MODELING THE HEAT TRANSFER IN THE PISTON HEAD OF A SPARK IGNITION ENGINE SUPPLIED WITH ETHANOL–GASOLINE BLEND

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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.

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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.

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°C. The minimum temperature (about 160-180 °C) 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.

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.[1]

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.

Using Fourier’s equation, the conduction resistance is:

\[ R_{\text{cond}} = \frac{\Delta T}{Q} = \frac{L}{\lambda} \] (1)

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

\[ R_{\text{conv}} = \frac{\Delta T}{Q} = \frac{1}{\alpha} \] (2)

3. PISTON MESH

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 2 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 [2].

As mentioned earlier, the piston can be treated as steady and driven by an average heat flux since the penetration layers are small.

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

\[ \bar{T}_g = \frac{1}{4 \cdot \pi \cdot \alpha_g} \int_{0}^{4\pi} \alpha_g T_g \, d\theta \] (3)

\[ \alpha_g = \frac{1}{4 \cdot \pi} \int_{0}^{2\pi} \alpha_g \, d\theta \] (4)

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 [3].

The finite heat release model can include the differential heat transfer \( \delta Q_w \) to the cylinder walls, if the instantaneous average cylinder heat coefficient \( h_g (\theta) \) and speed engine \( n \) are known. The finite release equation with addition of wall heat transfer is:

\[ \frac{dp}{d\theta} = \frac{\gamma - 1}{V} Q_{\text{in}} \frac{dx}{d\theta} - \frac{\delta Q_w}{d\theta} - \frac{\gamma p}{V} \frac{dV}{d\theta} \] (5)

The heat transfer rate at any crank angle \( \theta \) to the exposed cylinder wall at an engine speed \( n \) is determined with a Newtonian convection equation [3]:

\[ \frac{\delta Q_w}{d\theta} = \alpha_g (\theta) A_w (\theta) (T_g (\theta) - T_w) / n \] (6)

The cylinder wall temperature \( T_w \) in the above equation is the area-weighted mean of the temperatures of the exposed cylinder wall, the head and the piston crown.

The heat transfer coefficient \( \alpha_g (\theta) \) is the instantaneous area averaged heat transfer coefficient.

The exposed cylinder area \( A_w (\theta) \) is the sum of the piston crown area, the cylinder bore area, and the cylinder head area, assuming a flat cylinder head [4].

\[ A_w (\theta) = A_{\text{piston}} + A_{\text{wall}} + A_{\text{head}} \] (7)

4. FIELD OF TEMPERATURE IN THE PISTON

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:

\[ \frac{\partial^2 T}{\partial r^2} + \frac{1}{r} \frac{\partial T}{\partial r} + \frac{\partial^2 T}{\partial z^2} = 0 \] (8)
Or using undimensional coordinates: \( \eta = \frac{r}{R} \) si \( \xi = \frac{z}{\delta} \) the equation is written [5]:

\[
\frac{\partial^2 \Theta}{\partial \eta^2} + \frac{1}{\eta} \frac{\partial \Theta}{\partial \eta} + \frac{R^2}{\delta^2} \frac{\partial^2 \Theta}{\partial \xi^2} = 0
\] (9)

The solution is given by integration:

\[
T = \bar{T}_g - \Theta = \bar{T}_g - \frac{\phi}{10^6 \pi} \frac{D^2}{4} \left( \frac{1}{\alpha_g} + \frac{10^{-3} \delta}{\lambda_p} + \frac{10^{-3} D}{\lambda_p} \right) \] (10)

In the center of the piston head the temperature is:

\[
T_{cp} = \bar{T}_g - \frac{\phi}{A} \left( \frac{1}{\alpha_1} + \frac{h}{\lambda_p} \right)
\] (11)

At the edge of the piston head the temperature is:

\[
T_{ep} = \bar{T}_g - \frac{\phi}{A} \left( \frac{1}{\alpha_2} + \frac{h}{\lambda_p} \right)
\] (12)

5. CASE 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.

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) [9,10]. The effective power of the engine is \( P_e = 65 \text{ kW} \), the bore is \( D = 92 \text{ mm} \),

\( \bar{\alpha}_g = 350 \text{ W/m}^2 \text{ K} \) is an average heat transfer coefficient, \( \lambda_p = 160 \text{ W/m K} \) is piston thermal conductivity and \( \varphi_1 = 1.02 ; \varphi_2 = \varphi_3 (D, \lambda) \varphi_2 = 1.28 \).

We can also consider the heat flow in the piston is:

\( \phi = 0.1 P_e \)

- Molecular mass of fuel

\[
\begin{align*}
\mu_{\text{gasoline}} &= 110 \text{ [kg/kmol]} \\
\mu_{\text{ethanol}} &= 46 \text{ [kg/kmol]}
\end{align*}
\]

- Gravimetric composition

\[
\begin{align*}
c &= \frac{z \cdot 24 + (1-z) \cdot 0.855 \cdot 110}{z \cdot 46 + (1-z) \cdot 110} \text{ [kgC/kgfuel]} \\
h &= \frac{z \cdot 6 + (1-z) \cdot 0.145 \cdot 110}{z \cdot 46 + (1-z) \cdot 110} \text{ [kgH/kgfuel]} \\
o &= \frac{z \cdot 16}{z \cdot 46 + (1-z) \cdot 110} \text{ [kgO/kgfuel]}
\end{align*}
\] (13)

It results:

\[
\begin{align*}
gasoline \ c &= 0.855 \ h &= 0.145 \\
E10 \ c &= 0.84 \ h &= 0.144 \ o &= 0.015
\end{align*}
\]

- Polytropic compressing exponent

\[
\begin{align*}
n_g &= 1.36 \ \text{Fig. 3} \\
n_{\text{E10}} &= 1.25 \ [8]
\end{align*}
\]

\[
\begin{align*}
\text{Fig. 3}
\end{align*}
\]

- Parameters at the end of combustion

\[
\begin{align*}
gasoline \ p_z &= 62 \ 10^5 \text{ [N/m}^2\text{]} \ T_z = 2805 \text{ [K]} \\
E10 \ p_z &= 60 \ 10^5 \text{ [N/m}^2\text{]} \ T_z = 2705 \text{ [K]}
\end{align*}
\]

- Fuel consumption

\[
\begin{align*}
gasoline \ c_e &= 0.388 \text{ [kg/kWh]} \\
E10 \ c_e &= 0.413 \text{ [kg/kWh]}
\end{align*}
\]

- Mean temperature of the burnt gases

\[
\begin{align*}
gasoline \ T_g &= 1365 \text{ [K]} \\
E10 \ T_g &= 1305 \text{ [K]}
\end{align*}
\]

- Thermal efficiency

\[
\begin{align*}
gasoline \ \eta &= 22.4 \% \\
E10 \ \eta &= 21.8\%
\end{align*}
\]

Using the equations (11) and (12) we obtain the temperatures in the center of the piston head and at the edge of piston head.

- In the case of gasoline supply:

\[
\begin{align*}
T_{cp} &= 523 \text{ K - piston head center temperature} \\
T_{ep} &= 474 \text{ K - piston head edge temperature}
\end{align*}
\]

- In the case of ethanol – gasoline blend (E10) supply:

\[
\begin{align*}
T_{cpE} &= 464 \text{ K - piston head center temperature} \\
T_{epE} &= 415 \text{ K - piston head edge temperature i.e. a reducing a level of temperatures in the center and at the edge of the piston with 11-12%}.
\end{align*}
\]
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 [2].

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.

The input data in the program are:
- longitudinal elasticity module, $E = 0.75 \times 10^{11}$ [N/m²]
- thermal conductivity, $\lambda_p = 160$ [W/mK]
- Poisson's number, $\mu = 0.3$
- linear thermal expansion coefficient, $\alpha = 20 \times 10^{-6}$ [K⁻¹]
- model of calculation : static
- label : PLANE 82

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

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).

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 supply 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 supplied with a gasoline-ethanol blend.

Nomenclature

<table>
<thead>
<tr>
<th>Variable</th>
<th>Unit</th>
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<tbody>
<tr>
<td>$D$</td>
<td>[m] piston bore</td>
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<tr>
<td>$h$</td>
<td>[m] piston head height</td>
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<tr>
<td>$L$</td>
<td>[m] length scale, such as the cylinder bore</td>
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<tr>
<td>$p$</td>
<td>[N.m⁻²] gas pressure</td>
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<tr>
<td>$Q_0$</td>
<td>[W] heat addition</td>
</tr>
<tr>
<td>$V$</td>
<td>[m³] gas volume</td>
</tr>
<tr>
<td>$\chi_b$</td>
<td>heat release fraction; $\chi_b = 0.90$ - 0.99</td>
</tr>
<tr>
<td>$\alpha$</td>
<td>[W.m⁻².K⁻¹] heat transfer coefficient</td>
</tr>
<tr>
<td>$\alpha_i$</td>
<td>[W.m⁻².K⁻¹] instantaneous heat transfer coefficient</td>
</tr>
<tr>
<td>$\alpha_a$</td>
<td>[W.m⁻².K⁻¹] average heat transfer coefficient</td>
</tr>
<tr>
<td>$\lambda$</td>
<td>[W.m⁻¹.K⁻¹] working fluid thermal conductivity</td>
</tr>
</tbody>
</table>
\[ \lambda_p \text{ [W.m}^{-1}\text{ K}^{-1}] \] piston thermal conductivity
\[ \gamma \] adiabatic exponent
\[ \varphi_1 = \varphi_1 (D, \lambda) \] coefficient as a function of bore and thermal conductivity
\[ \phi \text{ [W]} \] heat flow taken by the piston
\[ z \] ethanol percentage in the ethanol - gasoline blend

REFERENCES


Acknowledgements

The authors would like to acknowledge the CNCSIS Romania for the financial support of the research within the grant CEEX 183/2006.

SEMINAR DE TERMODINAMICĂ, FIZICĂ STATISTICĂ ŞI APLICAŢII

2008

Marţi 22 ianuarie 2008 orele 14.00

**Prof. Vlad Popa-Nita**
(Facultatea de Fizică, Universitatea din Bucureşti)
**Amestec ternar cristal lichid-coloizi-impurităţi: diagrama de fază, tensiune interfacială, unde capilare**

Marţi 19 februarie 2008 orele 14.00

**Prof. Alexandru Morega**
(Catedra de Maşini, Materiale, A. Electrice, Facultatea de Inginerie Electrică, Universitatea Politehnica din Bucureşti)
**Aplicaţii ale supraconductoarelor de temperatură înaltă (High Temperature Superconductors - HTS)**

Miercuri 29 octombrie 2008 orele 14.00

**Prof. Gheorghe Popescu**
(Catedra de Termotehnică, Facultatea de Inginerie Mecanică, Universitatea Politehnica din Bucureşti)
**Cercetări teoretice şi experimentale pentru noi agenţi frigorifici ecologici**

Miercuri 12 noiembrie 2008 orele 14.00

**Dr. Ruxandra Crutescu**
AMVIC S.R.L. Bucureşti
**Proiectarea unei case pasive în oraşul Bragadiru**

Miercuri 17 decembrie 2008 orele 14.00

**Ing Nadine Laaser**
Universitatea din Berlin
**Passive House Planning Package (PHPP) implemented at AMVIC PH**