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THERMAL STRESSES IN THE COMBUSTION CHAMBER OF A SPARK IGNITION ENGINE FUELED WITH ETHANOL-GASOLINE BLENDS

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Abstract: *The thermal stresses are due to the difference in the parts temperatures. 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. If an ethanol - gasoline mixture is used to fuel an unmodified spark ignition engine a lower level of the part temperatures is obtained as compared with the case when only gasoline is used.*

Key words: *heat transfer/ thermal network model/ piston head/ field of temperature/ ethanol - gasoline blend.*

1. Introduction

The thermal state of the parts of an internal combustion engine has an effect on the strength characteristics of the material of which the parts are made, on the rate at which deposits appear on the parts, on the lubrication conditions of the parts, on friction, wear and stresses in the parts. 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.

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

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the flame in the period of the most intense radiation.

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 [1].

2. 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].

The problem posed requires solution of the heat conduction equation:

$$\frac{\partial T}{\partial t} = a\,\frac{\partial^2 T}{\partial x^2} \qquad (1)$$

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

Subject to the following boundary conditions:

$$-\lambda \frac{\partial T}{\partial x} = q_0'' + q_1'' \sin(\omega t) \quad \text{at } x = 0$$

$$T = T_L \quad \text{at} \quad x = L \qquad (2)$$

As well as initial condition:

$$T = T_i(x) \text{ at } t = 0 \qquad (3)$$

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:

$$\omega \cdot t >> 1 \text{ and } \frac{\omega \cdot L^2}{2 \cdot a} >> 1 \qquad (4)$$

In this case the temperature field is given by:

$$T = T_L + \frac{q_0''}{\lambda}(L-x) + \frac{q_1''}{\left(\frac{a}{\omega}\right)^{\frac{1}{2}}} \exp\left[\left(-\frac{\omega}{2a}\right)^{1/2} \cdot x\right] \sin\left[\omega t - \left(\frac{\omega}{2a}\right)^{1/2} \cdot x - \frac{\pi}{4}\right] \qquad (5)$$

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:

$$\delta = -\ln(0{,}10)\left(\frac{2a}{\omega}\right)^{1/2} = 2{,}3\left(\frac{2a}{\omega}\right)^{1/2} \qquad (6)$$

Example: For a two-stroke engine operating at 2000 rpm ($\omega = 209\ s^{-1}$) and made of cast iron ($a = 21 * 10^{-6}\ m^2/s$) this length is rather small, $\delta = 1$ mm. [2]

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.

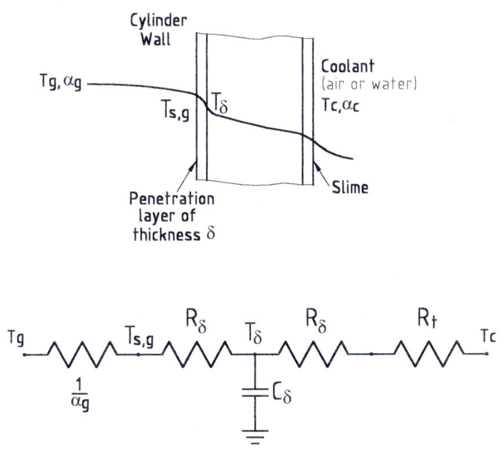

Fig. 1

A five mode thermal network for a cylinder wall is given in Fig. 1. 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.

3. Piston head model

The level of the temperatures in the piston determines the regular operation of the whole engine.

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:

$$\frac{\partial^2 T}{\partial r^2} + \frac{1}{r}\frac{\partial T}{\partial r} + \frac{\partial^2 T}{\partial z^2} = 0 \tag{7}$$

Or using undimensional coordinates: η = r/R si ξ = z/δ and Θ = $\overline{T}_g - T$ the equation is written [5]:

$$\frac{\partial^2 \Theta}{\theta \eta^2} + \frac{1}{\eta}\frac{\partial \Theta}{\partial \eta} + \frac{R^2}{\delta^2}\frac{\partial^2 \Theta}{\partial^2 \xi^2} = 0 \tag{8}$$

The solution is given by integration:

$$T = \overline{T}_g - \Theta = \overline{T}_g - \frac{\phi}{10^{-6}\pi \frac{D^2}{4}}\left(\frac{1}{\overline{\alpha}_g} + \frac{10^{-3}\delta}{\lambda_p}\xi + \frac{10^{-3}D}{\lambda_p}\eta\right)[K] \tag{9}$$

In the center of the piston head the temperature is:

$$T_{cp} = \overline{T}_g - \frac{\phi}{A}\left(\frac{1}{\overline{\alpha}} + \frac{D}{\lambda_p}\cdot\varphi_1\right) \qquad (10)$$

At the edge of the piston head the temperature is:

$$T_{ep} = \overline{T}_g - \frac{\phi}{A}\left(\frac{1}{\overline{\alpha}} + \frac{h}{\lambda_p} + \frac{D}{\lambda_p}\varphi_2\right) \qquad (11)$$

4. Experimental study

The experiments conducted in the internal combustion engines laboratory of University "Dunarea de Jos" of Galati have mainly used a mixture of gasoline and ethanol to feed an unmodified spark ignition engine.

When blended with gasoline, the ethanol tendency to separate is much lower than the methanol. The high volatility of the gasoline alcohol blends is a significant inconvenient because they cause fuel losses by evaporation.

Thus, the temperature of the fuel constant level chamber reaches 50 to 55 degrees during operation and in summer even 80-85 degrees. The losses by evaporation of an engine supplied with 10% alcohol and gasoline increases by 90%.

The gasoline - alcohol blends behave normally relative to the abnormal detonation combustion also revealed by the increase in their self-ignition temperature. The generally high anti-detonation qualities of the gasoline-alcohol or gasoline ether blends can be turned into a good account by optimizing the fuel or the engine [8].

As far as the fuel is concerned, the basic gasoline composition (and therefore the fabrication technology) can be maintained. This is extremely important for prevention of the polluting exhaust gases.

Using pure ethanol to fuel engines is limited by the need to make significant changes to the fuel supply systems. Since the ethanol caloric power is twice lower than that of the gasoline, a double quantity of alcohol is necessary to reach the same output.

Pure ethanol supplying calls for specially designed and manufactured engines.

We can consider a case of spark ignition engine supplied by gasoline and by a blend of 10% ethanol and 90% gasoline (E 10). [2] The effective power of the engine is P_e = 65 kW, the bore is D = 92 mm, $\overline{\alpha}_g$ = 350 W/m^2 K is an average heat transfer coefficient, λ_p = 160 W/m K is piston thermal conductivity and φ_1 = 1,02; φ_2 = φ_2 (D,λ) φ_2 = 1,28.

We can also consider the heat flow in the piston is: $\phi = 0.1\ P_e$.

- Molecular mass of fuel

μ = z 46 + (1-z) 110 [kg/kmol]

gasoline μ = 110 [kg/kmol]

ethanol μ = 46 [kg/kmol]

- Gravimetric composition

$$c = \frac{z\cdot 24 + (1-z)\cdot 0{,}855\cdot 110}{z\cdot 46 + (1-z)\cdot 110} \ \text{[kgC/kgfuel]}$$

$$h = \frac{z\cdot 6 + (1-z)\cdot 0{,}145\cdot 110}{z\cdot 46 + (1-z)\cdot 110} \ \text{[kgH/kgfuel]}$$

$$o = \frac{z\cdot 16}{z\cdot 46 + (1-z)\cdot 110} \ \text{[kgO/kgfuel]} \qquad (13)$$

It results:

gasoline c = 0,855 h = 0,145

E10 c = 0,84 h = 0,144 o = 0,015

- Polytropic compressing exponent
  $n_g = 1{,}36$ Fig. 3
  $n_{Eg} = 1{,}25$ [8]

- Parameters at the end of combustion
gasoline $p_z = 62\ 10^5$ [N/m^2] $T_z = 2805$ [K]
E10 $p_z = 60\ 10^5$ [N/m^2] $T_z = 2705$ [K]
- Fuel consumption
  gasoline $c_e = 0.388$ [kg/kWh]
  E10 $c_e = 0.413$ [kg/kWh]
- Mean temperature of the burnt gases
  gasoline $T_g = 1365$ [K]
  E10 $T_g = 1305$ [K]
- Thermal efficiency
  gasoline $\eta = 22{,}4$ %
  E10 $\eta = 21{,}8\%$

Using the equations (10) and (11) we obtain the temperatures in the center of the piston head and at the edge of piston head. [3]

- In the case of gasoline supply:
T_{cp} = 523 K - piston head center temperature T_{ep}= 474 K - piston head edge temperature
- In the case of ethanol – gasoline blend (E10) supply:
T_{cpE} = 464 K - piston head center temperature T_{epE} = 415 K, .- piston head edge temperature i.e. a reducing a level of temperatures in the center and at the edge of the piston with 11-12%.

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 [5, 6, 7].

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.

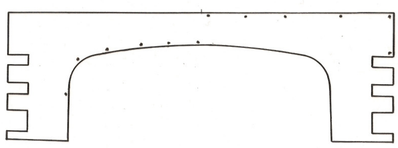

Fig. 2

The input data in the program are:
- longitudinal elasticity module, E = 0,75 10^{11} [N/m^2]
thermal conductivity, $\lambda_p = 160$ [W/mK]
Poisson's number, $\mu = 0{,}3$
linear thermal expansion coefficient, α =20 10^{-6} [K^{-1}]
model of calculation : static
label : PLANE 82

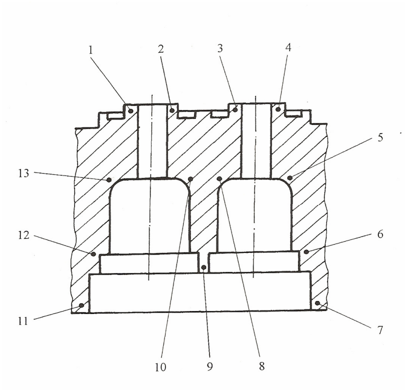

Fig. 3

It is considered a section through piston head and the temperature measured in steady conditions are input data. The points in Fig 2 and Fig 3 indicate he places were thermocouples are put.

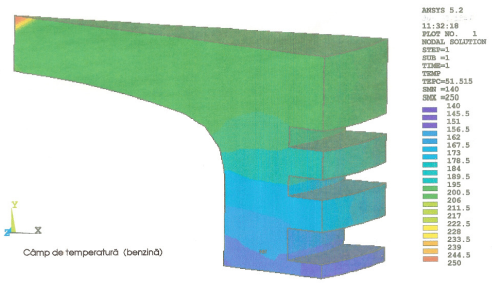

Fig. 4

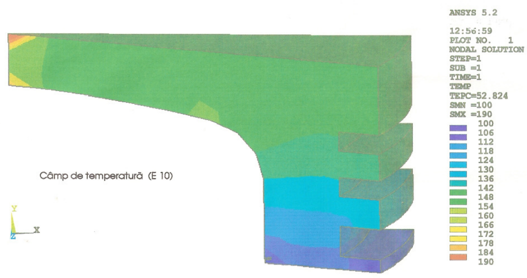

Fig. 5

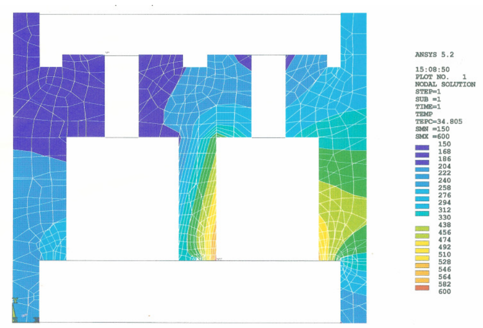

Fig. 6

The field of temperature in the piston head shows the lower level when the engine is supplied with gasoline - ethanol blend (Fig 4) than with gasoline (Fig 5). The field of temperature in cylinder head is shown in Fig. 6.

Conclusions

The methods of controlling the thermal stress level are dependent on the factors which determine this level. A definite permissible level of thermal loads corresponds to specific designs of parts, to the material used and cooling conditions.

The physical and chemical properties of the ethanol are significantly different from those of the conventional liquid oil fuels.

An efficient use of alcohols as fuels calls for construction modifications and adjustments to the engine in order to diminish the negative influences and turn into good account the good properties. That is why, to avoid modifications to the spark ignition engine, the use of the mixture E10 (10 % ethanol -90% gasoline) is worth being considered.

Using this mixture to fuel the spark ignition engine, a number of positive results are obtained, such as:

- a lower polytropic exponent of compression which results in lower pressures and temperatures by the end of the compression and burning and also lower burnt gas mean temperatures, [3, 4];
- the mean thermal stress of the spark ignition engine is lower when using E10 then gasoline in the same engine operating conditions;[4]
- the extent to which heat is saved by using E10 shows that, although the engine efficiency does not increase, supplying two types of fuels to the same engine represents an important research trend; [3]
- the tendency to reduce the effective power at a constant fuel rate as a consequence of the combustion value which is much lower than that of the gasoline,
- the decrease of polluting emissions when using E10 indicates that the future belongs to those engines able to operate while protecting the environment, the atmosphere, i.e. life.

From what has been shown it is clear how the thermal loads can be reduced. Besides the use of cooled constructions which make possible an appreciable lower temperature of piston, particularly of its critical zones, or heat insulating coatings aid in reducing the temperature and temperature gradients of parts, the use of unconventional fuels could be a solution in reducing the temperature in the parts.

The experiments show the use of a blend of 10% ethanol with gasoline (E10) to fuel a spark ignition engine lead to a lower level of temperature field of the parts. This means the values of thermal stresses and deformations are lower when the same engine is fueled with a gasoline - ethanol blend.

Nomenclature

D [m]- piston bore
h [m]– piston head height
L [m] - a length scale, such as the cylinder bore
p [$N.m^{-2}$] – gas pressure
Q_{in} [W] – heat addition
V[m^3] – gas volume
x_b – heat release fraction x_b= 0,90 – 0,99
α [$W\ m^{-2}\ K^{-1}$] - heat transfer coefficient
α_g [$W\ m^{-2}\ K^{-1}$] - instantaneous heat transfer coefficient
$\overline{\alpha}_g$ [$W\ m^{-2}\ K^{-1}$] - average heat transfer coefficient
λ [$W.m^{-1}\ K^{-1}$] - working fluid thermal conductivity.
λ_p [$W.m^{-1}\ K^{-1}$] - piston thermal conductivity
γ - adiabatic exponent
$\varphi_1 = \varphi_1\ (D,\lambda)$ – coefficient as a function of bore and thermal conductivity
ϕ [W] - heat flow taken by the piston
z- ethanol percentage in the ethanol – gasoline blend

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