Influence of wall heat transfer on knocking in gasoline engines
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Abstract

The aim of this work is to develop a numerical calculation procedure for studies of two-dimensional, unsteady and compressible flow and heat transfer of the end-gas in a combustion chamber as it is compressed by a progressing flame front. Initially a Cartesian coordinate system is employed and the flame front is assumed to be straight. An implicit finite volume computer code based on the SIMPLE-algorithm is used. Preliminary prediction of variations of temperature, pressure and density as a function of time (crank angle) are presented.

1 Introduction

Knock occurs in gasoline engines especially during high load and low rotation speeds. Knock is believed to occur as the result of compression of the end-gas in the combustion chamber by the progressing flame front. If the temperature in the end-gas region becomes high enough for sufficiently long times the gas in the whole region will ignite very fast. This fast combustion can damage the engine and has to be avoided. This is normally done by monitoring the knock by a sensor and reducing it by retarding the ignition. Unfortunately the retardation of ignition reduces the efficiency of the engine. Avoiding knock then becomes a primary concern even when engines are fitted with knock sensors, see ref. [1].

As the flame front is progressing towards the cylinder wall the gas is further compressed and as an effect the gas temperature is increased. Energy may also be released due to chemical reactions (preflame reactions). The cooling of the cylinder wall and thus the wall temperature is very important for the eventual appearance of autoignition, see ref. [2]. The present work aims to model the heat transfer process during such circumstances. The project is a sub-project of a comprehensive project on knocking phenomena in gasoline engines.
2 Problem Statement

In this paper a two-dimensional and simplified calculation procedure of the importance of wall heat transfer for knock occurrence is presented. The simplified model adopted in this paper is sketched in Fig. 1.

![Diagram of simplified model](image)

Figure 1. Simplified model for the compression of the end-gas by the flame front.

The end-gas is compressed by the progressing flame front. The temperature of the flame front ($T_f$) is constant in time and its propagation velocity ($u_f$) is assumed to be known. The combustion chamber walls are kept at either a uniform temperature ($T_w$) or a specified heat flux ($Q_w$). At the starting time of the present calculation the unburned gas has a certain temperature distribution and a certain velocity distribution. These may be found experimentally or by a global CFD-calculation considering the whole engine cylinder and the moving piston. The aim of the present work is to predict the time development of temperature, pressure, density, velocity, etc. in the end-gas as the flame front propagates towards the wall. The mass conservation equation, momentum and energy equations are to be solved numerically. The gas is also assumed to follow the ideal gas law. By varying the wall temperature or wall heat flux, the importance of wall heat transfer on knocking occurrence may be simulated.

3 Governing Equation

In this section the governing equations of mass conservation, momentum and energy are given. These equations are valid for two-dimensional, unsteady and compressible flow. The equations given are applicable for laminar and turbulent flow.
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Continuity:

$$\frac{\partial \rho}{\partial t} + \frac{\partial (\rho u_j)}{\partial x_j} = 0$$  \hspace{1cm} (1)

Momentum:

$$\frac{\partial (\rho u_i)}{\partial t} + \frac{\partial (\rho u_i u_j)}{\partial x_j} = -\frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_j} \left[ (\mu + \mu_\Omega) \frac{\partial u_i}{\partial x_j} \right] + \frac{\partial}{\partial x_j} \left[ (\mu + \mu_\Omega) \frac{\partial u_j}{\partial x_i} \right]$$  \hspace{1cm} (2)

Energy:

$$\frac{\partial (\rho T)}{\partial t} + \frac{\partial (\rho u_j T)}{\partial x_j} = \frac{\partial}{\partial x_j} \left[ \left( \frac{\mu}{Pr} + \frac{\mu_\Omega}{Pr_\Omega} \right) \frac{\partial T}{\partial x_j} \right]$$  \hspace{1cm} (3)

The turbulence field is modelled by the standard $k$-$\varepsilon$ model, see [3]. The equations of $k$ and $\varepsilon$ are:

Turbulent kinetic energy $k$:

$$\frac{\partial (\rho k)}{\partial t} + \frac{\partial (\rho u_j k)}{\partial x_j} = \frac{\partial}{\partial x_j} \left[ \left( \frac{\mu + \mu_\Omega}{\sigma_k} \right) \frac{\partial k}{\partial x_j} \right] + \mu_\Omega \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \frac{\partial u_i}{\partial x_j} - \rho \varepsilon$$  \hspace{1cm} (4)

Turbulent dissipation $\varepsilon$:

$$\frac{\partial (\rho \varepsilon)}{\partial t} + \frac{\partial (\rho u_j \varepsilon)}{\partial x_j} = \frac{\partial}{\partial x_j} \left[ \left( \frac{\mu + \mu_\Omega}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial x_j} \right] + C_1 \mu_\Omega \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \frac{\partial u_i}{\partial x_j} \varepsilon - C_2 \rho \frac{\varepsilon^2}{k}$$  \hspace{1cm} (5)

The turbulent viscosity $\mu_\Omega$ is calculated from:

$$\mu_\Omega = \frac{\rho C_\mu k^2}{\varepsilon}$$  \hspace{1cm} (6)

The constants in the turbulence model are:

$$C_\mu = 0.09 \quad ; \quad C_1 = 1.44 \quad ; \quad C_2 = 1.92 \quad ; \quad \sigma_\varepsilon = 1.3 \quad ; \quad \sigma_k = 1.0 \quad ; \quad Pr_\Omega = 0.89$$

4 Boundary Conditions

In order to compensate for the velocity of the flame front (the moving boundary) a convective term $\rho u_f A\Phi$ is added to all $\Phi$ differential equations.
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(Φ= u, v, T, etc.) as a source term similar to the procedure in [4]. The walls of the combustion chamber have either a prescribed constant temperature (T_w) or a prescribed constant heat flux (Q_w).

If laminar flow prevails, the velocities are set to zero at the solid walls and the fluid temperature equals the wall temperature. When turbulent cases are considered the standard wall functions are introduced, see e.g. [5]. This means that velocities, temperature, turbulent kinetic energy and dissipation at the wall adjacent grid points are matched to wall functions.

5 Numerical Method

A transient, compressible, two-dimensional and implicit finite volume code based on the SIMPLE-algorithm in Cartesian coordinates is employed. The code is developed in collaboration with Department of Mechanical Engineering, University of Rhode Island, USA.

The code uses a staggered grid location for vectorial quantities while scalar quantities are stored in the centre of each control volume. The convection and diffusion terms are treated by the Power-law-scheme, see [6].

Besides the regular pressure-velocity coupling (which is treated by the SIMPLE-algorithm) there is a coupling between the density correction and pressure correction (as \( \rho' = \frac{\partial \rho}{\partial \rho} \times P' \)) and the coefficients in the discretized pressure correction equation are being updated with respect to the new density in each iteration, see [7].

The number of nodes in the x-direction was chosen to 32 and the number of nodes in the y-direction was chosen to 22. The computations were carried out for 30 time steps so that the flame front was moved one control volume, in the x-direction, at each time step.

6 Sample Calculations

In order to check the model and find out how it works, adiabatic walls (Q_w= 0) were considered in the first sample calculation and there was thus no heat transfer between the gas and the surrounding environment in this case. This calculation was carried out only for laminar flow.

In this paper, in order to simplify the problem, the properties of the end-gas (n-heptane) are regarded as constants.

The values used for temperature and velocity of the flame front were:

\( T_f = 1900 \text{ K}, \quad u_f = 5.12 \text{ m/s} \)

The following initial values were used:

- \( P_{\text{start}} \): Initial pressure field of the end-gas = 2.11 MPa
- \( T_{\text{start}} \): Mean temperature of the end-gas = 1060 K
The values given above were taken from an overall and global CFD-calculation considering a specific test engine cylinder with a moving piston. Further details can be found in [8]. A corresponding experimental investigation is presently carried out and results for this test engine will be available at a later time, see [9,10].

The Prandtl number of the gas was set to $Pr = 1.02$

A second sample calculation was performed for $T_w = 400$ K. The flow field was also in this latter case laminar.

7 Results

7.1 Case 1, $Q_w = 0$

As the flame front is progressing towards the right hand wall (Fig.1) both the pressure and temperature are increased, see Figures 2 and 3. Over a short time period (corresponding to 12 degrees increase in crank angle) the average pressure is increased a factor of 4 while the average temperature approaches the flame temperature. However, experimental investigation and simplified overall simulations, see [11], show a more rapid temperature rise. This deviation in the results may be due to the neglect of any heat release by chemical reactions (preflame reactions) in the present model. There are also some weaknesses in the two-dimensional model which have to be relaxed.

The following definitions of the dimensionless pressure and temperature are used:

\[ P_1 = \frac{P}{P_{\text{start}}} \]  
\[ T_1 = \frac{(T - T_{\text{start}})}{(T_f - T_{\text{start}})} \]

Figure 2. Variation of average pressure as a function of crank angle
Figure 3. Variation of average temperature as a function of crank angle

Figure 4 shows a temperature distribution at a time corresponding to the crank angle 416 degrees (x-axis is scaled up 7.5 times). The dimensionless temperature $T'$ is here defined as:

$$T' = \frac{T - T_w}{T_r - T_w}$$  \hspace{1cm} (9)

Figure 4. Temperature field of the end-gas for Crank angle 416 degrees

Close to the flame front the isotherms have an irregular shape. This is partly due to the inlet data and partly due to insufficient number of control volumes. Close to the vertical wall the shape of the isotherms becomes smoother and also the temperature gradient is high.

The dimensionless density is shown in Fig.5. The density is increased at each time step, due to the decrease of the volume (constant mass of end-gas) and change of the temperature and increase in pressure. The density is presented in dimensionless form as:

$$\rho_1 = \rho/\rho_{\text{start}}$$  \hspace{1cm} (10)
7.2 Case 2, $T_w = 400$ K

In this case a prescribed constant temperature on the walls is used but since the same initial field as in case 1 was used, there is only a slight difference compared with case 1. Actually the real time for compression in this work is 1.76 ms. Over such a short time the effect of wall temperature can hardly be seen but in the real situation the effect of wall temperature may be very important since the initial field is affected by the wall temperature boundary condition.

8 Concluding Remarks

A simplified two-dimensional numerical model has been used in order to solve the mass conservation, momentum and energy equations for the unsteady and compressible flow of the end-gas in a combustion chamber. An implicit, finite volume computer code based on the SIMPLE-algorithm was employed. The flame front was considered as a moving boundary. Variations of pressure, temperature and density for a few sample calculations were presented. The average pressure was increased a factor of 4 over a time period of 1.76 ms (or 12 degrees increase in crank angle). The increase of temperature was less than in reported experimental investigations because the heat release by chemical reactions (preflame reactions) was neglected in the calculations and because of the presence of simplifications in the two-dimensional model.

9 Acknowledgement

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10 Nomenclature

B: Additive constant in the law of wall
C₁: Constant in the k-ε model
C₂: Constant in the k-ε model
C₅: Constant in the k-ε model
k: Turbulent kinetic energy
P: Pressure of the end-gas
P₁: Dimensionless pressure
Pₚₐrₜ: Initial pressure of the end-gas
P': Pressure correction
Pr: Prandtl number
Prₜ: Turbulent Prandtl number
Qₚₜ: Specified heat flux at the walls
T: Time
T: Temperature of the end-gas
T₁: Dimensionless temperature
T': Dimensionless temperature
Tₚ: Temperature of the flame front
Tₚₐrₜ: Initial temperature of the end-gas
Tₖ: Temperature of the wall
u: Velocity of the end-gas in x-direction
uf: Velocity of the flame front
v: Velocity of the end-gas in y-direction
x: Coordinate
y: Coordinate
ε: Turbulent dissipation
κ: Karman's constant
μ: Dynamic viscosity
μₜ: Turbulent viscosity
ρ: Density of the end-gas
ρ₁: Dimensionless density
ρ': Density correction
σₖ: Constant in the k-ε model
σₐ: Constant in the k-ε model
11 References


8. G. Fernström (to be published) 1996.

