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# Mathematical modeling of hydrocarbon emissions from oil film for different fuels

M. Ihsan Karamangil [a,*], Ali Sürmen [a,b], Seckin Yenice [c]

[a] Uludag University, Faculty of Engineering and Architecture, Dept. of Automotive Eng., Bursa, Turkey
[b] Bursa Technical University, Rector, Bursa, Turkey
[c] Uludag University, Faculty of Engineering and Architecture, Dept. of Mechanical Eng., Bursa, Turkey

## HIGHLIGHTS

- HC absorption and release amounts by oil film are determined for five fuels.
- Henry constant and diffusion coefficient are the most important parameters.
- The highest amount of absorption and release is observed for iso-octane.
- The lowest amount of absorption and release is observed for methane.
- Fuels are ranked as HC emissions: iso-octane, methanol, ethanol, LPG and methane.

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## ABSTRACT

Oil film on the inner surface of the cylinder liner is one of the major sources of the vehicle-out HC emissions as fuel vapor is absorbed by the oil film under high pressure and then released after late expansion stroke when the pressure is low. This process is extensively affected by type of the fuel and lubricating oil. In this theoretical study, the effect of different engine parameters on oil film HC emissions for various fuels, such as iso-octane, methanol, ethanol, LPG and methane, is investigated. The results show that fewer HCs are released from the oil film when using gaseous fuels, such as LPG and methane, than when using liquid fuels. The fuels can be ranked according to their effect (from greatest to least) on HC emissions as follows: iso-octane, methanol, ethanol, LPG and methane. The most important parameters affecting the HC absorption/release mechanism are found to be Henry's coefficient and the diffusion coefficient. As interaction time of oil film-fuel vapor was longer at low engine speeds, the quantities of HC absorbed/desorbed increased. The quantities of HC absorbed/desorbed increased with increasing inlet pressure and compression ratio.



## 1. Introduction

The basic function of the oil film on the cylinder liner is to ensure lubrication between the piston rings, the piston skirt and the cylinder liner, thus decreasing friction and wear. This film, a few microns thick, absorbs HCs during inlet, compression strokes and combustion, and releases them during the expansion and exhaust strokes. It is estimated that 10–25% of the tailpipe HC emissions are from the oil film [1].

The oil film in a spark ignition engine is exposed to severe pressure and temperature changes during the power cycle. During the inlet stroke, the temperature and pressure are the lowest and the oil film absorbs a small amount of fuel vapor. During compression, the absorption increases due to higher pressure and temperature; however, as the piston moves towards the TDC, it covers more of the oil film area, reducing the area available for HC absorption. Only the uncovered film area continues to absorb fuel vapor. Fuel vapor absorption continues during combustion, until the flame reaches the oil film, resulting in a rapid increase of pressure and temperature. Then, the oil film most likely becomes saturated with fuel vapor. As the absorbed HC mass increases, it penetrates deeper into the film thickness.

During the expansion and exhaust strokes, the oil film is uncovered by the piston, the oil film is exposed to lower pressures and temperatures, and the HC concentration in the combustion products is nearly zero. This is a suitable atmosphere for the absorbed fuel to be released from the oil film into the cylinder space. If the released HCs are not exposed to the burning conditions during their travel through the exhaust port, they are included in the tailpipe HC emissions.

* Corresponding author. Tel.: +90 224 2941978; fax: +90 224 2941903.
*E-mail addresses:* ihsan@uludag.edu.tr (M. Ihsan Karamangil), surmen@uludag.edu.tr, ali.surmen@btu.edu.tr (A. Sürmen), seckinyenice@hotmail.com (S. Yenice).

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Many of the studies in literature are focused on behavior of oil film. This behavior is generally characterized by oil film thickness and oil film velocity map (or velocity distribution). Laser induced fluorescence (LIF) method is applied to measure oil film thickness and particle image velocimetry (PIV) method is used for determining the velocity map. Baba et al. [2] used these two techniques to determine the oil film behavior of an engine. Dwyer-Joyce et al. [3] measured the oil film thickness of a single-cylinder engine as 2–21 μm around piston skirt. Dhar et al. [4], on the other hand, measured the oil film thickness as 0.2–8 μm between the piston ring and cylinder liner in an engine simulator without combustion. In another study, where both method were used, Kato et al. [5] measured a radial gap of 30–100 μm between the piston skirt and the cylinder liner.

Korematsu [6] investigated the effect of fuel absorption by oil film on the unburned HC emissions in the exhaust. He modeled the absorption of fuel vapor into and desorption of it from oil film. It was founded out that quantity of unburned HC released from oil film depends on position of oil film, engine speed, diffusion coefficient, oil film thickness, number of engine cycles and fuel concentration in the oil film just before start of the desorption.

Yu et al. [7] studied the dynamic behavior of fuel absorption/desorption into/from oil the film respectively. They found out that the most effective parameter on this process is Henry constant. They also concluded that upper oil film regions contribute more to this process than the lower oil film regions. In a similar parametric study, based on engine speed, engine load and oil film temperature, Yu and Min [8] came up with a result that Henry constant regarding the solubility of fuel in the oil film is the most dominating factor in fuel absorption/desorption process.

Shenghua et al. [9] studied the effect of oil film on HC emissions. In their experiments with and without oil film they reasoned that effect of oil film in total HC emission of the engine is 28%. Sodre and Yates [10], and Frolund and Schramm [11] calculated the contribution of oil film in total HCs as 15–30% in their model studies.

Salazar and Ghandhi [12] declared that solubility of propane in the lubricating oil is approximately 10 times lower than that of gasoline.

This study is aiming to fill this gap to some extent. Regarding this, the authors numerically examined the effect of oil film on HC emissions for various alternative engine fuels such as ethanol, methanol, LPG and NG. The results are presented in a comparative form regarding some engine operational and constructive parameters.

## 2. Mathematical modeling

In this study, a mathematical model is developed to determine the oil film-originated HC emissions for various fuels used in SI engines. The model consists of two primary sub-models as follows:

1. Thermodynamic model
2. Oil film-originated HC emission model

In the thermodynamic model, the geometrical parameters of the engine and the fuel–air mixture properties are defined. Thermodynamic and gas properties of the cylinder contents, e.g., pressure, temperature, specific heat and viscosity, at every stage of the power cycle are defined as functions of crank rotation. The composition of the cylinder content must also be determined as a function of crank rotation. Then, the coefficient of convective heat transfer and the heat loss from the engine are formulated.

In the HC emission model, the effect of fuel type on the mass rate of absorption/release of fuel vapor in the oil film and the variation of this effect with engine speed, compression ratio, inlet pressure and excess air coefficient are formulated.

The model also includes the partial volume of the fuel vapor in the cylinder space and the cooling effect caused by the high latent heats of ethanol and methanol. Stoichiometric fuel–air ratios are assumed for all fuels.

### 2.1. Thermodynamic model

#### 2.1.1. Calculation of the temperature and pressure in the cylinder

Calculation of temperature and pressure is based on the first law of thermodynamics with introduction of heat losses into the heat term. The model is valid for the period between the opening and closing of the inlet valve.

For a closed system, the first law of thermodynamics can be written as follows:

$$\delta Q - \delta W = dU \tag{1}$$

$$(\delta Q_{in} - \delta Q_{loss}) - pdV = mc_v dT. \tag{2}$$

Using the ideal gas equation, $pV = mRT$, the following relationships are derived:

$$mdT = \frac{1}{R}(pdV + Vdp) \tag{3}$$

and

$$dU = \frac{c_v}{R}(pdV + Vdp). \tag{4}$$

Then, Eq. (2) is rearranged as follows:

$$(\delta Q_{in} - \delta Q_{loss}) - pdV = \frac{c_v}{R}(pdV + Vdp) \tag{5}$$

$$\left(1 + \frac{c_v}{R}\right)p\frac{dV}{d\theta} = \left(\frac{dQ_{in}}{d\theta} - \frac{dQ_{loss}}{d\theta}\right) - \frac{c_v}{R}V\frac{dp}{d\theta} \tag{6}$$

Using the definitions $R = c_p - c_v$ and $k = c_p/c_v$ the following equation is derived:

$$\frac{dp}{d\theta} = \frac{k-1}{V}\left(\frac{dQ_{in}}{d\theta} - \frac{dQ_{loss}}{d\theta}\right) - k\frac{p}{V}\frac{dV}{d\theta} \tag{7}$$

The finite difference equation can then be derived as follows:

$$p(\theta) = p(\theta-1) + \frac{k-1}{V(\theta)}\{[Q_{in}(\theta) - Q_{in}(\theta-1)] - [Q_{loss}(\theta) - Q_{loss}(\theta-1)]\} - k\frac{p(\theta-1)}{V(\theta)}[V(\theta) - V(\theta-1)] \tag{8}$$

An explicit solution method is used to solve the above finite difference equation, i.e., the results of the previous iteration are used for the next one. In the solution, $\Delta\theta$ is set as 1°, corresponding to 0.083 ms. The problem is also solved using a Runge–Kutta solution technique, and the results are nearly identical. The order of magnitude of the error in the Runge–Kutta method is $h^4$ ($8.33 \times 10^{-5}$); therefore, the results are satisfactory, and a different step size is not required.

The total amount of heat supplied to the cylinder by the fuel in one cycle is calculated as follows:

$$Q_{in,top} = m_{f0}H_u \tag{9}$$

where $m_{f0}$ is the mass of fuel introduced to the cylinder in one cycle and $H_u$ is the lower caloric value of fuel. The Wiebe function is used to determine the burning rate of the fuel as follows:

$$x_b = 1 - \exp\left[-a\left(\frac{\theta - \theta_s}{\Delta\theta}\right)^n\right] \tag{10}$$