

Mathematical modelling of the HC Emission Mechanism in a Model SI Engine

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ABSTRACT

In this study, the diffusion of HCs, into the cylinder, which was compressed into the piston-crevice during compression and combustion stroke, was simulated by finite-volume technique. The in-cylinder flow pattern is modified by the flow of unburned piston-crevice and oil film HCs during the exhaust stroke. This in turn effects the concentration distribution and the amount of engine-out HCs. The multi-dimensional model, which was simulated for an axisymmetric engine, was combined with a phenomenological model based on empirical information from the literature. This has meant a great reduction in complexity and computational requirements. In the model, the effect of oil layers together with the effect of piston-cylinder crevice and oxidation in the cylinder were taken into account. The variation of hydrocarbon mole-fraction by the crank angle at the exhaust manifold was found to be qualitatively in good agreement with the measurements. This is thought to be, basically due to the multi-dimensional framework and the turbulence model used. The model gives excellent predictions of piston crevice HC emissions for variations in engine speed, compression ratio, top land crevice volume, etc.

Keywords: Thermodynamic simulation model, HC concentration

INTRODUCTION

Some of the gasoline HCs entering the engine, does not burn during the normal flame propagation process due to several HC emissions *source mechanisms*. Subsequently some, but not all, of these unburned HCs will oxidise within the cylinder. Moreover, not all the unoxidized in-cylinder unburned HC exit the cylinder. Some HC remain inside the cylinder with the residual burned gases, mix with the next cycle's fuel and air, and participate in the next cycle's combustion and emission processes. As a result of the processes summarised above engine-out HC emission levels become 1.5-2.0 % of the gasoline fuel flow into the engine. At the present time, the most widely accepted mechanisms for HC emissions in the exhaust gases of the SI engines are;

Fuel-air mixture compressed into the combustion chamber crevice volumes which keeps the fuel vapour unburned (contributes about 40% of the total losses).

- The oil film on the cylinder liner absorbing the fuel vapour (about 20%).
- Surface deposits absorbing the fuel vapour in a manner akin to the above case (about 20%).
- Incomplete combustion of fuel-air mixture near flammability limit.
- Quench layers on the combustion chamber wall left as the flame extinguishes close to the walls (more than 5%).

- Liquid fuel within the cylinder that does not evaporate and mix with sufficient air (10%).
- Leakage of unburned mixture through the (nominally) closed exhaust valve (about 5%).

The piston-crevice is the most important source of unburned HCs among all crevices. The high pressure created during compression and combustion phase's forces the mixture to squeeze into the piston-crevice. Since these crevices are very small, their surface to volume ratio becomes quite large and consequently, the mixture temperature in the crevice will quickly decrease almost to the wall temperature due to the heat transfer to the coolant.

The pressure in the cylinder increases during compression and combustion. Unburned and then the burned mixture is pushed into the region above the rings, and the regions behind and between the rings. During the expansion stroke, part of the mixture in the top-land crevice expands out into the combustion chamber. However, part of the gas in the top-land crevice continues to flow into the regions behind and between the rings. The flow of gas into the regions behind and between the rings continues in the expansion stroke until the increasing pressure within this crevice region becomes equal to the decreasing pressure in the cylinder. At this point (~ 120° ATC) the flow reverses direction, and the trapped gas flows out of the volumes behind and between the rings into the cylinder.

The observed jet-like flow at about 120° ATC does correspond to the reversal in direction of the flow into and out of the regions behind and between the rings. This flow reversal was not observed before 30-50° ATC. This earlier gas expansion out of the top-land crevice occurs at low velocity and the gas therefore remains against the wall; thus, it is obscured from observation in the experiments by the dark thermal boundary layer (Namazian *et al.*, 1982).

The outflow of HC from the piston crevice during the expansion process are mostly oxidised (Min *et al.*, 1994), except for a thin layer adjacent to the wall less than the boundary layer thickness. Liner is moving upwards at a speed faster than the crevice outflow gas velocity.

Thus the outflow of crevice gas is stretched to a thin layer which is laid down in a laminar fashion along the liner. In the expansion process, the turbulent motion of the charge to mix with the hot bulk gas strips the layer of crevice gas on the liner. Results from oxidation kinetics calculation (Weiss *et al.*, 1981) have indicated that the crevice gas would oxidise completely if the temperature is above ~ 1300 K.

Part of this crevice gas would exhaust in the blowdown process. The upward moving piston to form a scrape-up vortex will scrape up the remaining part, which is located along the liner. At the end of exhaust process, a part of this vortex will be pushed out of the cylinder by the piston. The amount of outflow from these crevices is not significant in the exhaust displacement process (after the blowdown process) during which the chamber pressure is approximately constant. The piston crevice outflow in the blowdown process is trapped at the centre of the piston scrape-up vortex, and is therefore not likely to exit the cylinder (Min *et al.*, 1994).

Previous modelling studies have been reported by, for example, Shyy and Adamson (1983), Lavoie *et al.* (1980), Dent and Lakshminarayanan (1983). These studies are essentially phenomenological, and include as much mechanism as possible with a detailed modelling in each. Schramm and Sorenson (1990) suggested a more simplified approach with reasonably accurate trend-wise predictions. Studies by Trinker *et al.* (1993), Min and Cheng (1994) and Schramm *et al.* (1990) are supported by simple one-dimensional finite difference solutions for oil film HC contribution.

Moreover, it is also assumed that the rate at which HCs pass through the exhaust valve is proportional to the total exhaust gas discharge. However, during the exhaust process in-cylinder vortices, especially those created by ring causes the assumption deviate considerably from the reality.

In that case it is important to predict the air motion in the cylinder, in a multidimensional manner, for the exhaust period. This prediction is expected to modify the figures for the amount of ring crevice HC entering the high temperature core and the amount to be oxidised.

The model predicts piston-crevice HC emissions against the engine speed, compression ratio, top land crevice volume. The results are encouraging in which the qualitative trends compare well with the various experimental data.

MATHEMATICAL MODEL

The model consists of two main parts, one dealing with the solution of flow, concentration and temperature fields by the finite volume techniques another dealing with the thermodynamic nature of the problem.

MULTIDIMENSIONAL FRAMEWORK

The work presented in this paper is for 2-dimensional axisymmetric representation of in-cylinder engine flow. Although this is somewhat removed from the 3-D flow in real engines, it still retains many of the features of engine flow and allows one to use high grid densities without incurring CPU time penalties.

The geometry consist of centrally located one valve and a flat piston. Numerical grids divide the engine chamber into enough number of control volumes so that the grid independency is achieved. For temporal accuracy, computations must be performed over small crank-angle intervals.

The fundamental equations that govern the in-cylinder flow processes, like any other compressible flow, are the Navier-Stokes equations for conservation of momentum, continuity equation for the conservation of mass and the stagnation enthalpy or specific internal energy equation for the conservation of energy, together with an equation of state of the gas relating local values of density, pressure and temperature.

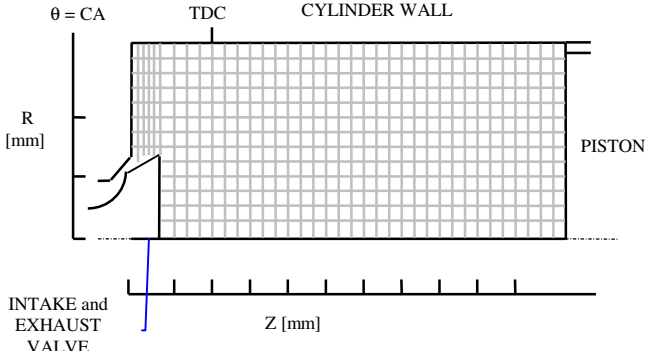


Figure 1. Schematic Illustration of the Computational Grid and Boundaries of the Solution Domain.

In the case of reacting flows, additional equations are required for the concentrations of chemical species and for the reaction rates. All of these conservation equations are non-linear, time dependent and partial differential equations. Recasting these equations in terms of Favre-averaged quantities is the most appropriate averaging procedure for calculating the in-cylinder flow (Gosman, 1986). They are as follows:

Here $\tilde{\cdot}$, $\bar{\cdot}$ represent, density weighted ensemble averaging and time averaging respectively. Unknown correlation's of the fluctuating component, which arose from the averaging process, need to be determined with the aid of turbulence modelling. The standard two-equation $k-\epsilon$ turbulence model has some drawbacks for in-cylinder engine flows, which encounter high rate of compression and expansion. El-Tahry (1983) and Morel & Mansour (1982) proposed some modifications to the standard model to predict the correct behaviour of turbulence length scale development. The later is employed here.

All the transport equations and those for other scalar quantities, such as species concentrations, possess the following similar form within the cylindrical-polar co-ordinate system :

$$\begin{aligned} \frac{1}{V_c} \frac{\partial}{\partial t} (\rho V_c \varphi) + \frac{\partial}{\partial z} (\rho U_r \varphi) + \frac{1}{r} \frac{\partial}{\partial r} (\rho r V \varphi) \\ = \frac{\partial}{\partial z} (\Gamma_\varphi \frac{\partial \varphi}{\partial z}) + \frac{1}{r} \frac{\partial}{\partial r} (r \Gamma_\varphi \frac{\partial \varphi}{\partial r}) + S_\varphi \end{aligned} \quad (1)$$

Here V_c is the instantaneous cylinder volume, U_r is the mean axial fluid velocity relative to moving grid lines, V is the radial velocity, ρ is the fluid density, φ is the subject of the transport equation, Γ_φ is the diffusivity and S_φ is the source term of φ . Details of the source terms and the definitions of the coefficients appearing in the above equations can be found, for example, in Gosman et. al. (1980).

Continuity equation:

$$\frac{\partial}{\partial t} (\bar{\rho}) + \frac{\partial}{\partial x_j} (\bar{\rho} \tilde{U}_j) = 0 \quad (2)$$

Conservation of momentum

$$\begin{aligned} \frac{\partial}{\partial t} (\bar{\rho} \tilde{U}_i) + \frac{\partial}{\partial x_j} (\bar{\rho} \tilde{U}_j \tilde{U}_i) = \frac{\partial}{\partial x_j} (\mu_r \frac{\partial \tilde{U}_i}{\partial x_j}) - \frac{\partial \bar{p}}{\partial x_j} \\ + \frac{\partial}{\partial x_j} (\mu_r \frac{\partial \tilde{U}_j}{\partial x_i}) - \frac{2}{3} \frac{\partial}{\partial x_j} (\mu_r \frac{\partial \tilde{U}_1}{\partial x_j} + \bar{\rho} k) \end{aligned} \quad (3)$$

Conservation of energy

$$\frac{\partial}{\partial t} (\bar{\rho} \tilde{h}_s) + \frac{\partial}{\partial x_j} (\bar{\rho} \tilde{U}_j \tilde{h}_s) = \frac{\partial}{\partial x_j} (\Gamma_{h,s} \frac{\partial \tilde{h}_s}{\partial x_j}) + S_h \quad (4)$$

Concentration of HC

$$\frac{\partial}{\partial t} (\bar{\rho} \tilde{y}_{HC}) + \frac{\partial}{\partial x_j} (\bar{\rho} \tilde{U}_j \tilde{y}_{HC}) = \frac{\partial}{\partial x_j} (\sigma_{HC} \frac{\partial \tilde{y}_{HC}}{\partial x_j}) + S_{HC} \quad (5)$$

Moreover, a method is employed which ensures that the grid network will always remain within the volume occupied by the fluid irrespective of the piston motion, by means of a transformation of the axial co-ordinate.

Thermodynamic model

The organisation of the thermodynamic model is shown in Figure 3. The term 'thermodynamic' refers in this instance to all the computational elements necessary to calculate the basic gas states throughout the engine cycle and the overall balance of energy for the engine system.

The assumptions made in the development of the model are the following;

1. The fuel is completely vaporised and has the composition $C_n H_{2n+2}$.
2. All substances are ideal gases with temperature dependent specific heats.
3. There is no dissociation of the products of combustion.
4. Heat transfer is calculated through the use of the Woschni (1967) heat transfer correlation.
5. It is assumed that the spark plug is located in the centre for the assumption piston crevice is filled-up with unburned mixture (Heywood, 1989).

The dimensions of the piston-cylinder geometry are summarised in Table 1.

Table 1. Model engine specification.

Type :	Single cylinder, iron block and liner	
Compression ratio	8.1	
Bore x stroke	96.50 mm x 86.40	mm
Connecting rod length	15.15	mm
Crank radius	4.32	mm
Compression Volume	8.9×10^{-5}	m^3
Stroke Volume	6.3×10^{-4}	m^3
Total Cylinder Volume	7.2×10^{-4}	m^3
Top Land Crevice Volume	0.93	m^3
Volume Behind the First comp. Ring	0.467	m^3
Volume Between The Rings	0.681	m^3
Volume Behind the First comp. Ring	0.467	m^3
Total volume between the rings	2.55	m^3
Valve timings	IVO : 10° BTC IVC : 34° ABC EVO : 31° BBC EVC : 55° ATC	

The thermodynamic model is based on the conservation of energy principle applied to the combustion chamber of the engine as a whole.

$$\delta Q_{FUEL} = dU + \delta W + \delta Q_{LOST} \quad (6)$$

In this equation; δQ_{FUEL} gain energy from fuel, δU , measurable inside energy of the mixture that is in the cylinder, δQ_{LOST} , heat transfer from the combustion bowl surface, δW , determining the work that is made by piston.

If we rewrite the equation for the difference of crank angle;

$$\frac{dQ_{FUEL}}{d\theta} = m_{MIX} c_v \frac{dT}{d\theta} + p \frac{dV}{d\theta} + \frac{dQ_{LOST}}{d\theta} \quad (7)$$

or alternatively :

$$T(\theta)_i = \frac{m_{burn}(\theta) H_{FUEL} + A(\theta)_i h(\theta)_i T_w(\theta)_i + m_{mix} c_v(\theta)_i T(\theta)_{i-1}}{A(\theta)_i h(\theta)_i + F[V(\theta)_i - V(\theta)_{i-1}] + m_{mix} c_v(\theta)_i} \quad (8)$$

At the equation above;

m_{mix} , total gas mass in the cylinder, T average gas temperature inside the cylinder, c_v , specific heat of the gas inside the cylinder (at constant volume) A , area of the surface where the heat loose from, h heat transfer coefficient, T_w , wall temperature, P , pressure inside the cylinder, where θ is the crank angle, C_p and C_v are specific heats at constant pressure and volume respectively. Calculation of specific heats for combustion products needs special care:

$$c_p = \frac{1}{M} (a_i + b_i T(\theta) + c_i T^2(\theta) + d_i T^3(\theta)) \quad (9)$$

and

$$c_v = c_p - \tilde{R} \quad (10)$$

Specific heats of the mixture then becomes:

$$c_{p_{mix}} = \frac{\sum_{i=1}^n N_i Y_i c_{p_i}}{\sum_{i=1}^n N_i Y_i} = \frac{\sum y_i c_{p_i}}{\sum y_i} \quad (11)$$

where Y_i is the molar weight of the i -th component of the mixture and M is the amount of the mass of the mixture before the exhaust valve opening. Here instantaneous density $\rho(\theta)$ is calculated as $\rho(\theta) = M/Vol_{cyl}(\theta)$. The viscosity of the gases can be calculated in a similar manner as a function of in-cylinder temperature as:

$$\mu = \mu_o \left(\frac{T}{T_o} \right)^{\bar{x}} \quad (12)$$

Here, \bar{x} is a constant and given as 0.71 by Perry (1962). The viscosity of mixture then becomes:

$$\mu_{o,mix} = \frac{\sum_{i=1}^n N_i \mu_{i,0} \sqrt{Y_i}}{\sum_{i=1}^n N_i \sqrt{Y_i}} \quad (13)$$

The heat release model is constructed on a simple trigonometric function used by Blumberg (1971) and Lavoie (1981). On the other hand, model based on the cosine type Wiebe function that gives the amount of unburned gas mass. Mass fraction of propane in the combustion chamber is;

$$m_{propane} = \frac{V_{cycle} \rho_{mix}}{[(HY)_{stoc} l + I]} \quad (14)$$

and the mass fraction of propane oxidised at every crank angle is,

$$m_{burned}(\theta) = \frac{1}{2} \left[1 - \cos \left(\frac{\pi(\theta - \theta_{aa})}{\theta_{ys} - \theta_{aa}} \right) \right] m_{propane} \quad (15)$$

calculated with the above expressions, where θ_{aa} is advance angle in the beginning of combustion and, θ_{ys} is angle of end of combustion. Apparent heat released heat rate from burned propane at every crank angle are calculated with equation (15).

$$Q_{app}(\theta) = m_{burned} Hu_{prop} \quad (16)$$

where Hu_{prop} Is lower heating value of propane. The amount of oxygen get into reaction during combustion process is given with equation (16),

$$M_{O_2} = \frac{m_y}{M_y} \lambda O_{min} M_{O_2} \quad (10) \quad (17)$$

total oxygen mass,

$$M_{TO_2} = \frac{m_y}{15.67\lambda O_{\min}} - 3.762m_{N_2} \quad (18)$$

calculated with the above equations. Mol fractions of fuel-air mixture and products which change depends on crank angle are calculated by the equation (19).

$$GN_i(\theta) = \frac{m_{burned}(\theta)}{M_y} N_i(\theta) \quad (19)$$

where M_y is molar fraction of gases.

CALCULATING THE HEAT TRANSFER COEFFICIENT INSIDE THE CYLINDER

Against to the issue that was developed by Annand ;

$$Nu = a Re^b, Re = \left(\frac{\rho U_p D}{\mu}\right), Pr = \frac{\mu c_p}{k}, \left(\frac{h_c D}{k}\right) = a \left(\frac{\rho U_p D}{\mu}\right)^b \quad (20)$$

where;

Re , Reynolds number, ρ , density of the gases inside the cylinder, m , mass of the gases inside the cylinder, V volume at the top of the piston, μ , viscosity of the gases inside the cylinder, U_p , average velocity of the piston, D , diameter of the cylinder.

$$h_c(\theta) = \frac{a}{Pr} \mu(\theta)^{1-b} \frac{c_p(\theta)}{m^{-b}} \frac{1}{D^{1-b}} \left[\frac{U_p}{V(\theta)} \right]^b \quad (21)$$

where, a and b are constants, \bar{U}_p is mean piston velocity, Pr is Prandtl number. and C_p is the specific heat (J/kgK) of the gasses inside the cylinder at constant pressure. At the spark ignited engines, prandtl number of the gases inside the cylinder does not change considerably against the temperature so that Pr could be accepted as 0.71. The thermodynamic model is then implemented in the multi-dimensional code mainly by use of subroutines.

POST FLAME OXIDATION MODEL IN CYLINDER

Post-flame oxidation in the cylinder is assumed to occur in a boundary layer near the cylinder wall, where the temperature is equal to the average of the wall and cylinder gas temperatures. Dent and Lakshminarayanan (1983) use the same assumption. If the temperature is assumed to be equal to the cylinder gas temperature, this will result in the complete oxidation of the hydrocarbons, if only there is sufficient oxygen available.

A one-step kinetic mechanism for the oxidation of propane is employed.

$$\frac{d[C_3H_8]}{dt} = -1.0 \times 10^{23} [C_3H_8][O_2] \exp\left(\frac{-25000}{T}\right) \quad (22)$$

where $[]$ denotes concentrations in moles per cubic centimetre, t time in seconds and T temperature in Kelvin. This oxidation model was obtained from fitting the results detailed of kinetics calculation based on the hydrocarbon oxidation mechanism developed by Heywood (1989). In the present model it is assumed that combustion products does not involve any O_2 or unburned fuel.

Therefore fuel and oxygen concentrations in this equation is determined by solution of equation of conservation of species in which crevice and oil film HCs are introduced as source term. The boundary layer is assumed to leave the cylinder via the exhaust valve in the same proportion as that portion of the total cylinder contents which leaves the cylinder at a given time.

NUMERICAL SOLUTION

The numerical procedure is of the finite-volume variety in which numerical solutions of the PDEs are obtained first by dividing the solution domain into a finite number of separate cells then discretizing the conservation equations into algebraic equations and finally determining the values of the dependent variables at the centres of the cells. Integration is made over control volumes as formed by the computational grids where the staggered grid arrangement is employed. The integration over the time step (i.e. the crank angle) is performed using the implicit formulation. Source terms are treated as constants over each control volume and linearized wherever possible to enhance stability.

The diffusion terms are approximated by second-order accurate central-differences. Using the HYBRID scheme of Spalding (1972) treats the convection terms. The scalar quantities; pressure, HC concentration, enthalpy, turbulence variables, density and viscosity are located at the centre of cells denoted by capital letters P, N, S, W etc. All vectorial quantities are displaced to lie on the boundaries of scalar cells, following the well-established practice of staggered grid arrangement of Patankar (1981)].

The solution procedure that links the continuity, momentum and enthalpy equations in order to obtain the pressure field is a modified version of the PISO algorithm, which operates in a predictor-corrector fashion for engine flow calculation (EPISO). Here, though, a modification to the EPISO method, in the form of an inter-time-step iteration is employed, due to the strong coupling of the sub-model equations in question as described in (Gül, 1994).

Boundary conditions

The initial conditions for the average properties of the in-cylinder fluid were generated by use of the thermodynamic model. The proper temperature and flow field at the start of exhaust valve opening (EVO) were provided by starting the computation some 40° CA before the EVO in conjunction with the thermodynamic model. Although crevice out flow starts much earlier than this value, the amount of hydrocarbons out of the crevice is not considerable and they get laid along the cylinder wall due to the high cylinder pressure during this period.

The exhaust flow was assumed to exit the cylinder tangential to the valve seat angle with a uniform profile. Centre-line boundary conditions were zero gradient, with fluxes set to zero. For the wall boundary conditions *one-layer treatment* of Launder and Spalding (1974) is adopted.

The amount of HC released by the oil film depends on the temperature, pressure, engine speed, diffusion coefficient, film thickness and composition. Here this amount is calculated by using one-dimensional diffusion equation since it presents a one-dimensional behaviour due to very fast variation of the HC concentration on the oil film.

$$\frac{\partial c}{\partial t} = D \frac{\partial^2 c}{\partial x^2} \quad (23)$$

The boundary condition at the interface of cylinder gases and oil film is provided from the Henry's law (Karamangil 2000). It is also assumed that only the first 1/3 of the HCs released from the oil film is oxidised.

In order to predict the flow rate through the piston crevice, one need to know the pressure differences between the cylinder and piston-crevice. This pressure difference is assumed to fit the results obtained with the mathematical model of Namazian & Heywood (1982). Since in this model pressure difference occurs after 120° ATDC, it meets the requirements for the period subject to solution (i.e. exhaust period which starts 149° ATDC. This is done by modifying the source term of the concentration equation at the cells of the cylinder-wall piston corner by introducing the volumetric flow rate as (Yildirim, 1997):

$$v_{mix.} = AC \sqrt{\frac{\Delta P}{\rho_{mix.}}} \quad (24)$$

where A is the cross-sectional area of the piston crevice, C is the discharge coefficient and ΔP is the

pressure difference. When we combine the above formulas, we can get an average output velocity of the mixture released from the crevice at one cycle.

$$v_{crev}(\theta) = \frac{\dot{m}(\theta)}{\rho_{crev}(\theta)A_{crev}} \quad (25)$$

Here; $v_{crev}(\theta)$: output velocity of the mixture from the area between piston-cylinder & ring, $\dot{m}(\theta)$: mass flow of the output mixture, $\rho_{crev}(\theta)$, density of the mixture (kg/m^3), A_{crev} : cross section area of the piston-cylinder & ring.

Density of the mixture between the piston – cylinder & ring is calculated by the following formula :

$$\rho_{rev}(\theta) = \frac{m_{y.crev}(\theta)}{V_{crev}} \quad (26)$$

A jet like flow of mixture from the ring gap into the cylinder was observed at about 480° CA with a velocity of 11-17 m/s Namarizan & Heywood (1983). The velocity given by (27,28) on the other hand is the overall velocity that corresponds to the total mass of mixture released from the piston cylinder gap and the jet velocity can be approximated as :

$$v_{rev}(\theta) = \frac{\dot{m}(\theta)}{\rho_{crev}(\theta)A_{ring}} \quad (27)$$

$$v_{crev}(\theta) = v_{crev.absolute} \pm U_p \quad (28)$$

The HC-concentration of piston-cylinder gap was calculated depending on the value of λ as

$$Y_{HC.crev} = \frac{1}{1 + \lambda \left(\frac{H}{Y} \right)_{STOK}} \quad (29)$$

RESULTS

The simulation is basically done during the periods, which cover late expansion and exhaust strokes. The combustion is assumed to be completed at 120° CA after TDC. The exhaust valve opens at 31° CA before BDC and closes at 55° CA after TDC in accord with those of Namazian & Heywood (1982) for comparison purposes. Average cylinder pressures and temperatures are given for various crank angles in

Table 2 for an engine operating between 1000-2000 rpm with a compression ratio of 8.1 and equivalence ratios between 0,7-1,2.

Engine Cycle		Press (bar)	Temp. (K)
Inlet valve closure	(214°)	1.05	293
Beginning of comb.	(322°)	10-14	550-740
Max.values during comb.	(368)	23-32	2130-2400
End of combustion	(480°)	12-15	1100-1300
Exhaust valve open	(509°)	3-5	980-1050
End of cycle	(720°)	1.05	620-740

Table 2. Computed values of pressures and temperatures at various points.

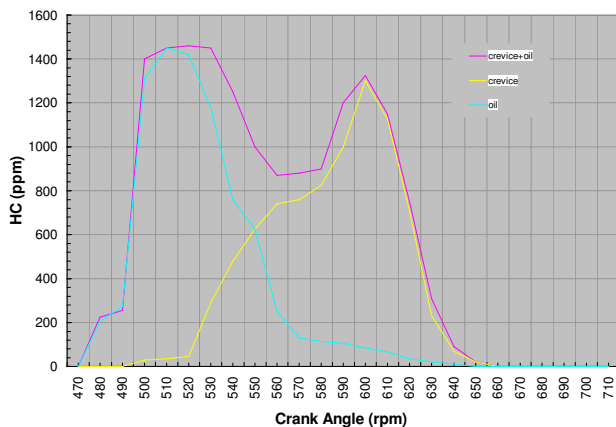
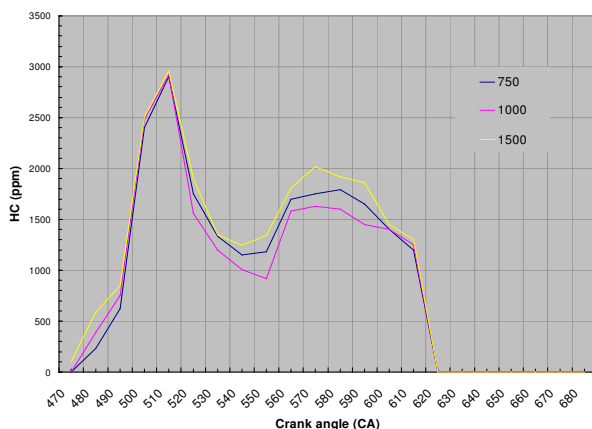


Figure 2. The effect of piston-crevice and oil-film hydrocarbons at the exhaust valve exit. N= 2000 rpm.

Figure 2 shows the components of HC concentration at the exhaust valve exit arising from both the piston-cylinder crevice and oil film. The results were provided from two runs of the computer code, one of which considers the oil film effect only and the other one considers piston-cylinder crevice effect. It can be seen from the figure that for a third run which takes into account the effect of both components, the superposition principle is only approximately true. It can also be observed that oil-film HCs leave the cylinder earlier than the piston-crevice HCs. One of the reasons for this is that during the late expansion stroke unburned fuel is released from the oil-film into the cylinder and the amount of this HC sharply reduces as the piston approaches BDC. Moreover, piston-crevice is further away from the valve exit compared with the oil-film so that HCs released from the oil-film reach the valve exit earlier than those of



the piston-crevice.

Figure 3. The HC concentration at the valve exit is plotted with the engine speed as a parameter

HC concentration at the valve exit is plotted with the engine speed as a parameter in Figure 3. The curves exhibit similar characteristics for engine speeds of 750, 1000, and 1500 rpm which can be considered as low engine speeds. Immediately after the exhaust valve opens very high values of HC concentrations are reached (approximately 3000 ppm). These values vary between 500-1700 ppm with the increasing crank angles and end at about 610° CA. Other predictions (not shown on the Fig.) together with these ones indicate that less unburned-HCs leave the cylinder with increasing engine speed.

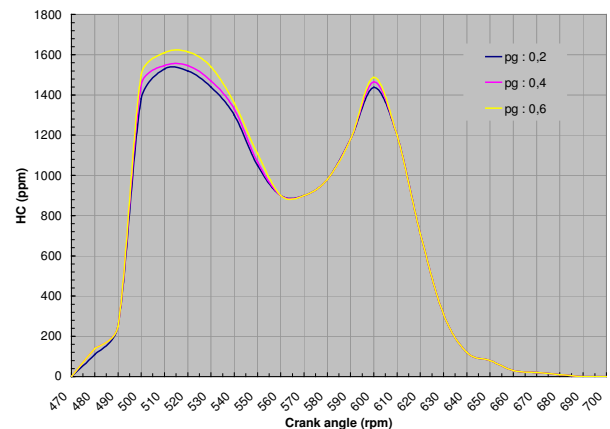


Figure 4 the effect of inlet pressure on the HC emission

In Figure 4 the effect of inlet pressure on the HC emission is presented. The combustion approach applied here does not consider the effect of inlet pressure on the combustion efficiency and the flame speed. Therefore, only minor differences were observed between 510-550° crank angles which is possibly due to the increase in the amount of HC released from the oil-film. Since only one central valve geometry was simulated it was not possible to see the effect of the ratio of the inlet pressure to the exhaust pressure which is a more prominent parameter on the exhaust emissions than the inlet pressure only.

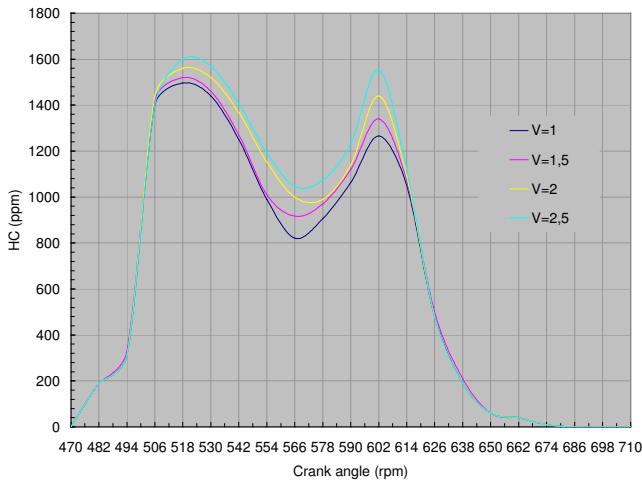


Figure 5. The effect of gap between piston-cylinder and piston rings is shown for 2000 rpm

The effect of gap between piston-cylinder and piston rings is shown for 2000 rpm in Figure 5. In an average size engine the volume of this gap varies between 0.6-1.2 cm³. As can be seen from the figure that with the increasing volume of the crevice the amount of HCs emitted from the valve exit increases.

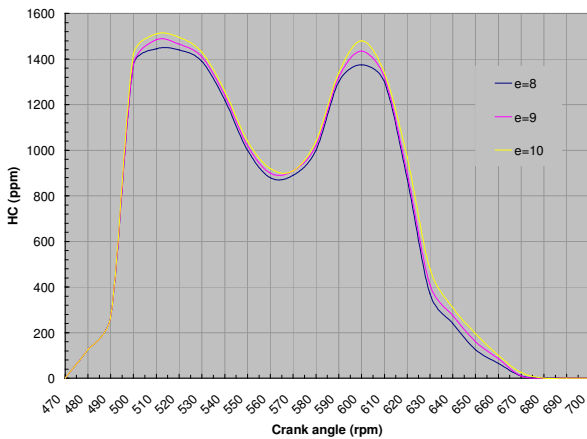


Figure 6. the effect of compression ratio on the HC emissions at the valve exit for engine speed of 2000 rpm.

Figure 6 shows the effect of compression ratio on the HC emissions at the valve exit for engine speed of 2000 rpm. With the increasing compression ratio average in-cylinder pressure increases and this result with an increased level of HC emission because the amount of unburned HCs pushed into the crevices increases. As can be seen from the figure this is also confirmed with the simulation but the differences is not as high as expected. This is possibly due to the lower amount of HCs forced in to the crevice by the model.

Conclusion

This study presents the finite volume analysis combined with a thermodynamic modelling of in-cylinder HC behaviour following exhaust valve opening in SI engines. The simulation comprises a finite-volume solution of governing conservation equations, a thermodynamic cycle simulation, which provide proper initial conditions and a turbulence model. The out-flow of piston-crevice hydrocarbons was incorporated into the code basically by properly modifying the boundary conditions of the piston crown-cylinder wall corner cells. On the other hand, oil-film hydrocarbons was modelled by solving one-dimensional time-dependent concentration equation together with the Henry's law.

Although our model simplification and assumptions don't match one-to-one its counterparts in the literature, qualitative trends are very much similar while quantitative results indicate that the same similarity might be obtained if some simplifications was avoided. The effect of of piston-crevice and oil-film hydrocarbons at the exhaust valve exit were compared, and the effect of other parameters such as engine speed, volume of piston crevice, compression ratio, and the inlet pressure were studied in depth.

Although the results are encouraging, additional studies are needed for better simulation of real engines. Such a more comprehensive study might include two or more valve configurations, double zone combustion modelling and effect of other contributants of HC emissions as well. Some of these are already being studied as extension of the previous work.

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