{2}$$

Where:

Q [W] - heat flow taken by the piston

λ [W/m K] - working fluid thermal conductivity

α_g [W/m^2 K] - instantaneous heat transfer coefficient

L [m] - a length scale, such as the cylinder bore

ΔT – difference of temperature

3 Cylinder wall model

We now consider the unsteady nature of heat flux from the combustion gas to the cylinder wall. The cylinder wall has a periodic heat flux on the gas side and a constant surface temperature on the coolant side [4], [9].

The problem posed requires solution of the heat conduction equation:

$$\frac{\partial T}{\partial t} = a \frac{\partial^2 T}{\partial x^2} \tag{3}$$

where: a [m^2/s] is the thermal diffusivity of the wall material, $a = \lambda/\rho\, c$

Subject to the following boundary conditions:

$$-\lambda \frac{\partial T}{\partial x} = q_0^{"} + q_1^{"} \sin(\omega t) \quad \text{at } x = 0$$

$$T = T_L \quad \text{at } x = L \tag{4}$$

As well as initial condition:

$$T = T_i(x) \quad \text{at } t = 0 \tag{5}$$

An exact solution can be written in closed form but it is quite cumbersome and as a result, no more illustrative than a computer solution.

Fortunately, an approximate solution can be derived for the practical case where:

$$\omega \cdot t >> 1 \quad \text{and} \quad \frac{\omega \cdot L^2}{2 \cdot a} >> 1 \tag{6}$$

In this case the temperature field is given by:

$$T = T_L + \frac{q_0^{"}}{\lambda}(L - x) + \\ + \frac{q_1^{"}}{\left(\frac{a}{\omega}\right)^{\frac{1}{2}}} \exp\left[\left(-\frac{\omega}{2a}\right)^{1/2} \cdot x\right] \sin\left[\omega t - \left(\frac{\omega}{2a}\right)^{1/2} x - \frac{\pi}{4}\right] \tag{7}$$

The inspection of this solution shows that:
The surface temperature at x = 0 oscillates with the same frequency as the imposed heat flux but with a phase difference of $\pi/4$.
The amplitude of the oscillations decays exponentially with the distance x from the surface.
The amplitude is reduced to 10% of that at the surface at the distance given by:

$$\delta = -\ln(0{,}10)\left(\frac{2a}{\omega}\right)^{1/2} = 2{,}3\left(\frac{2a}{\omega}\right)^{1/2} \tag{8}$$

The penetration distance δ is a measure of how far into the material fluctuations about the mean heat flux penetrate. For distances x greater than δ, the temperature distribution is more or less steady and driven only by the time average heat flux. Since the length δ is rather compared to the dimensions (wall thickness, bore, etc) over which conduction heat transfer occurs, two simplifications can be made:
- Conduction heat transfer in the various parts can be assumed steady and driven by the average flux;
- Heat transfer from the gas can be coupled to the conduction analysis accounting for capacitance only in a penetration layer of thickness δ in series with a resistance computed or measured for steady state.

A five mode thermal network for a cylinder wall is given in Fig. 2.
The modeling of the penetration layer can be complicated by the presence of an oil film or deposits.
Fortunately an accurate model is not required as the fluctuations about the mean $\overline{T}_\delta$ tend to be small compared to the gas – penetration depth temperature difference $T_g - \overline{T}_\delta$.For an engine operated at a steady state, the penetration layer is thin because the engine frequency, which dictates the frequency components of the heat flux imposed on the gas – solid interfaces, is rather high. On the other hand, in the case of an engine being accelerated or decelerated, the penetration layer is thicker because lower frequency components of the heat flux are characteristic of the rates of change of engine speed.

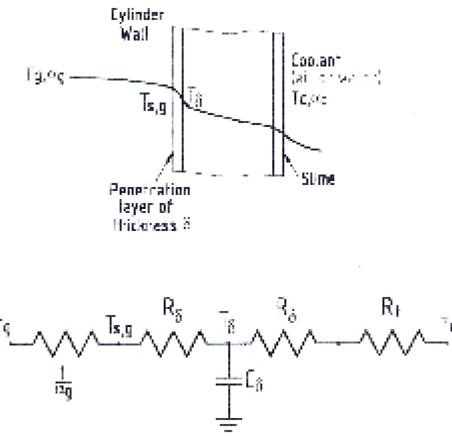

Fig. 2

Where:
R_t is the conduction path resistance to coolant
$R_\delta = \delta/2\lambda$ and $C_\delta = \rho c \lambda$ [2], [4].

3.1 Field of temperature in the cylinder head

The differences of the temperature between the parts of the cylinder head are important due to its shape, design and cooling. [1], [5]. According to figure 2 we can assume:

- Heat transfer flow in the cylinder head wall:

$$q = \frac{\overline{T}_g - T_c}{\frac{1}{\overline{\alpha}_{gw}} + \frac{\delta_1}{\lambda_1} + \frac{\delta_2}{\lambda_2} + \frac{1}{\overline{\alpha}_c}} \quad [\mathrm{W/m^2}] \tag{9}$$

- Inner temperature of cylinder head wall with carbon soot layer:

$$T_{w1} = \overline{T}_g - \frac{q}{\overline{\alpha}_{gw}} \quad [\mathrm{K}] \tag{10}$$

- Inner temperature of cylinder head wall without carbon soot:

$$T_{w2} = T_{w1} - q\frac{\delta_1}{\lambda_1} \quad [\mathrm{K}] \tag{11}$$

- Cylinder head wall to cooling fluid:

$$T_{w3} = T_{w2} - q\frac{\delta_2}{\lambda_2} \quad [\mathrm{K}] \tag{12}$$

Where:
T_g [K]- mean temperature of the gases
T_c [K]- cooling fluid temperature

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$\overline{\alpha}_{gw}$ [W/m^{2}K] - heat transfer coefficient gas- wall
λ_1 [W/mK] - thermal conductivity of carbon soot
δ_1 [m]- thickness of carbon soot layer
λ_2 [W/mK] - thermal conductivity of cylinder head material
δ_2 [m] - thickness of cylinder head wall
$\overline{\alpha}_c$ [W/m^2 K] - heat transfer coefficient wall – cooling fluid

4 Piston model

Determination of the temperature profile of an engine component such as a piston requires solution of the three – dimensional heat conduction equation. As an illustration, a case study of heat transfer in a piston will be presented. Figure 3 shows how a piston can be divided into a number of elements for analysis. Only one quadrant of the piston in the x – y plane needs to be considered as the piston is symmetrical [12].
The piston can be treated as steady and driven by an average heat flux since the penetration layers are small.

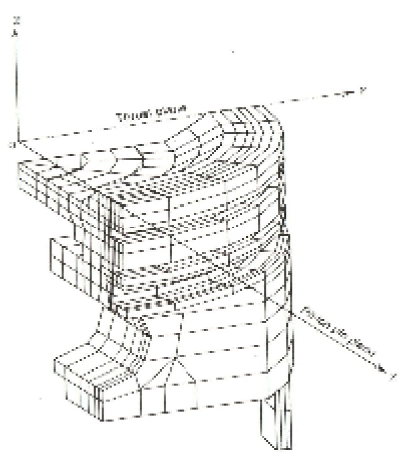

Fig. 3 Piston mesh [4]

The mean cylinder gas temperature is computing using a cycle simulation to predict instantaneous gas temperature and then integrated according to:

$$\overline{T}_g = \frac{1}{4\cdot\pi\cdot\overline{\alpha}_g}\int_0^{4\pi}\alpha_g T_g \, d\theta \tag{13}$$

$$\overline{\alpha}_g = \frac{1}{4\cdot\pi}\int_0^{4\pi}\alpha_g \, d\theta \tag{14}$$

Where:
$\overline{\alpha}_g$ [W/m K] - average heat transfer coefficient

The average heat transfer coefficient is used in estimating the heat transfer coefficients on the crown of the piston in contact with the cylinder gas [5].

4.1 Field of temperature in the piston

The level of the temperatures in the piston determines the regular operation of the whole engine [10].
It is the most important to determine the field of temperature in the piston head, in the center and at the edge.
If we consider the piston head a symmetrical – axial body, the equation of conduction is:

$$\frac{\partial^2 T}{\partial r^2} + \frac{1}{r}\frac{\partial T}{\partial r} + \frac{\partial^2 T}{\partial z^2} = 0 \tag{15}$$

Or using undimensional coordinates:
$\eta = r/R$
$\xi = z/\delta$ and
$\Theta = \overline{T}_g - T$ the equation is written [5]:

$$\frac{\partial^2 \Theta}{\theta \eta^2} + \frac{1}{\eta}\frac{\partial \Theta}{\partial \eta} + \frac{R^2}{\delta^2}\frac{\partial^2 \Theta}{\partial^2 \xi^2} = 0 \tag{16}$$

The solution is given by integration:

$$T = \overline{T}_g - \Theta = \overline{T}_g - \\ - \frac{\phi}{10^{-6}\pi\frac{D^2}{4}}\left(\frac{1}{\overline{\alpha}_g} + \frac{10^{-3}\delta}{\lambda_p}\xi + \frac{10^{-3}D}{\lambda_p}\eta\right)[K] \tag{17}$$

In the center of the piston head the temperature is:

$$T_{cp} = \overline{T}_g - \frac{\phi}{A}\left(\frac{1}{\overline{\alpha}} + \frac{D}{\lambda_p}\cdot\varphi_1\right)[K] \tag{18}$$

At the edge of the piston head the temperature is:

$$T_{ep} = \overline{T}_g - \frac{\phi}{A}\left(\frac{1}{\overline{\alpha}} + \frac{h}{\lambda_p} + \frac{D}{\lambda_p}\varphi_2\right)[K] \tag{19}$$

Where:
D [m]- piston bore
h [m]– piston head height
L [m] - a length scale, such as the cylinder bore
α_g [W/ m K] - instantaneous heat transfer coefficient
$\overline{\alpha}_g$ [W/ m^2 K] - average heat transfer coefficient
λ_p [W/.m K] - piston thermal conductivity
$\varphi_1 = \varphi_1$ (D,λ), $\varphi_2 = \varphi_2$ (D,λ) – coefficients as a function of bore and thermal conductivity [15]
ϕ [W] - heat flow taken by the piston

5 Case Study

We can consider a case of spark ignition engine with the effective power P_e = 65 kW, the bore D = 92 mm, capacity V_t = 2500 cm^3, and speed n = 4500 min^{-1} [15].

T_g= 1365 K

T_c = 363 K

$\overline{\alpha}_{gw}$ =250 W/m^2K

λ_1 = 0,232 W/mK

δ_1 = 0,02 mm

λ_2 = 160 W/m K

δ_2 = 8 mm

$\overline{\alpha}_c$ = 3500 W/m^2 K

We can also consider the heat flow in the piston is: $\phi = 0.1\ P_e$. and φ_1 = 1,02; $\varphi_2 = \varphi_2\ (D,\lambda)$ φ_2 = 1,28.

Using the equations (9) and (12) we obtain

$q = 2.26 \cdot 10^5 \quad W/m^2$

T_{w1} = 461 K

T_{w2} = 441 K

T_{w3} = 429 K

Using the equations (18) and (19) we obtain the temperatures: T_{cp} = 523 K - piston head center temperature and T_{ep}= 474 K - piston head edge temperature [15].

The measured and theoretic temperatures are inputs to the ANSYS program to determine the temperature field, thermal stresses and deformations due to the temperature difference in the piston head.

The more recent computation finite element programs have implemented thermal finite elements too. For the purpose of this paper, use was made of the ANSYS program which contains 20 types of elements for the heat transfer out of which the types of "thermal elements" were used: for preset nodal temperatures, axial- symmetric solid, thin plate, three- dimensional solid [13].

Using these elements the piston and cylinder head were investigated in terms of thermal steady conditions and the temperature field, heat flow thermal stresses and displacements along different directions were obtained.

In the thermal approach, the rigidity matrix becomes conductivity matrix, the nodal displacement vector becomes the nodal temperature vector and the tensions become heat flows [14].

The input data in the program are:

- longitudinal elasticity module, E = 0.75 10^{11} [N/m^2]
- thermal conductivity, λ_p = 160 [W/m K]
- Poisson's number, μ = 0.3
- linear thermal expansion coefficient, α =20 10^{-6} [K^{-1}]

It is considered a section through piston head and cylinder head and the temperature is measured in steady conditions are input data.

The engine was instaled on a testing stand. Chromel -alumel thermocouples were provided to measure the outlet gas temperature, the cylinder - head wall temperature near the inlet and outlet ports and the piston head temperature.

The dots in figure 4 and 5 indicate the places were thermocouples are put.

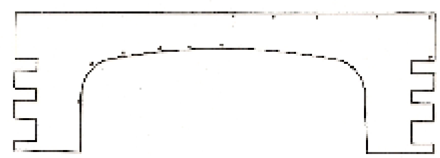

Fig. 4

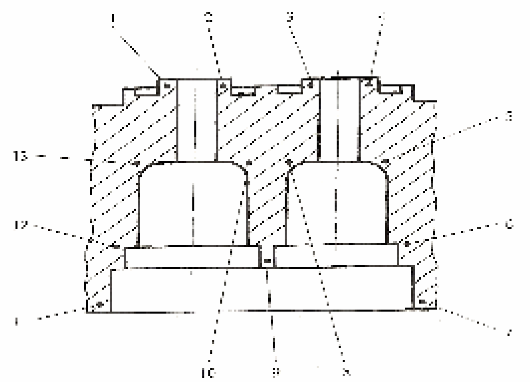

Fig.5

The measured temperatures are inputs to the ANSYS program to determine the temperature field, thermal stresses and deformations due to the temperature difference in the cylinder head wall [5]

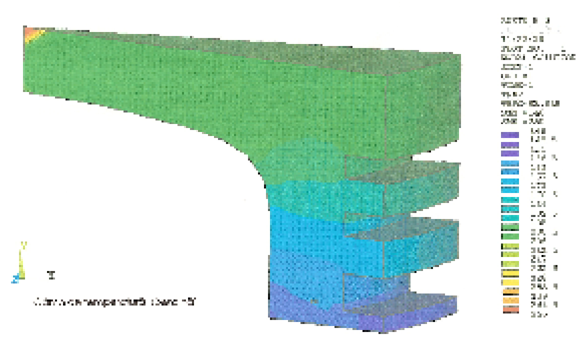

Fig. 6 – Field of temperature in the piston

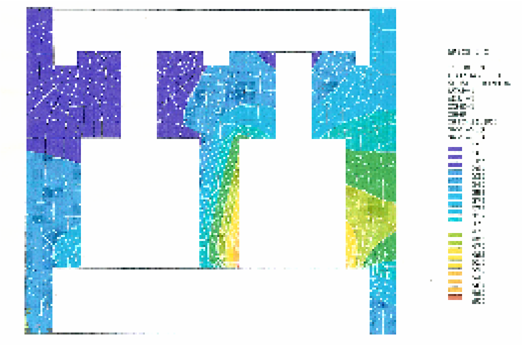

Fig. 7 – Field of temperature at the bridge area of the cylinder head

References:

[1] Asad A Salem, Heat Transfer in an axisymmetric stagnation flow on an infinite circular cylinder - *5th WSEAS Int. Conf. on Heat and Mass transfer (HMT'0)*, Acapulco, Mexico, January 25-27, 2008.

[2] Bhatti, S., Kumari, S., Chaitanya, V., Steady state stress analysis and heat transfer analysis on an axial flow gas turbine blades and disk, *Proceedings of the IASME/WSEAS International Conference on Energy & Environmental Systems*, Chalkida, Greece, May 8-10, 2006, pp. 110-116.

[3] Esfahanian,V., Thermal analysis of an SI engine piston using different combustion boundary condition treatments, *Applied Thermal Engineering* 26, 2006, pp. 277–287.

[4] Ferguson, R.C., Kirkpatrick, T.A. *Internal Combustion Engines*, John Wiley &Sons, New York, 2001.

[5] Han, S., Chung, Y., Empirical Formula for Instantaneous Heat Transfer Coefficient in Spark Ignition Engine. *SAE* paper 972995, 1997.

[6] Heywood, J., *Internal Combustion Engines Fundamentals*, McGraw – Hill, New York, 1988.

[7] Husnawan M., Masjuki H.H., Mahlia, T.M.I. Saifullah M.G., Thermal analysis of cylinder head carbon deposits from single cylinder diesel engine fueled by palm oil–diesel fuel emulsions, *Applied Energy* 86, 2009, pp. 2107–2113.

[8] Lee K.S., Assanis N., Thermo-mechanical analysis of optically accessible quartz cylinder under fired engine operation, *International Journal of Automotive Technology*, 1 (2), 2000, pp. 79−87.

[9] Li, C., Piston Thermal Deformation and Friction Considerations, *SAE* paper 820086, 1982.

[10] Mahdi, H., Rashidi, M., Determination of Piston and Cylinder Head Temperature Distribution in a 4-Cylinder Gasoline Engine at Actual Process, *Proceedings of the 4th WSEAS Int. Conf. on HEAT TRANSFER, THERMAL ENGINEERING and ENVIRONMENT*, Elounda, Greece, August 21-23, 2006, pp. 153-158.

[11] Mohammadi, A., Yaghoubi, M., Rashidi, M., Analysis of local convective heat transfer in a spark ignition engine, *International Communications in Heat and Mass Transfer*, 35, 2008, pp.215–224.

[12] Riyad, M., Abdelkader, H., Analytical solution of one-dimensional transient heat conduction problem in thick metal slab, *Proceedings of the 3rd IASME/WSEAS Int. Conf. on HEAT TRANSFER, THERMAL ENGINEERING AND ENVIRONMENT*, Corfu, Greece, August 20-22, 2005, pp. 172-177

[13] Riyad, M., Abdelkader H., Comparison between the analytical and numerical methods of solving one-dimensional transient heat conduction problems, *Proceedings of the 4th WSEAS Int. Conf. on HEAT TRANSFER, THERMAL ENGINEERING and ENVIRONMENT*, Elounda, Greece, August 21-23, 2006, pp. 380-384.

[14] Shojaefard, M.H., Ghaffarpour M.R., Noorpoor A.R., Alizadehnia S., Thermomechanical Analysis of an Engine Cylinder Head, *Journal of Automobile Engineering* 220 (5) 2006, pp. 627-636.

[15] Uzuneanu, K., Scarpete, D., Panait, T. - Study on Thermal Stress Occuring in the Burning Chamber of an Ethanol-Gasoline Fueled Spark Ignition Engine, *Conference "MOTAUTO' 07"* , Varna, Bulgaria; Proceedings ISBN 954-90272-6-0, Vol.I, 2007, pp. 99 - 103.