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Heat Release Model of DI Diesel Engine: A Review

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Abstract: Heat transfer to the cylinder walls of internal combustion engines is recognized as one of the most important factors that in influences both engine design and operation. A heat transfer model has been developed that uses quasi-steady heat flux relations to calculate the heat transfer from combustion gases through the cylinder wall to the coolant in an internal combustion engine. The treatment of convective heat transfer accounts for the physical problems of rotating and impinging axial flow inside the engine cylinder. The radiative heat transfer includes gas radiation (CO2, H20, and CO) and soot-particle radiation. Cylinder wall temperatures can be accurately predicted from this model for both the gas and the coolant sides. The present model's heat transfer results for the motoring case are in good agreement with results from empirical correlations based on instantaneous heat flux data. The calculated radiative heat flux and gas emissivity show reasonable agreement with data in the literature.

Key Words: Diesel Engine; Transient Operation; Convective heat flow; Turbocharged Engine

INTRODUCTION

The Direct Injection (DI) diesel engine both naturally aspirated and turbocharged is a highly efficient heat engine suitable for use in road, rail and sea transport application and also for numerous stationary applications. Significant progress has been made in the last decade in further improving the thermal efficiency of the diesel engine and reducing soot and noise levels. [6] The process of heat release can largely the performance and emission characteristics of a diesel engine, when certain additional information is available such as the exhaust oxygen concentration, engine load level or ignition delay.[1-2]. The heat release of conventional diesel engines has traditionally been phase close to the end of the compression stroke i.e. top dead centre.[3] Cylinder head is one of the most complicated parts of internal combustion engine. It is directly exposed to high combustion pressures and temperatures. In addition, it needs to house intake and exhaust valve ports, fuel injector and complex cooling passages. Compliance of all these requirements leads to many compromises in design. As a result, cylinder heads tend to fail in operation (distortions, fatigue cracking) due to overheating in regions of limited cooling.

HEAT RELEASE MODELING

The basis for the modeling of the heat release is the first law of thermodynamics for an open system. Open system is that system in which the mass entering in the system and reject from the system. Assuming the cylinder charge as a single zone and using the ideal gas law, the heat release during combustion,dQ_{gr} on a crank angle basis is given by [1]:

$$\frac{dQ_{gr}}{d\theta} = \frac{1}{1-\gamma}\left[\gamma p\frac{dV}{d\theta} + V\frac{dp}{d\theta} + (u - C_v T)\frac{dm_c}{d\theta}\right] - \sum h_i \frac{dm_i}{d\theta} + \frac{dQ_{ht}}{d\theta} \quad (1)$$

APPARENT HEAT RELEASE MODEL

By neglecting the heat transfer, crevice volume, blow-by and fuel injection effects in eq (1), the resulting heat release rate is termed as the apparent or net heat release rate [4-5]

$$\frac{dQ_{app}}{d\theta} = \frac{dQ_{gr}}{d\theta} - \frac{dQ_{ht}}{d\theta} = \frac{1}{\gamma-1}\left[\gamma p\frac{dV}{d\theta} + V\frac{dp}{d\theta}\right] \quad (2)$$

Heat Release during Premixed and Mixing-Controlled Combustion Phases:

During the ignition delay period, some of the injected fuel is evaporated and mixed with air, forming a combustible mixture. Following ignition, the first phase of combustion occurs under premixed conditions at a rate RR_p given by the following Arrhenius type kinetic equation [7].

$$RR_p = B_1 \rho_{mix}^2 x_{fv} x_{ox}^5 exp\left(-\frac{1200}{T_z}\right) V_z \quad (3)$$

After the entire initial fuel vapor has been consumed, combustion is assumed to proceed to the mixing-controlled and late combustion phases. The combustion rate for the mixing-controlled and late combustion phase. RP_m are governed by the following expression

$$RP_m = \frac{B_2 m_{fv} P_{ox}}{P} P^{0.25} exp\left(-\frac{2500}{T_z}\right) \quad (4)$$

Convective heat losses with Cylinder wall

Convective heat losses are given by [8]:

$$dQ_{ht,conv} = Ah_c(T - T_{wall}) \quad (5)$$

The area A is calculated using the geometric properties of the engine. The gas temperature used is the global in cylinder gas temperature T. The wall temperature T_{wall} is fixed at 450 K and h_c is calculated using the Woschni expression [8]

$$h_c = CB^{m-1} p^m w^m T^{0.75-1.62m} \quad (6)$$

Heat Loss from Combustion Chamber

The heat loss from the combustion chamber can also be expressed by using following formula [9]:

$$\frac{dQ}{d\theta} = \frac{Q_l}{\omega} = \frac{Q_b + Q_u}{\omega} \quad (7)$$

Where

$$Q_b = hA_b(T_b - T_w) \quad (8)$$

$$Q_u = hA_u(T_u - T_w) \quad (9)$$

The surface areas of the two different zones are given by A_u and A_b. These areas can be related to a mass fraction burned x by using an empirical formula

$$A_b = \left(\frac{\pi d^2}{2} + \frac{4V}{d}\right) x^{0.5} \quad (10)$$

$$A_u = \left[\left(\frac{\pi d^2}{2} + \frac{4V}{d}\right)(1 - x^{0.5})\right] \quad (11)$$

Heat transfer is a must in IC engines to maintain cylinder walls, cylinder heads and piston faces at safe operating temperatures. Heat is transferred from or to the working fluid during every part of each cycle, and the net work done by the working fluid in one complete cycle is given by [9]:

$$W_{net} = \oint \left(p + \frac{\Delta p}{2}\right) \Delta V \quad (12)$$

Where Δp is the pressure change inside the cylinder as a result of piston motion, combustion, flow into or out of the cylinder and heat transfer.

The pressure change Δp due to heat transfer is given by

$$\frac{\Delta p}{p} = \frac{h_c A(T_w - T)}{M C_v T} \Delta T \quad (13)$$

WEIBE HEAT RELEASE MODEL

The weibe heat release pattern is based on the exponential rate of the chemical reactions. In this model, it is assumed that all the fuel is injected before the end of ignition delay itself.

The fraction of heat released is expressed by the non-dimensional equation

$$x = 1 - exp\left[-a\left(\frac{\theta-\theta_i}{\Delta\theta_c}\right)^{m+1}\right] \tag{14}$$

HEAT TRANSFERRING CALCULATION

The heat transferring quantity is evaluated by the NEWTON equations according to heat transferring theory [10]

$$\frac{dQ_w}{d\emptyset} = \frac{1}{6n}\sum_{i=1}^{3}\alpha_g A_i\,(T_{w1} - T) = \frac{1}{6n}\alpha_g[A_1(T_{wi} - T) + A_2(T_{w2} - T) + A_3(T_{w3} - T)] \tag{15}$$

HEAT TRANSFER CORRELATIONS

The instantaneous wall heat transfer can be calculated by means of a correlation such as the one

developed by Woschni [12]

$$q_w = hA(T_w - T) \tag{16}$$

$$h = 0.00326p^{0.8}\frac{(v_{mot}-v_{comb})^{0.8}}{B^{0.2}T^{0.53}} \tag{17}$$

$$v_{mot} = c_1 v_{pis} \tag{18}$$

$$v_{comb} = c_2\frac{v_d T_1}{p_1 v_1}(p - p_{mot}) \tag{19}$$

$$v_{pis} = \frac{2Srpm}{60} \tag{20}$$

GAS-WALL HEAT TRANSFER

The film coefficient, h_{gas}, necessary to calculate the conductance between the gas and the walls, is obtained with an enhanced version of Woschni equation [11]:

$$h_g = 1.2\cdot 10^{-2}D^{-0.2}p^{0.8}T_g^{-0.53}\cdot\left[(C_{w1}c_m + C_{w2}c_u) + C_2\left(\frac{V_T T_{CA}}{p_{CA}V_{CA}}\right)(p - p_0)\right]^{0.8} \tag{21}$$

Here, D is the bore; T_g the instantaneous gas temperature calculated with the measured in-cylinder pressure, p; c_m is the mean piston speed; c_u is the tangential velocity at the cylinder wall due to swirl; V_T is the displacement volume; T_{ivc}, p_{ivc} and V_{ivc} are the gas temperature, pressure and cylinder volume at intake valve closing (IVC) and p_0 is the in-cylinder pressure under motoring conditions.

In the wall temperature model the heat transfer coefficient between the gas and the walls is calculated with the previously described formula using the instantaneous gas temperature, $T_{gas}(\alpha)$, and pressure, $p(\alpha)$, from a homemade combustion predictive program. Then the cycle average heat transfer coefficient and gas temperature are calculated.

PISTON – CYLINDER LINER HEAT TRANSFER

Because measurements were available of both the piston and the liner temperature, an empirical model could be fitted to the data. It was supposed that the segments have a conductance K_{seg} per unit of length. The possible influence of the piston speed was turned to be not significant. Hence the conductance between a node of the piston "*Pis_i*" and a node of the liner "*Lin_j*" which make contact through a segment is given by the following formula:

$$K_{Pis\_i-Lin\_j} = \frac{t_{con}}{T_{cycle}}K_{seg}\alpha_{Lin\_j}\frac{D}{2}$$

Where, t_{con} is the contact time between the segment and the liner node, T_{cycle} the duration of a cycle, α_{Lin_j} the angular width of the liner node and D the bore. The contact time is calculated from the instantaneous piston position taking into account the position of the segment and the axial position of the liner node.

LINER – OIL HEAT TRANSFER

Oil is continuously splashed against the cylinder wall and, in the piston some channels coming from the cooling gallery feed the third groove. So the cylinder wall is continuously wetted with oil. This oil is heated by the cylinder wall and scrapped of during the downward stroke. For this conductance ($K_{lin\text{-}oil}$), a piston speed dependence was taken into account just like for the oil – piston heat transfer. During the optimization phase of the model this speed dependence did not appear to be very significant and only the constant part was included in the model. An equivalent heat transfer coefficient, $h_{lin\text{-}oil}$, for this mechanism can be extracted from the following expression:

$$K_{lin-oil} = A_{ij} h_{lin-oil} \tag{23}$$

Nomenclature

m_c = mass of the cylinder charge

C_v = Specific Heat at constant volume

u = Specific Internal Energy

T = Mean Charge Temperature

p = Cylinder Pressure

V = Cylinder Volume

γ = The ratio of Specific Heats

dQ_{ht} = Charge-to-wall heat transfer

$\sum h_i m_i$ = The enthalpy flux across the system boundry

B_1 = Frequency Factor

ρ_{mix} = Density of Mixture

x_{fv} = Mass Fraction of Oxygen

T_z = Temperature of the zone

V_z = Volume of the zone

B_2 = A Constant

m_{fv} = Mass of fuel vapor

P_{ox} = Partial pressure of oxygen

P = Total Pressure

B = Cylinder bore

p = Cylinder Pressure

w = Average gas cylinder velocity

h_c = heat transfer coefficient

A= Interior surface area of engine volume

T_w = Interior surface temperature

M = Mass of working fluid

h = The film heat transfer coefficient

v_{pis} = Average piston speed

v_d = Displacement volume $\left(\frac{\pi B^2 S}{4}\right)$

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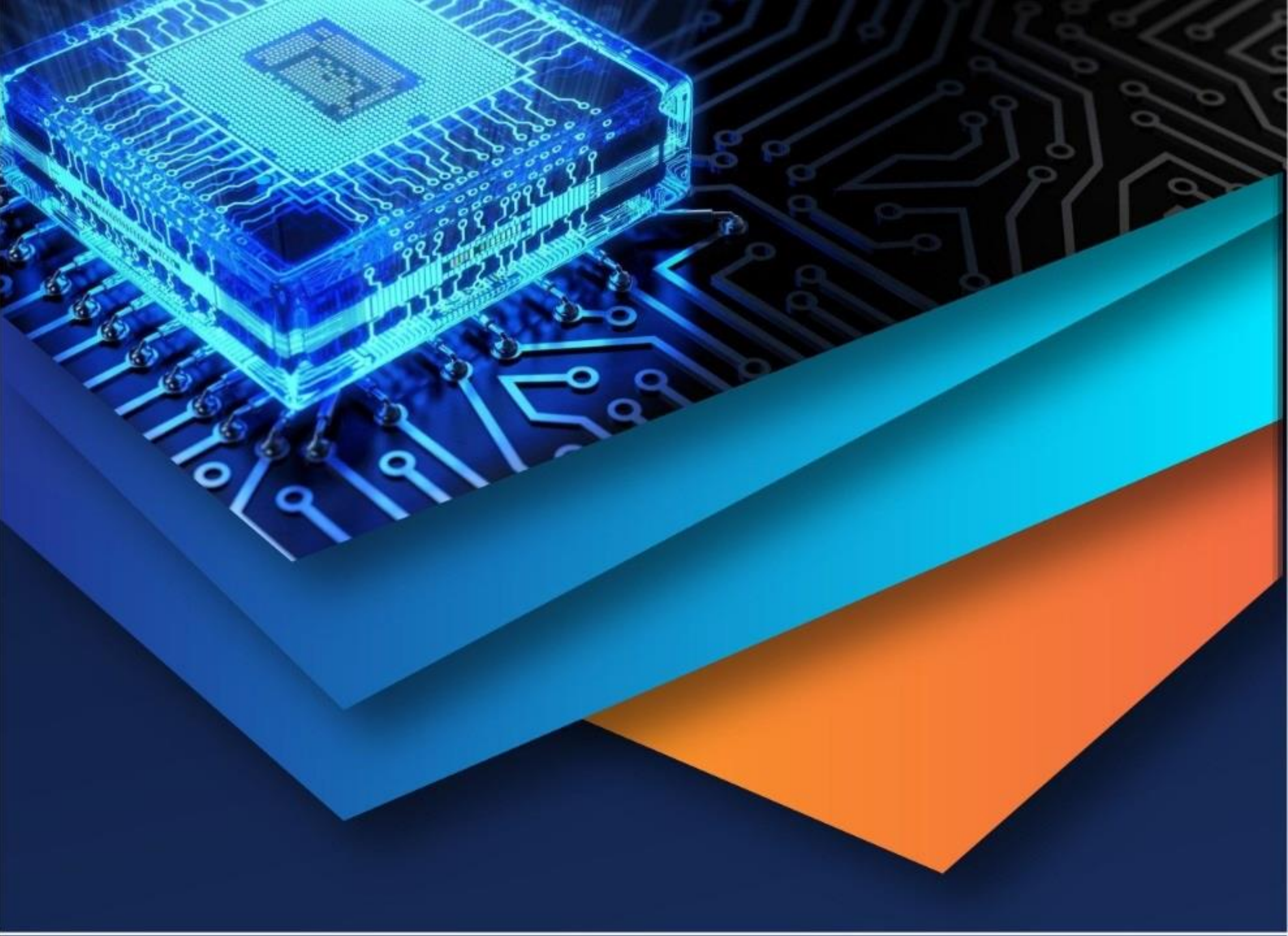

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