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Heat Transfer in Internal Combustion Engines

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ABSTRACT

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 (CO_2, H_20, 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.

NOMENCLATURE

A	area, m^2
a	parameter defined in Eq. 19
Cp	heat capacity, $J{\bullet}kg^{-1}{\bullet}K^{-1}$
D	diameter, m
Gr	Grashof number
H	heat transfer coefficient, $W{\bullet}m^{-2}. {\bullet}K^{-1}$
I	Planck function, $W{\bullet}m^{-1}$
k	thermal conductivity, $W{\bullet}m^{-1}{\bullet}K^{-1}$, or refractive index
K	absorption coefficient, m^{-1}
L	distance between cylinder head and piston top, m
M	mass flow rate, $kg{\bullet}s^{-1}$
n	refractive index
Pr	Prandl number
q	heat transfer rate, $W{\bullet}m^{-2}$
Q	total heat loss per cycle, $W{\bullet}m^{-2}$
R	radius, m
Re	Reynolds number
T	Temperature, K
U	overall heat transfer coefficient, $W{\bullet}m^{-2}{\bullet}K^{-1}$
V	axial velocity, $m{\bullet}s^{-1}$

Greek Symbols

α	gas absorptivity
δ	wall thickness, m
ε	emissivity
ρ	density, $kg{\bullet}m^{-3}$
ξ,τ	optical thickness
σ	Stefan-Boltzmann constant, $W{\bullet}m^{-2}{\bullet}K^{-4}$
μ	viscosity, Pa•s
ω	scattering albedo or rotating speed
θ	crank angle, degrees
Φ	equivalence ratio

Subscripts

b	bulk
c	convective or coolant-side
e	effective
g	gas stream or gas-side
h	hydraulic
i	inside or index
o	outside or wall surface
r	radiative
ν	spectral variable
v	valve
w	wall surface

INTRODUCTION

In internal combustion (IC) engines, heat loss from combustion gases through the cylinder wall to the coolant strongly influences the thermodynamics of the engine cycle. This heat loss is an important part of the energy balance, which influences gas temperature and pressure, piston work, engine performance, and emissions.

Ordinarily, the heat transfer from the cylinder gases to the walls is calculated by estimating the engine-wall temperature and basing the heat flux on the difference between the gas and wall temperatures. The heat transfer coefficient used for the calculation is usually adopted from empirical correlations (1-4). The weaknesses of this approach are as follows:

- The calculated results are sensitive to the wall temperature, which is treated as a given quantity. The empirical correlations include no means of incorporating significant changes in engine geometry or flow field.

- These correlations do not correctly predict the contribution of radiative heat transfer.

All of these weaknesses may be expected to have a strong effect on the heat transfer calculations. A more accurate means of calculating the heat flux is required for analysis of the IC engine's thermodynamic cycle. Also, more detailed information on thermal conditions (such as wall temperatures) would be useful to engineers and designers.

The objective of this paper is to provide a heat transfer model that can be used to calculate the heat flux more accurately. This greater accuracy is to be achieved by estimating the coolant temperature, rather than the wall temperature, because coolant temperature can be correctly measured (or estimated) much more easily than wall temperature can. A useful side benefit of this approach is that, because the wall temperatures are computed as an integral part of the calculations, their variation as a function of the crank angle is provided. This side benefit is particularly important for the adiabatic engine studies.

In contrast to common practice, the actual heat transfer mechanisms are not lumped together; instead, convection, radiation, and conduction are treated explicitly. The treatment of convective heat transfer accounts for the physical problems of rotating and impinging axial flow inside the combustion chamber and cross flow outside the cylinder, using empirical correlations established for flows similar in nature to those occurring on the inside and outside (coolant side) of the engine cylinder. The radiative heat transfer includes gas radiation (CO_2, H_2O, and CO) and soot-particle radiation. Conduction is treated in a conventional manner.

The wall temperatures, heat flux rates, and total heat loss are calculated. The results of the present analysis are compared with the existing predictions based on empirical correlations (1-3). The predicted radiative heat fluxes and gas emissivities from the present model will also be compared with experimental measurements (5).

HEAT TRANSFER MODEL

The physical system under consideration is quasi-steady, one-dimensional heat flow through the solid medium separating the cylinder gas and coolant. The basic assumptions used with the present model are as follows:

- The coolant temperature is known.

- The cylinder wall comprises seven heat transfer areas (intake valve, exhaust valve, cylinder head, liner, piston top, cup wall, and cup bottom), each of which is at its uniform temperature and has its uniform heat flux at every instant of the cycle. (See Fig. 1.)

- The convective heat transfer coefficients on both sides of the walls are uniform over each heat transfer area.

- The equivalent wall thickness at each heat transfer area is the same.

- The possible effects of surface scales or deposits on either side of the cylinder wall are not considered, and the heat losses through leaks at the valve seats or at the gap between the liner and the piston are neglected.

- The heat flow is quasi-steady, and the wall temperatures may be determined using simple network analysis.

The process by which heat is transferred from the cylinder gas through the wall to the coolant consists of three parts: convective and radiative heat transfer from cylinder gas to combustion-chamber surface, conductive heat transfer through the cylinder wall, and convective heat transfer from the cylinder wall to the coolant. Under the quasi-steady flow assumption, the heat flux is considered to be the same across each element:

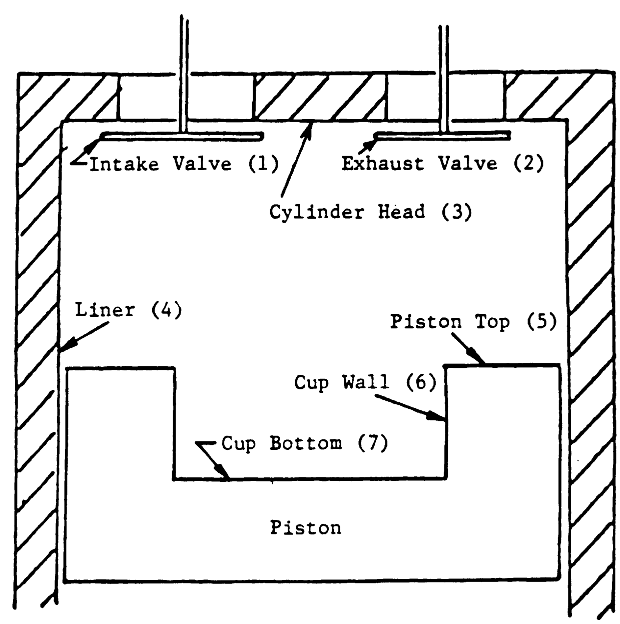

Fig. 1 Schematic of Engine Cylinder

$$\dot{q}/A = U\,(T_g - T_c) \tag{1}$$

$$= h_g\,(T_g - T_{wg}) \tag{2}$$

$$= k\,(T_{wg} - T_{wc})/\delta \tag{3}$$

$$= h_c\,(T_{wc} - T_c) \tag{4}$$

where h_g, the heat transfer coefficient at gas-side, includes both convection and radiation. The overall heat transfer coefficient, U, may be written in terms of thermal resistances corresponding to gas-side ($1/h_c$ and coolant-side ($1/h_g$) heat transfer processes:

$$\frac{1}{U} = \frac{1}{h_g} + \frac{\delta}{k} + \frac{1}{h_c} \tag{5}$$

The thermal resistance of the wall, δ/k, should be replaced by $R_i \bullet \ln(R_O/R_i)/k$ for the heat transfer at the liner, where R_i and R_O are the inner radius and outer radius of the liner, respectively; and L is a distance between the cylinder head and piston top. The gas-side and coolant-side wall temperatures (T_{wg} and T_{wc}, respectively) can be obtained iteratively from Eqs. 1-4 by satisfying Eq. 5. The instantaneous heat flux can be obtained by summing the heat fluxes through all heat transfer areas at each crank angle, θ.

$$\dot{q}(\theta) = \sum_i h_{gi}\,[T_g(\theta) - T_{wgi}(\theta)]\,A_i \tag{6}$$

where the index i denotes the number of heat transfer areas. The total heat loss per cycle from the combustion-chamber walls can be integrated according to the following equation:

$$\dot{Q} = \int_0^{720} \dot{q}(\theta)\,d\theta \tag{7}$$

CONVECTIVE HEAT TRANSFER

Heat Transfer from Cylinder Gas to Walls

A very fundamental rotating-disk flow and forced rotating-tube flow are considered for the calculation of heat transfer between the cylinder gas and the wall in swirl engines. For piston top and cylinder head, the turbulent flow over a disk rotating at constant speed about its axis is assumed. The analysis, based on Hartnett (6), is performed by integrating the continuity, momentum, and energy equations for the fully turbulent flow. The heat transfer coefficient, h_{gc} , over the disk plate is written as follows:

$$h_{gc}R/k_g = 0.0166\ \mathrm{Pr}\ \mathrm{Re}^{0.8} \tag{8}$$

where Re (= $\rho_g R^2\omega/\mu_g$), Pr, k_g and R are Reynolds number, Prandtl number, thermal conductivity of gas, and equivalent radius, respectively. For the liner and the piston-cup wall, the turbulent fluid flowing inside a pipe rotating about its longitudinal axis is assumed. The heat transfer coefficient is obtained from the results of Cannon and Kays (7):

$$h_{gc}/\rho_g CpV = 0.022\ \mathrm{Re}^{-0.2}\mathrm{Pr}^{-0.5}(T_g/T_{wg})^{-0.5} \tag{9}$$

where Re (= $\rho_g VD/\mu_g$) is a Reynolds number.

Heat Transfer from Cylinder Wall to Coolant

The heat transfer from the cylinder head and liner to the coolant is modeled by turbulent cross-flow forced convection. An average heat transfer over the cylindrical surface was correlated by McAdams for water and hydrocarbon oils (8).

$$h_c D_o/k_c = (0.35 + 0.56\ \mathrm{Re}^{0.52})\ \mathrm{Pr}_c^{\ 0.3} \tag{10}$$

where Pr_c, k_c, Re (= $\rho_c \overline{V}_c D_O/\mu_c$), and D_O are the Prandtl number of the coolant, thermal conductivity of the coolant, Reynolds number, and outside diameter of the engine cylinder, respectively. $\overline{V}_c$ is a mean velocity of the coolant.

The heat transfer from the back surface of the valve to the inlet gas is treated differently during the periods when the valves are open and closed. When the valve is open, the correlation of Kapadia and Borman is used for the heat transfer from the back surface of an open valve to the flowing air, based on steady-flow analysis and experiment (9):

$$h_c D_h/k_g = 1.012 \times 10^{-4}\ \mathrm{Re}^{1.27} \tag{11}$$

where Re (= $MD_h/\mu_g A_e$), M, D_h and A_e are Reynolds number, mass flow rate, hydraulic diameter, and effective flow area, respectively. When the valve is closed, the heat transfer is treated as natural convective heat transfer over a horizontal plate facing upward. The empirical correlations to be used are given in Eqs. 12 and 13, taken from Rohsenow and Hartnett (10):

$$h_c D_v/k_g = 0.54\ (\mathrm{Pr\ Gr})^{1/4} \quad \text{for } 10^4 \le \mathrm{Pr\ Gr} \le 10^9 \tag{12}$$

$$h_c D_v/k_g = 0.14\ (\mathrm{Pr\ Gr})^{1/4} \quad \text{for } \mathrm{Pr\ Gr} > 10^9 \tag{13}$$

where Gr and D_v are Grashof number and valve diameter, respectively.

The heat transfer from the piston wall to the lubricating oil depends on the movement of oil, the amount of oil in the cooling cavity, etc. Due to this complexity, the heat transfer coefficient in the oil side is assumed to be 2400 $W \bullet m^{-2} \bullet K^{-1}$, and the oil temperature is assumed to be 366 K (11, 12).

RADIATIVE HEAT TRANSFER

Radiative heat transfer in IC engines can be divided into two parts: the nonluminous gas radiation and the luminous solid-particle (soot) radiation. In spark-ignition (SI) engines, the radiation contributed by the soot particles can be neglected. In compression-ignition (CI) engines, the radiation is contributed mainly by the soot particles.

The radiative heat transfer, q_r, from a gas to its surroundings is given by

$$q_r = \sigma(\varepsilon_g T_g^4 - \alpha_{gw} T_{wg}^4) \quad (14)$$

where:

σ = Stefan-Boltzmann constant ($W \cdot m^{-2} \cdot K^{-4}$),

T_g, T_{wg} = gas and wall temperatures (K),

ε_g = gas emissivity, and

α_{gw} = gas absorptivity evaluated at wall temperature.

By solving the radiation transport equation, one can also express the radiative heat flux as follows (13):

$$q_r = \frac{4\pi}{3^{1/2}} \frac{\varepsilon_w}{2 - \varepsilon_w} \int_0^\infty d\nu \int_0^{\tau_0} K_\nu(\tau_0, \xi)\, a_\nu^2\, (I_{b\nu} - I_{w\nu})\, d\xi \quad (15)$$

where:

$$K_\nu(\tau,\xi) = \frac{[\exp(a_\nu\xi) + \exp(-a_\nu\xi)]}{2(1 - \alpha_\nu)a_\nu} [\alpha_\nu \exp(a_\nu\tau) + \exp(-a_\nu\tau)], \quad \tau > \xi \quad (16)$$

$$K_\nu(\tau,\xi) = \frac{[\alpha_\nu \exp(a_\nu\xi) + \exp(-a_\nu\xi)]}{2(1 - \alpha_\nu)a_\nu} [\exp(a_\nu\tau) + \exp(-a_\nu\tau)], \quad \tau < \xi \quad (17)$$

$$\alpha_\nu = \frac{(2 - \varepsilon_w)a_\nu - 3^{1/2}\varepsilon_w}{(2 - \varepsilon_w)a_\nu + 3^{1/2}\varepsilon_w} \exp(-2a_\nu\tau_0) \quad (18)$$

$$a_\nu = [3(1 - \omega_\nu)]^{1/2} \quad (19)$$

and where:

ε_w = wall emissivity,

$I_{b\nu}, I_{w\nu}$ = Planck functions at T_g and T_{wg},

ξ, τ, τ_0 = optical thickness and its value at the wall,

ν = spectral variable,

α_ν = parameter defined in Eq. 18,

a_ν = parameter defined in Eq. 19, and

ω_ν = scattering albedo.

Using the above equations, one can define total emissivity and absorptivity as follows:

$$\varepsilon_g = \frac{\frac{4\pi}{3^{1/2}} \frac{\varepsilon_w}{2 - \varepsilon_w} \int_0^\infty d\nu \int_0^{\tau_0} K_\nu(\tau_0,\xi) a_\nu^2\, I_{b\nu} d\xi}{\sigma T_g^4} \quad (20)$$

$$\alpha_{gw} = \frac{\frac{4\pi}{3^{1/2}} \frac{\varepsilon_w}{2 - \varepsilon_w} \int_0^\infty d\nu \int_0^{\tau_0} K_\nu(\tau_0,\xi) a_\nu^2\, I_{w\nu} d\xi}{\sigma T_{wg}^4} \quad (21)$$

Carbon dioxide, water vapor, and carbon monoxide are considered to be the principal participating combustion gases. There are five wave bands (15, 10.4, 9.4, 4.3, and 2.7 µm) with band centers at 667, 960, 1060, 2350, and 3715 cm^{-1} for CO_2; four wave bands (6.3, 2.7, 1.87, and 1.38 µm) with band centers at 1600, 3750, 5350, and 7250 cm^{-1} for H_20; and two wave bands (4.7 and 2.35 µm) with band centers at 2143 and 4260 cm^{-1} for CO. The spectral absorption coefficients for CO_2, H_20, and CO are evaluated from the exponential wideband model (14). It is difficult to measure or predict the concentration of soot particles in CI engines, so the soot-particle radiation is calculated using a mean absorption coefficient for the entire range of optical thicknesses (15).

$$K_\nu = 3.72\, C_0 T_g / C_2 \quad (22)$$

where

$$C_0 = \frac{36\pi nk}{(n^2 - k^2 + 2)^2 + 4n^2k^2}$$

and where C_2 is a second Planck's constant.

As a mean absorption coefficient for the entire spectrum, K_ν takes the average value of the Planck mean in the optically thin limit and the Rosseland mean in the optically thick limit. The accuracy is within 3% of the two limiting cases for the entire range of optical thicknesses.

With these radiation properties, and given the mean beam length, gas temperature, wall temperature, wall emissivity, and concentrations of CO_2, H_20, and CO, Eqs. 20 and 21 can be used to calculate the total gas emissivity and absorptivity. The resulting radiative heat flux can be calculated from Eq. 14.

RESULTS AND DISCUSSION

In order to calculate the heat transfer in a reciprocating IC engine, an engine cycle simulation code was incorporated with the present heat transfer model.

Figure 2 shows the heat transfer rates for the motoring case calculated by the present model and under the same engine geometry and comparable conditions, using the empirical correlations available in the literature. The present (model) results are in good agreement with those of the LeFeuvre model (3), and LeFeuvre's correlation is known

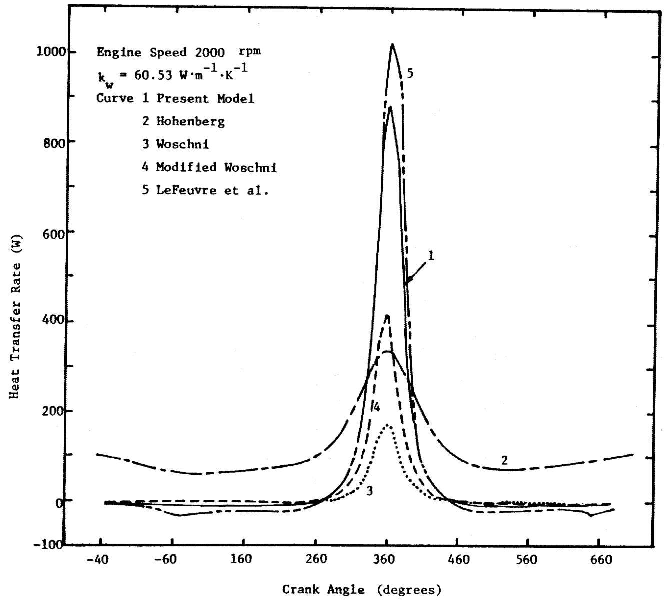

Fig. 2 Comparison of Heat Transfer Rates

to be good for the motoring case. LeFeuvre et al. applied Dorfman's rotational- and turbulent flow concepts (16) to correlate their instantaneous temperature and heat flux measurements with the analytical predictions. The present results and the results using Woschni's model (1) are in reasonable agreement for the intake and exhaust periods. Woschni's model required an average wall temperature for several cycles. Although his empirical equation was based on the turbulent flow in the duct, it required several constants that were determined by the engine geometry and operating conditions. The peak heat transfer rate of the present model is about five times higher than that of Woschni's model. This may be due to the omission of swirl, which is important in compression and expansion processes, in his equation. Woschni's equation was modified by Hiraki to consider the swirl of the gas inside the combustion chamber (17). The results are shown on curve 4 of Fig. 2, which displays a better fit to that of the present model. Hohenberg claimed that his heat transfer calculation (2) was precise due to his accurate surface temperature, combustion pressure, and heat flux measurements. His heat transfer correlation required the mean velocity, mean wall temperature, and mean gas temperature during the cycle calculation; because of this averaging process, his results could be expected to be higher at intake and exhaust processes and to be lower at compression and expansion processes than the results obtained using the present model; these trends are observable on curve 2 of Fig. 2. The majority of the above mentioned correlations, except LeFeuvre's, are not based on instantaneous data but on time-averaged data. Although LeFeuvre's correlation can predict the instantaneous heat flux, it is highly dependent on the engine geometry and operating conditions. The present model can also predict the instantaneous heat flux by using very physical fundamentals to calculate convection and a basic spectral absorption model to calculate radiation.

Figure 3 shows the histories of the gas and wall temperatures for the entire cycle. Note that $(T_{wg})_{ave}$ and $(T_g)_{ave}$ are the averaged wall and gas temperatures, respectively, over the entire cycle.

The calculated gas emissivities and radiative heat fluxes obtained using the present model and those obtained from the measurements by Flynn et al. (5) are compared in Table 1. Flynn et al. performed a detailed investigation of radiative heat flux in a diesel engine. They studied the radiative heat flux by varying a number of operating conditions, such as engine speed, fuel-to-air ratio, fuel-injection timing, fuel type, etc. The radiation intensity was measured at seven wavelengths by using an infrared detector and an infrared monochromator. Only those data obtained using No. 2 diesel oil were compared. In order to compare the present calculated radiation results with those of Flynn et al., two important parameters mean beam length and concentrations of gas species are required. The value of the mean beam length was taken as 3.6 times the ratio of cylinder gas volume to its surface area based on Flynn's test engine at top dead center (TDC). The concentrations of the gas species and the in-cylinder pressures were obtained from the engine cycle simulation code, based on the values of peak gas temperatures and the equivalence ratios from Flynn et al. The reported maximum wall temperatures (5), which were assumed to occur at the same time (or crank angle) as the peak gas temperatures for the present calculations, were also used for the present comparison. The calculated and measured gas emissivities were in reasonable agreement (within 10%). The heat fluxes calculated by using the present model are also in reasonable agreement with the measured values, except for run no. 54 at an engine speed of 1010 rpm.

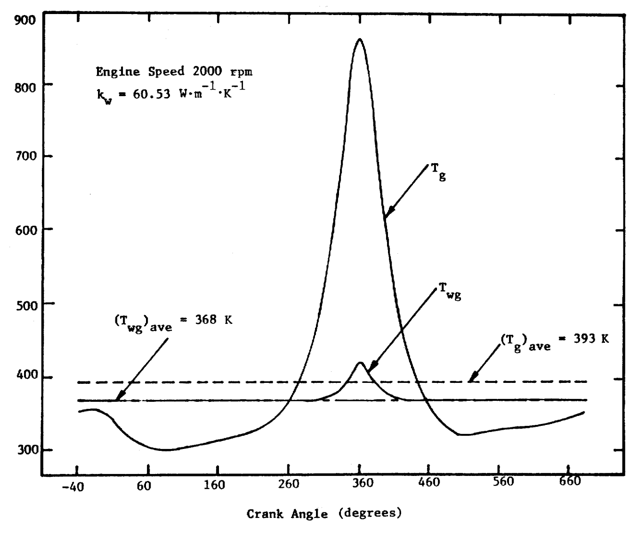

Fig. 3 Wall- and Gas-Temperature Histories

Table I Gas Emissivity and Heat Flux Results Obtained from the Present Model and from the Measurement

Run No.	Gas Temp T_g(K)	Equivalence Ratio, ϕ	Engine Speed (rpm)	Gas Emissivity ϵ_g^a	Gas Emissivity ϵ_g^b	$\frac{\epsilon_g^b - \epsilon_g^a}{\epsilon_g^a}$ x 100%	Heat Flux 10^6 W/m^2 $\dot{Q}''^a$	Heat Flux 10^6 W/m^2 $\dot{Q}''^b$	$\frac{\dot{Q}''^b - \dot{Q}''^a}{\dot{Q}''^a}$ x 100%
20	2200	0.514	2000	0.925	0.8325	-10.0	1.224	2.148	-6.2
27	2200	0.230	2010	0.920	0.8325	-9.5	1.234	1.148	-7.0
54	2290	0.398	1010	0.760	0.8302	9.2	1.065	1.296	21.7
62	2290	0.469	1505	0.850	0.8302	-2.3	1.308	1.296	-0.9
70	2187	0.463	2490	0.840	0.8336	-0.8	1.056	1.078	2.1
77	1944	0.749	2000	0.870	0.8388	-3.6	0.677	0.676	-0.1
84	2262	0.459	1995	0.813	0.8312	2.3	1.156	1.231	6.5
91	2273	0.469	2005	0.876	0.8309	-5.1	1.314	1.254	-4.5
98	2257	0.438	1995	0.875	0.8314	-5.0	1.285	1.220	-5.1
111	2294	0.455	2005	0.880	0.8302	-5.7	1.341	1.296	-3.3

[a] From the Ref. 5 measurement.
[b] From the present model.

The present model can be used to analyze the thermal effects in IC engines, such as wall cooling, material selection, etc. Figure 4 indicates that the material of the wall is one of the major parameters affecting the heat loss from the engine. Figure 5 shows that the effect of swirl on heat transfer rate is significant during the compression and expansion periods, particularly at TDC.

CONCLUSIONS

A heat transfer model that uses quasi-steady heat flux relations to calculate the heat transfer from combustion gases to the coolant in IC engines has been developed. 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 (CO_2 , H_2O, and CO) and soot-particle radiation. The wall temperatures can be accurately predicted from this model for both the gas and coolant sides of the combustion-chamber walls.

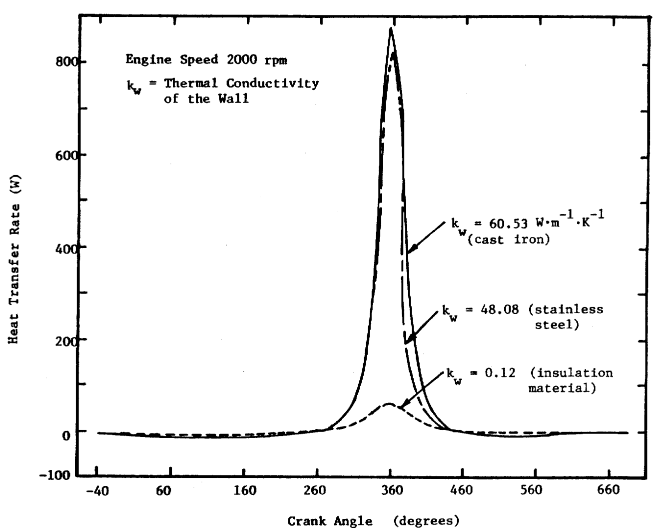

Fig. 4 Effect of Wall's Thermal Conductivity on Heat Transfer Rate

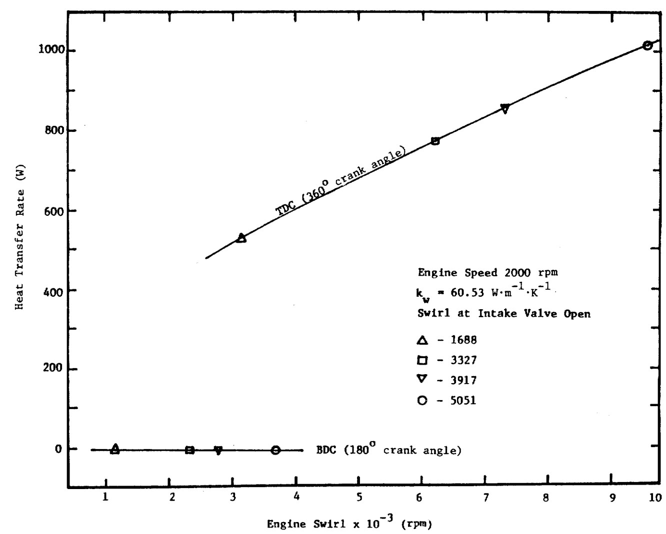

Fig. 5 Effect of Swirl on Heat Transfer Rate

The heat transfer results of the present model for the motoring case are in good agreement with those of LeFeuvre's empirical correlation, which was obtained from instantaneous heat flux data. The calculated radiative heat flux and gas emissivity show reasonable agreement with those based on Flynn's measurements.

The present model is a step toward analysis of the thermal effects in engines and of cooling systems. The model can be used to study the effects of a number of parameters on heat transfer and wall temperatures for example, the effects of in-cylinder flows and of wall thickness and wall materials.

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