Performance of a spark ignition engine under the effect of friction using a gas mixture model

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This paper presents the effect of friction on the performance of a spark ignition engine using a gas mixture as the working fluid. The results were compared to a frictionless engine. Engine parameters that were studied include equivalence ratio, engine speed, break mean effective pressure (BMEP), and cycle thermal efficiency. It was found for the frictionless engine operating at 6000 rev min^{-1} and stoichiometric air–fuel mixture that BMEP and efficiency were about 14 bar and 36% respectively. On the other hand, when friction is included under the same condition BMEP and efficiency were about 10 bar and 27% respectively. However, at lower engine speed and equivalence ratio, the deviations were much smaller. Therefore, it is more realistic to consider the effect of friction using the gas mixture model instead of air as the working fluid for the analysis of spark ignition engines, especially when running at high speeds and/or rich mixtures.

Keywords: SI engine, Gas mixture, Temperature dependent specific heats, Friction

List of symbols

- a constant in Weibe function
- a_s number of moles of air at stoichiometric condition, dimensionless
- A heat transfer area, m^2
- AF air/fuel ratio, dimensionless
- AF_s air/fuel ratio for stoichiometric condition, dimensionless
- A_t coefficient in oil film thickness equation, m
- B coefficient in oil film thickness equation, m
- C skirt clearance, m
- C_p constant pressure specific heat, kJ kg^{-1} K^{-1}
- C_v constant volume specific heat, kJ kg^{-1} K^{-1}
- D cylinder diameter, m
- $fmep$ friction mean effective pressure, bar
- h_{cg} heat transfer coefficient for gases in the cylinder, W m^{-2} K^{-1}
- k specific heat ratio, dimensionless
- ℓ connecting rod length, m
- L_{ring} ring thickness
- L_{skirt} skirt length
- LHV lower heating value, kJ kg^{-1}
- m mass of cylinder contents, kg
- m_f mass of burned fuel, kg
- M molar mass
- P pressure inside cylinder, Pa
- P_i inlet pressure, Pa
- Q heat transfer, kJ
- Q_{in} heat added from burning fuel, kJ
- R crank radius, m
- R_g gas constant, kJ kg^{-1} K^{-1}
- R_{mix} gas constant for the mixture, kJ kg^{-1} K^{-1}
- R_u universal gas constant=8314·5 J $kmol^{-1}$ K^{-1}
- T gas temperature in the cylinder, K
- T_i inlet temperature, K
- T_w cylinder temperature, K
- U internal energy, kJ
- U_p piston speed, m s^{-1}
- $\overline{U_p}$ average piston speed, m s^{-1}
- V cylinder volume, m^3
- V_c clearance volume, m^3
- V_d displacement volume, m^3
- w average cylinder gas velocity, m s^{-1}
- W_i indicated work, J
- W_b brake work, J
- $W_{friction}$ friction work, J
- W_{irrev} irreversible work, J
- x mass fraction
- x_b burning rate of the fuel, dimensionless
- X distance from top dead centre, m
- y mole fraction
- α number of carbon atoms in the fuel
- β number of hydrogen atoms in the fuel
- ε oil film thickness, m
- θ angle, °
- θ_s start of combustion or heat addition, °
- $\Delta\theta$ duration of combustion, °
- μ oil dynamic viscosity, N s m^{-12}
- η_{th} thermal efficiency, %
- Φ equivalence ratio

Subscripts

- irrev irreversible
- mix mixture

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Introduction

Many studies on air standard power cycles neglect friction for simplicity of the engine analysis.[1–4] However, the high revolution of the engine makes the design of this assumption less reasonable where a large percentage of engine power is converted into friction. It is a fact that air standard power cycle analysis gives only approximation to the actual conditions and outputs;[5] consequently, it would be very useful to embrace the effect of friction. Additionally, in most air standard power cycles air is generally assumed to be the working fluid and the details of the mixture of gases encountered in the combustion chamber are neglected. Thus, it would be very important to include the effects of both friction and gas mixture on the performance of the engine to have a better insight in the engine chamber and to get a better approximation to reality.

It is well known that the engine's efficiency increases by reducing its mechanical friction. There are many studies that have modelled the performances of Otto, Diesel and dual cycles including friction.[6–8] However, they dealt with average values of piston coefficient of friction and neglected type of lubricants and engine configurations details such as engine skirt and rings. It is understood that a large percentage of the mechanical friction loss in the engine occur on the lubricated surfaces between the skirt and the cylinder liner as well as between the cylinder rings and cylinder liner.[9] The lubrication between rings and cylinder liner is effected by oil viscosity, oil film thickness, piston ring configuration, and the operational specifications of the engine. Besides, the friction between the piston skirt and the cylinder liner is affected by the clearance, piston tilt, piston skirt design, and surface roughness. By considering energy consumption within the engine, it is found that friction loss contributes the major portion of the energy consumption developed in an engine. About two-thirds of it is caused by piston skirt friction, piston rings, and bearings, and the other third is due to the valve train, crankshaft, transmission, and gears.[10] Similar figures were reported by Kim *et al.*,[11] who were successful in reducing engine friction through liner rotation. Piston ring lubrication models were used for prediction of engine cylinder friction. An accurate representation of lubrication conditions at the piston ring-cylinder liner interface is required for the estimation of frictional losses. Examples of the work conducted on modelling and analysis of the frictional losses at the piston ring cylinder liner contact are given in Refs. 12–17. Moreover, Xu *et al.*[18] presented two theoretical models that predict friction for piston ring and cylinder liner. Their work was compared to experimental findings. Also, Xu *et al.* proposed an inter-ring gas flow model by considering the effect of orifice flow through the ring end gap and ring side clearance.[19]

In a recent study it was reported that piston ring friction force was higher than previously predicted. Also it was pointed that fuel economy improvements exceeding 4% may result from combined application of reducing lubricant viscosity and proper surface treatment.[20] Their work was based on numerical findings using four different simulation methodologies, such as Ringpak. Other studies have used Ringpak tool in their engine friction simulation.[21,22] Other models are also available in literature.[23,24]

In most models of air standard power cycles, the air–fuel mixture and combustion products are approximated as ideal gases. In such cases air was assumed as the working fluid with constant specific heats without taking into consideration temperature dependence of the specific heats of the working fluid.[25–29] However, due to the high rise in combustion temperature this assumption becomes less realistic. In a previous study a more realistic approach on the behaviour of variable specific heats was implemented on the performance evaluation of the spark ignition (SI) engine.[30] Then, the effect of piston friction on the performance of SI engine using new thermodynamic approach was published.[31]

Also, in another study the working fluid was modelled as a gas mixture with temperature dependent specific heats.[32] Similarly, the present work assumes the working fluid inside the combustion chamber of an SI engine as a gas mixture with temperature dependent specific heat, and with the inclusion of the effect of friction.

The objective of this paper is to include the effects of both friction and gas mixture model to simulate the thermodynamic performance of the SI engine. This allows a better understanding and insight in the engine chamber, and it will obtain a better approximation to more realistic processes.

Thermodynamic properties of air–fuel mixture and combustion products

The variation of specific heats of air for the temperature range 300–3500 K is found in literature.[33] It is based on the assumption that air is an ideal gas mixture containing 78·1% nitrogen, 20·95% oxygen, 0·92% argon, and 0·03% carbon dioxide (on mole basis). For example, constant pressure specific heat of air is presented in the following equation

$$\begin{aligned} C_p = {} & 2{\cdot}506\times10^{-11}T_g^2 + 1{\cdot}454\times10^{-7}T_g^{1{\cdot}5} - \\ & 4{\cdot}246\times10^{-7}T_g + 3{\cdot}162\times10^{-5}T_g^{0{\cdot}5} + 1{\cdot}3303 - \\ & 1{\cdot}512\times10^{4}T_g^{-1{\cdot}5} + 3{\cdot}063\times10^{5}T_g^{-2} - \\ & 2{\cdot}212\times10^{7}T_g^{-3} \end{aligned} \tag{1}$$

It is well known that in actual SI engines the combustion products have temperature dependent specific heats, also. The most common combustion products are CO_2, CO, H_2O, N_2, O_2, and H_2. The specific heats of these species have different dependence on temperature. Some species specific heats are strongly dependent on temperature others are less dependent. Thus, it should be more accurate to calculate the specific heat of the mixture as a summation of individual species specific heats rather than taking a rough estimation that the whole mixture behaves as air. In the present work, the following species are assumed as the combustion products: CO_2, CO, H_2O, N_2, O_2, and H_2. The temperature dependent specific heat for these combustion product species takes the general form[9]

$$\frac{c_p}{R_g} = a_1 + a_2T + a_3T^2 + a_4T^3 + a_5T^4 \tag{2}$$

The constant a_1–a_5 for all combustion species are given in Tables 1 and 2.[9] Also, the specific heat of the fuel is assumed to be temperature dependent and it takes the following form[34]

Table 1 Coefficients for species temperature dependent specific heats ($T \leqslant 1000$)

Species	a_1	a_2	a_3	a_4	a_5
CO_2	2·400779	$0·8735096\times10^{-2}$	$-0·6660708\times10^{-5}$	$0·2002186\times10^{-8}$	$0·632740\times10^{-15}$
H_2O	4·0701275	$-0·1108450\times10^{-2}$	$0·4152118\times10^{-5}$	$-0·296374\times10^{-8}$	$0·807021\times10^{-12}$
N_2	3·6748261	$-0·1208150\times10^{-2}$	$0·2324010\times10^{-5}$	$-0·6321756\times10^{-9}$	$-0·225773\times10^{-12}$
O_2	3·6255985	$-0·1878218\times10^{-2}$	$0·7055454\times10^{-5}$	$-0·6763513\times10^{-8}$	$0·215560\times10^{-11}$
CO	3·7100928	$-0·1619096\times10^{-2}$	$0·3692359\times10^{-5}$	$-0·2031967\times10^{-8}$	$0·239533\times10^{-12}$
H_2	3·0574451	$0·267652\times10^{-2}$	$-0·5809916\times10^{-5}$	$0·5521039\times10^{-8}$	$-0·181227\times10^{-11}$

$$\widetilde{C}_p = -0·55313 + 181·62\left(\frac{T}{1000}\right) - 97·787\left(\frac{T}{1000}\right)^2 + 20·402\left(\frac{T}{1000}\right)^3 - 0·03095\left(\frac{T}{1000}\right)^{-2} \quad (3)$$

where $\widetilde{C}_p$ has the unit of cal gmol^{-1} K^{-1}.

The gas constant for the mixture is calculated as follows

$$R_{mix} = \frac{R_u}{M_{mix}} \quad (4)$$

The molar mass of the mixture is determined as

$$M_{mix} = \sum_{i=1}^{n} y_i M_i \quad (5)$$

Before combustion is taking place, the mixture is considered as a combination of fuel vapour and air. Therefore, the molecular weight of mixture is written as

$$M_{mix} = y_a M_a + y_f M_f \quad (6)$$

The mole and the mass fractions for the fuel are given respectively, as

$$y_f = \frac{1}{1 + 4·76 a_s/\Phi} \quad (7)$$

$$x_f = \frac{1}{1 + AF_s/\Phi} \quad (8)$$

where Φ is the fuel air equivalence ratio and is given as

$$\Phi = \frac{AF_s}{AF} \quad (9)$$

a_s is the stoichiometric number of moles for the air and AF_s is the stoichiometric air/fuel ratio. The mole fraction and the mass fraction for the air are obtained respectively

$$y_a = 1 - y_f \quad (10)$$

$$x_a = 1 - x_f \quad (11)$$

Thus, the specific heat for the air–fuel mixture can be computed as

$$C_{p_{mix}} = C_{p_a} x_a + C_{p_f} x_f \quad (12)$$

On the other hand, the specific heat for the combustion products is calculated as

$$C_{p_{mix}} = \sum_{i=1}^{n} C_{p_i} x_i \quad (13)$$

where i goes for CO_2, CO, H_2O, N_2, O_2, and H_2. The mass fraction x_i is given as

$$x_i = \frac{n_i M_i}{m_{mix}} \quad (14)$$

where m_{mix} is the total mass of the mixture given as

$$m_{mix} = \sum_{i=1}^{n} n_i M_i \quad (15)$$

During combustion a flame front is assumed to travel throughout the combustion chamber. The gases ahead of this flame are assumed to have the air–fuel mixture properties whereas the gases behind it take the properties of the combustion products. Thus, it is very reasonable to estimate the specific heat for the mixture as follows

$$C_{p\,mix} = C_{p\,airfuel}(1 - x_b) + C_{p\,products}(x_b) \quad (16)$$

where x_b is evaluated from the Weibe function and represents the burn fraction of the mixture.

Finally, the specific heat ratio is calculated as

$$k = \frac{C_{p\,mix}}{C_{v\,mix}} = \frac{C_{p\,mix}}{C_{p\,mix} - R_{mix}} \quad (17)$$

Combustion reactions

By considering the existence of only six species (CO_2, H_2O, N_2, O_2, CO, and H_2), in the combustion products, the chemical reaction for burning one mole of hydrocarbon fuel is written as

$$C_\alpha H_\beta + \frac{a_s}{\Phi}(O_2 + 3·76N_2) \rightarrow n_1 CO_2 + n_2 H_2O + n_3 N_2 + n_4 O_2 + n_5 CO + n_6 H_2 \quad (18)$$

Table 2 Coefficients for species temperature dependent specific heats ($1000 < T < 3200$ K)

Species	a_1	a_2	a_3	a_4	a_5
CO_2	4·460800	$0·3098170\times10^{-2}$	$-0·1239250\times10^{-5}$	$0·2274130\times10^{-9}$	$-0·155259\times10^{-13}$
H_2O	2·7167600	$0·294513\times10^{-2}$	$-0·802243\times10^{-6}$	$0·102266\times10^{-9}$	$-0·484721\times10^{-14}$
N_2	2·89631	$0·151548\times10^{-2}$	$-0·572352\times10^{-6}$	$0·998073\times10^{-10}$	$-0·652235\times10^{-14}$
O_2	3·62195	$0·736182\times10^{-3}$	$-0·196522\times10^{-6}$	$0·362015\times10^{-10}$	$-0·289456\times10^{-14}$
CO	2·98406	$0·148913\times10^{-2}$	$-0·578996\times10^{-6}$	$0·103645\times10^{-9}$	$-0·693535\times10^{-14}$
H_2	3·100190	$0·511194\times10^{-3}$	$0·526442\times10^{-7}$	$-0·349099\times10^{-10}$	$0·369453\times10^{-14}$

This chemical reaction is applicable for lean, stoichiometric, or rich mixtures. For $\Phi \leqslant 1$ (stoichiometric and lean mixtures), the numbers of moles of the combustion products are given as

$$n_1 = \alpha; \quad n_2 = \frac{\beta}{2}; \quad n_3 = 3{\cdot}76\frac{a_s}{\Phi};$$
$$n_4 = a_s\left(\frac{1}{\Phi} - 1\right); \quad n_5 = 0; \quad n_6 = 0 \tag{19}$$

However, for $\Phi > 1$ (rich mixture), it is assumed that the there is no O_2 in the combustion products. Thus, the numbers of moles for the combustion products are given as

$$n_1 = \alpha - n_5; \quad n_2 = \frac{\beta}{2} - d_1 + n_5; \quad n_3 = 3{\cdot}76\frac{a_s}{\Phi};$$
$$n_4 = 0; \quad n_5 = n_5; \quad n_6 = d_1 - n_5 \tag{20}$$

where $n_5 = \left[-b_1 + \left(b_1^2 - 4a_1c_1\right)^{1/2}\right]/2a_1$

$$a_1 = 1 - K \tag{21}$$

$$b_1 = \frac{\beta}{2} + K\alpha - d_1(1 - K) \tag{22}$$

$$c_1 = -\alpha d_1 K \tag{23}$$

$$d_1 = 2a_s\left(1 - \frac{1}{\Phi}\right) \tag{24}$$

The equilibrium constant k in equation (17) is a curve fit of JANAF table for the temperature range $400 < T < 3200$ K.[9]

$$k = \exp\left(2{\cdot}743 - \frac{1{\cdot}761}{(T/1000)} - \frac{1{\cdot}611}{(T/1000)^2} + \frac{0{\cdot}2803}{(T/1000)^3}\right) \tag{25}$$

In deriving the previous relations for rich mixture, an equilibrium reaction is assumed to take place between the species CO_2, H_2O, CO and H_2. The equilibrium reaction is given as[9]

$$CO_2 + H_2 \rightleftharpoons CO + H_2O \tag{26}$$

Thermodynamic analysis

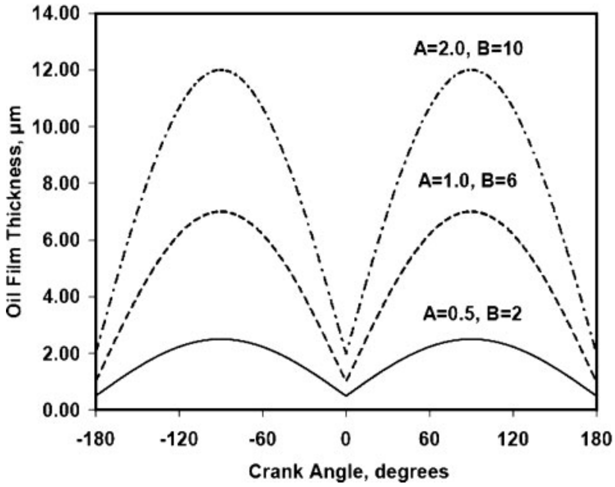
By considering the mixture inside the combustion chamber as a closed system then for small change of the process, the first law of thermodynamics is simply written as

$$\delta Q - \delta W = \mathrm{d}U \tag{27}$$

Therefore, by using the definition of work, the first law can be expressed as

$$\delta Q_{\mathrm{in}} - \delta Q_{\mathrm{loss}} - (P\mathrm{d}V - \delta W_{\mathrm{irrev}}) = \mathrm{d}U \tag{28}$$

where the irreversible work is mainly due to friction occurring inside the combustion chamber which is the piston friction work. The piston friction work consists of two major parts which are the skirt friction and pressure rings friction. By using the Newton's law of viscosity and friction work is defined as



1 Ring oil film thickness at various locations ranging from BDC to TDC

$$\delta W_{\mathrm{irrev}} = \left(\mu\frac{\mathrm{d}u}{\mathrm{d}y}\right)_{\mathrm{skirt}} L_{\mathrm{skirt}}\pi D\Delta x + \left(\mu\frac{\mathrm{d}u}{\mathrm{d}y}\right)_{\mathrm{ring}} L_{\mathrm{ring}}\pi D\Delta x \tag{29}$$

The friction work (irreversible work) can be expressed in terms of instantaneous piston speed as

$$\delta W_{\mathrm{irrev}} = \mu\frac{U_p(\theta)}{C} L_{\mathrm{skirt}}\pi D\Delta x + \mu\frac{U_p(\theta)}{\varepsilon} L_{\mathrm{ring}}\pi D\Delta x \tag{30}$$

where C is the skirt clearance and ε is the clearance between the liner and the pressure ring. In the present study, the value of C is taken as a constant to represent the average clearance between the cylinder liner and the skirt of the piston. However, ε is taken as the oil film thickness between the ring and cylinder liner. This thickness reaches a minimum value at the bottom dead centre (BDC) and top dead centre (TDC) and has higher values between them.[35] The distribution of oil film thickness with crank angle was reported.[36] The shape of the oil film thickness can be approximated by a trigonometric function where minimum values at BDC and TDC and higher values in between. In the current study the following distribution is assumed that agrees with the trend of previous published work

$$\varepsilon(\theta) = A_t + B|\sin(\theta)| \tag{31}$$

where A_t and B are constants. Figure 1 gives a representation of three different distributions of ring oil film thicknesses obtained by using different combinations of A_t and B. These values are chosen to agree with distributions reported in literature.[35–37]

The instantaneous piston speed, in equation (30), can be expressed in terms of average piston speed as[9]

$$U_p(\theta) = \overline{U_p}\frac{\pi}{2}\sin(\theta)\left(1 + \frac{\cos(\theta)}{\left[(\ell/R) - \sin^2(\theta)\right]^{1/2}}\right) \tag{32}$$

Thus, the friction work in equation (30) is written as

$$\delta W_{\mathrm{irrev}} = \mu\pi D\Delta x\overline{U_p}\frac{\pi}{2}\sin(\theta)\left(1 + \frac{\cos(\theta)}{\left[(\ell/R) - \sin^2(\theta)\right]^{1/2}}\right)\left(\frac{L_{\mathrm{skirt}}}{C} + \frac{L_{\mathrm{ring}}}{\varepsilon}\right) \tag{33}$$

For an ideal gas the equation of state is expressed as

$$PV = mR_g T_g \tag{34}$$

By differentiating equation (34), we can get

$$P\mathrm{d}V + V\mathrm{d}P = mR_g \mathrm{d}T_g \tag{35}$$

Also, for an ideal gas with constant specific heats the change in internal energy is expressed as

$$\mathrm{d}U = mC_v \mathrm{d}T_g \tag{36}$$

By substituting equation (36) into (35) then

$$\mathrm{d}U = \frac{C_v}{R_g}(P\mathrm{d}V + V\mathrm{d}P) \tag{37}$$

By substituting equation (37) into (28), the following equation is obtained

$$\delta Q_{\mathrm{in}} - \delta Q_{\mathrm{loss}} - (P\mathrm{d}V - \delta W_{\mathrm{irrev}}) = \frac{C_v}{R_g}(P\mathrm{d}V + V\mathrm{d}P) \tag{38}$$

The total amount of heat input to the cylinder by combustion of fuel in one cycle is

$$Q_{\mathrm{in}} = m_f LHV \tag{39}$$

The total heat added from the fuel to the system until the crank position reaches angle θ is given as

$$Q(\theta) = Q_{\mathrm{in}} x_b \tag{40}$$

where x_b is the Weibe function that is used to determine the combustion rate of the fuel and is expressed as[5]

$$x_b = 1 - \exp\left[-5\left(\frac{\theta - \theta_s}{\Delta\theta}\right)^3\right] \tag{41}$$

The total amount of heat loss from the system when the crank moves an increment of $\mathrm{d}\theta$

$$Q_{\mathrm{loss}} = \frac{h_{cg} A_h}{\omega}(T_g - T_w)\delta\theta \tag{42}$$

By substituting equations (39)–(42) into equation (38) followed by differentiation with respect to crank angle the following equation is obtained

$$\frac{\mathrm{d}P}{\mathrm{d}\theta} = \frac{k-1}{V}\left[Q_{\mathrm{in}}\frac{\mathrm{d}x_b}{\mathrm{d}\theta} - \frac{h_{cg}A_h}{\omega}(T_g - T_w)\frac{\pi}{180}\right] - k\frac{P}{V}\frac{\mathrm{d}V}{\mathrm{d}\theta} + \frac{k-1}{V}(\delta W_{\mathrm{irrev}})\frac{\mathrm{d}x}{\mathrm{d}\theta} \tag{43}$$

Equation (43) can be solved by using explicit finite difference technique with second order accurate differentiation. The result is given as

$$\begin{aligned} P(\theta) = {} & \frac{4}{3}P(\theta - \Delta\theta) - \frac{1}{3}P(\theta - 2\Delta\theta) + \\ & \frac{k-1}{3V}Q_{\mathrm{in}}[3x_b(\theta) - 4x_b(\theta - \Delta\theta) + x_b(\theta - 2\Delta\theta)] \\ & + \frac{2}{3}\frac{(k-1)}{3V}\left[h_{cg}A_h(\theta)(T_g - T_w)\right]\frac{1}{\omega} - \\ & \frac{2kP(\theta - \Delta\theta)}{3V(\theta)}\left[\frac{V(\theta + \Delta\theta) - V(\theta + \Delta\theta)}{2\Delta\theta}\right] \\ & + \frac{2}{3}\frac{(k-1)}{3V}\delta W_{\mathrm{irrev}}\frac{\mathrm{d}x}{\mathrm{d}\theta}\Delta\theta \end{aligned} \tag{44}$$

where $\mathrm{d}P/\mathrm{d}\theta$ is expressed as

$$\frac{\mathrm{d}P}{\mathrm{d}\theta} = \frac{3P(\theta) - 4P(\theta - \Delta\theta) + P(\theta - 2\Delta\theta)}{2\Delta\theta} \tag{45}$$

The instantaneous cylinder volume, area and displacement are given by the slider crank model as[5]

$$V(\theta) = V_c + \frac{\pi D^2}{4}x(\theta) \tag{46}$$

$$A_h(\theta) = \frac{\pi D^2}{4} + \frac{\pi DS}{2}\left\{R + 1 - \cos(\theta) + \left[R^2 - \sin^2(\theta)\right]^{1/2}\right\} \tag{47}$$

$$x(\theta) = (\ell + R) - \left\{R\cos(\theta) + \left[\ell^2 - \sin^2(\theta)\right]^{1/2}\right\} \tag{48}$$

Once the pressure is calculated, the temperature of the gases in the cylinder can be calculated using the equation of state as

$$T_g = \frac{P(\theta)V(\theta)}{mR_g} \tag{49}$$

The convective heat transfer coefficient in equation (43) h_{cg} is given by the Woschni model as[9,38]

$$h_{cg} = 3{\cdot}26D^{-0{\cdot}2}P^{0{\cdot}8}T_g^{-0{\cdot}55}w^{0{\cdot}8} \tag{50}$$

The velocity of the burned gas and is given as

$$w(\theta) = 2{\cdot}28\overline{U_p} + C_1\frac{V_d T_{gr}}{P_r V_r}[P(\theta) - P_m] \tag{51}$$

In the above equation, the displacement volume is V_d. However, V_r, T_{gr} and P_r are reference state properties at closing of inlet valve and P_m is the pressure at same position to obtain P without combustion (pressure values in cranking). The value of C_1 is given as:

(i) for compression process: $C_1 = 0$
(ii) for combustion and expansion processes: $C_1 = 0{\cdot}00324$.

By which the average piston speed is calculated from

$$\overline{U}_p = \frac{2NS}{60} \tag{52}$$

Having calculated the needed thermodynamic properties (P and T), the indicated power can be calculated by the integration of the PV diagram as

$$W_i = \int p\,\mathrm{d}V \tag{53}$$

Then, the brake power is calculated as the difference between the indicated power and the friction power. To estimate the total friction power one need to include not only the piston friction but to include all the constituents of friction in SI engine. The total engine friction is due to mechanical friction, accessory friction, and pumping friction, and piston friction. The mechanical friction is due to other moving parts such as crankshafts, and valve trains. The pumping represents the work done during intake and exhaust strokes. However, the accessories include oil pumps, fuel pump and fan.[9] It is worth mentioning that in the prescribed fist law model, for the mixture inside the combustion chamber, the piston friction was only included in the formulation and other friction parts are absent in the model. This resembles what happens in the combustion chamber. The mixture at any crank angle is affected by the friction that occurs between the system (the mixture) and the surrounding (the piston and the chamber walls). Thus, the friction work from the surrounding on the system during a process is only due to piston friction work and nothing is related to other parts of friction. Thus, it is expected, as will be demonstrated in the results, that piston friction will affect the pressure and temperature distribution inside the combustion chamber and this is realistic from a thermodynamics point of view. The effect of other constituents of friction is to reduce the power output from the engine because part of this indicated power is consumed by these friction parts.

Empirical formulas for friction mean effective pressure for all mentioned friction constituents for SI engines having a displacement volume, between 845 and 2000 as a function of engine speed, is given by[39]

$$fmep\ (\text{bar}) = \left[0{\cdot}97 + 0{\cdot}15(N/1000) + 0{\cdot}05(N/1000)^2\right] \quad (54)$$

Thus, the total friction work is given as

$$W_{\text{friction}} = V_{\text{d}}\left[0{\cdot}97 + 0{\cdot}15(N/1000) + 0{\cdot}05(N/1000)^2\right] \quad (55)$$

Thus, the brake work is given as

$$W_{\text{b}} = \int p\,\mathrm{d}V - V_{\text{d}}\left[0{\cdot}97 + 0{\cdot}15(N/1000) + 0{\cdot}05(N/1000)^2\right] \quad (56)$$

The thermal efficiency is calculated as

$$\eta_{\text{th}} = \frac{W_{\text{b}}}{m_{\text{f}}LHV} \quad (57)$$

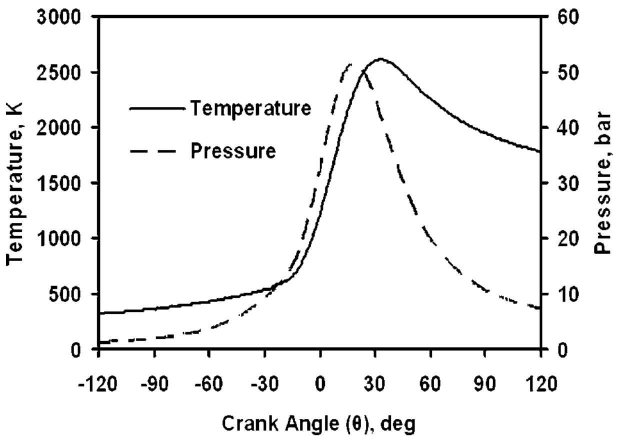
Engine specifications and operational parameters used in this study are provided in Table 3.

Solution methodology

Equation (44) is solved for each crank angle for the range of $-180° \leqslant \theta \leqslant 180°$ using a step size $\Delta\theta = 1°$. The values of $\theta = \pm 180°$ correspond to BDC whereas the value of $\theta = 0$ corresponds to TDC. The heat addition in equation (44) is only valid for $\theta_{\text{s}} < \theta < (\theta_{\text{s}} + \Delta\theta)$, i.e.

Table 3 Engine and operational specifications used in simulation

Fuel	C_8H_{18}
Compression ratio	8·3
Cylinder bore, m	0·0864
Stroke, m	0·0674
Connecting rod length, m	0·13
Crank radius, m	0·0337
Clearance volume, m^3	$5{\cdot}41 \times 10^{-5}$
Swept volume, m^3	$3{\cdot}95 \times 10^{-4}$
Engine speed, rev min^{-1}	2000–5000
Inlet pressure, bar	1
Ignition timing	−25° BTDC
Duration of combustion	70°
Wall temperature, K	400
Skirt clearance, m	23×10^{-6}
A_t	1
B	6
Ring thickness	2 rings each 1·5 mm
Oil type	SAE30
Oil temperature, °C	80



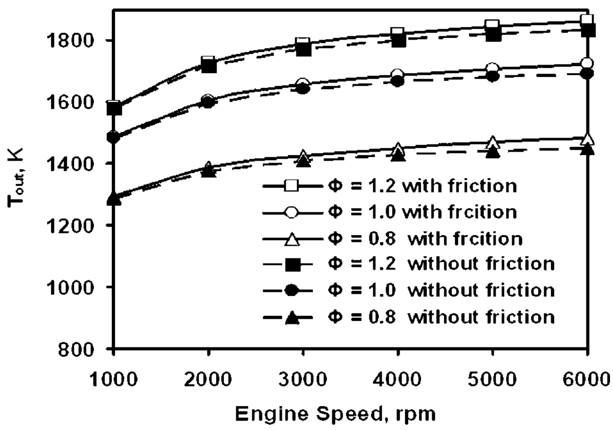
2 Variation of gas temperature and cylinder pressure versus crank angle for SI engine using friction and no-friction models running at 3000 rev min^{-1} and $\Phi = 1$

during the period of combustion. In solving equation (44), notice that k, P, T and h_{cg} are coupled, i.e. solution of one of these variables depends on the solution of others. Therefore, the solution methodology depends on using an iterative solution procedure. The solution procedure is as follows: by knowing the initial pressure of the gases at the BDC, the initial temperature of the gases is first calculated using equation (49). Once the value of the initial temperature is obtained, then the temperature dependent property C_{p} is calculated. For crank angle less than θ_{s}, the specific heat of the air is calculated using equation (1). Also, the specific heat for the fuel is calculated by using equation (3). Then, the specific heat for the air–fuel mixture before combustion is calculated by using equation (12). During combustion, i.e. $\theta_{\text{s}} < \theta < (\theta_{\text{s}} + \Delta\theta)$, equation (16) is used to calculate the specific heat. The number of moles for the product species for lean and rich mixture is calculated by using equations (19) and (20) respectively. The specific heats for the combustion products are calculated by using equation (13). After that, the gas constant for the mixture is determined by applying equations (4)–(7). After calculating C_{pmix} (T), using equation (16), the value of k is calculated using equation (17).

After getting the required gas mixture properties the new corrected temperature is calculated by using equation (49). The heat transfer coefficient is calculated using the Woschni model given by equation (50). Then, the friction work is determined from equation (33). At this point a new value of the corrected pressure is calculated by applying equation (49). Finally, the brake work and thermal efficiency is determined from equations (56) and (57) respectively. This procedure is repeated many times until the change between two successive iterations for pressure, and other variables such as T, k and h_{cg}, is $<0{\cdot}0001$. The previous mentioned procedure is repeated for each value of crank angle θ from -180 to $+180$.

Results and discussion

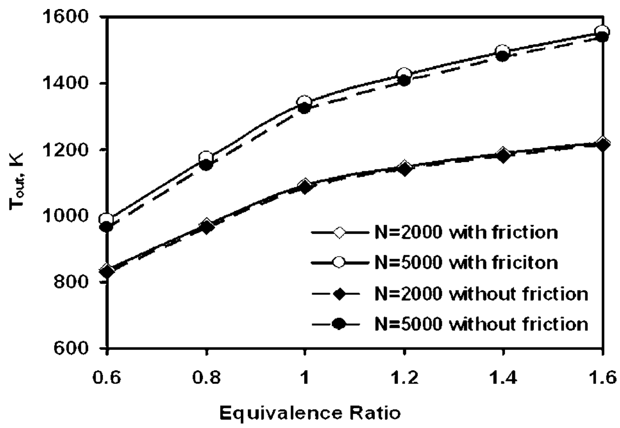
In order to examine the validity and sensitivity of the presented model Fig. 2 was plotted. It represents the variation of gas temperature and cylinder pressure versus crank angle for SI engine operating at



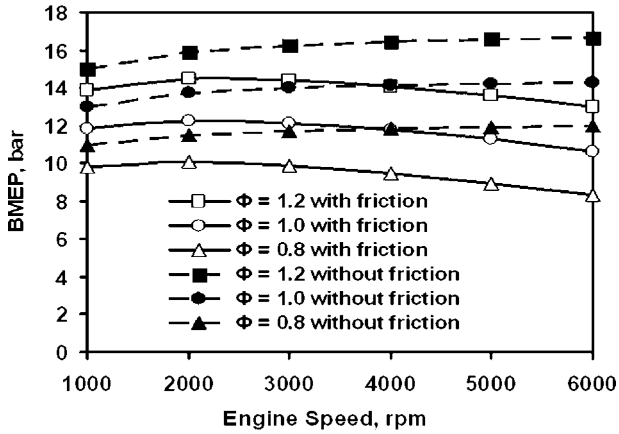
3 Outlet gas temperature versus engine speed at various equivalence ratios using friction and no-friction models

3000 rev min^{-1} and stoichiometric air–fuel mixture. The effects of engine speed and equivalence ratio on the gas outlet temperature, with and without friction are presented in Figs. 3 and 4. Figure 3 presents outlet gas temperature versus engine speed at equivalence ratios of rich mixture (Φ=1·2), stoichiometric mixture (Φ=1·0), and lean mixture (Φ=0·8). Whereas, Fig. 4 presents outlet gas temperature versus equivalence ratio at engine speeds of 2000 rev min^{-1} (low speed) and 5000 rev min^{-1} (high speed). Higher outlet temperatures are obtained at higher engine speeds and higher equivalence ratios. It is interesting to note that the effect of friction is not very significant on the gas outlet temperature, especially at low engine speeds. However, at higher engine speeds deviations between the two models become significant.

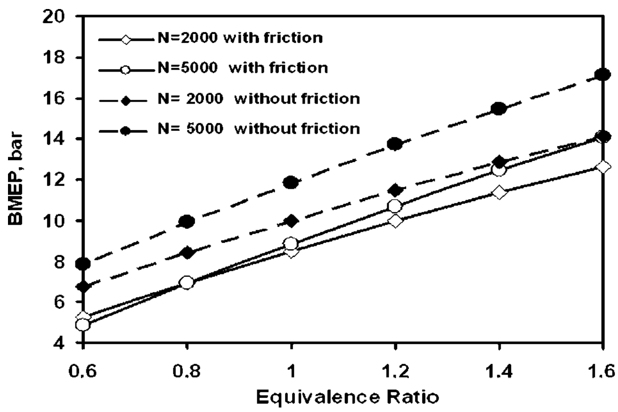
Similarly, the effects of engine speed and equivalence ratios on brake mean effective pressure (BMEP), using both cases (with and without friction), are presented in Figs. 5 and 6. It represents the variation of BMEP with engine speed for equivalence ratios representing rich, stoichiometric, and lean mixtures. Higher values of BMEP are obtained at higher equivalence ratios. Higher values of BMEP are obtained at higher engine speeds only when no friction model was used. However, when friction model was used BMEP increased with engine speed until reaching a maximum value at ~2500 rev min^{-1}, before it begins to decrease. Also, the deviations in BMEP between the two models were



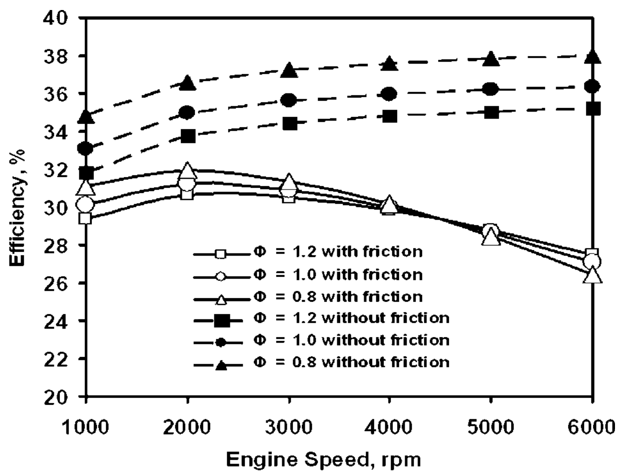
4 Outlet gas temperature versus equivalence ratio at low and high speeds using friction and no-friction models



5 Brake mean effective pressure versus engine speed at various equivalence ratios using friction and no-friction models



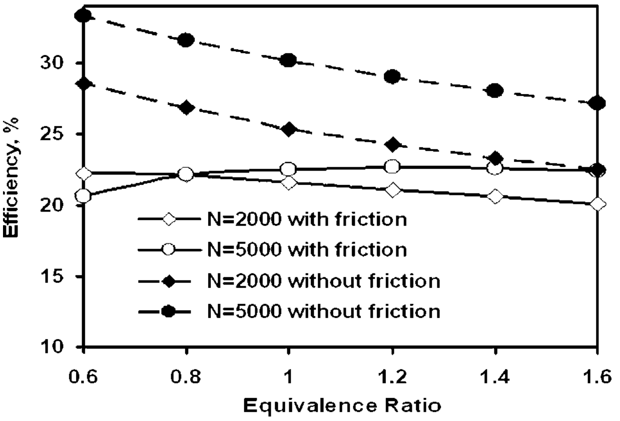
6 Brake mean effective pressure versus equivalence ratio at low and high speeds using friction and no-friction models



7 Efficiency versus equivalence ratio at different engine speeds using friction and no-friction models

very significant. The reason is that friction contributes significantly to reduce BMEP at higher engine speeds.

The effect of the two models on the cycle efficiency was also investigated. The results are presented in Figs. 7 and 8. It was found that higher values of thermal efficiencies are produced when frictionless model was considered. The reason is that friction reduces the cycle efficiency. When using the frictionless model, thermal

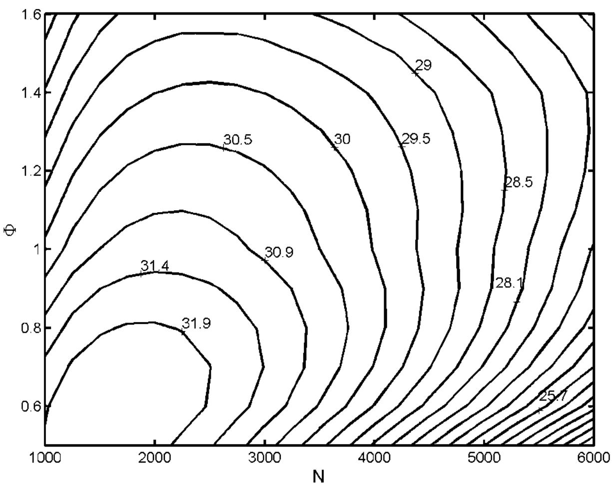


8 Efficiency versus equivalence ratio at low and high speeds using friction and no-friction models

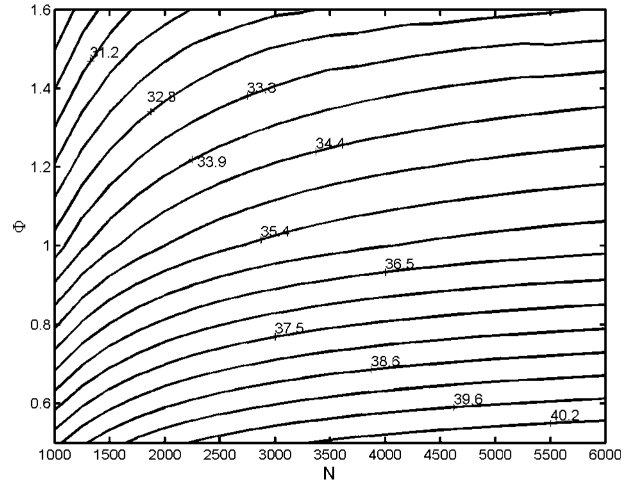
efficiency keeps increasing with engine speed (see broken lines in Fig. 7). It decreases with equivalence ratio (see broken lines in Fig. 8). However, it slightly increases until reaching a maximum at $\sim$2000 rev min^{-1} before decreasing (see solid lines in Fig. 7). It is interesting to note that efficiency decreases with equivalence ratio at low engine speeds, but it slightly increases with equivalence ratio at high engine speeds, until it levels out at stoichiometry and beyond. Finally, contour plots for efficiency for both models with and without friction are generated for various ranges and conditions. They are presented in Figs. 9 and 10. Theses plots show that cycle efficiency increases with speed when friction is not considered, which many thermodynamic models assume (Fig. 10). However, when, more realistically, friction is considered in the model, cycle efficiency decreases at higher engine speed (Fig. 9), which is more realistic. Actually, a maximum value of the efficiency can be located for the SI engine at low engine speed and very lean mixture. In our case, at engine speed of $\sim$2000 rev min^{-1} and equivalence ratio of $\sim$0·7, the maximum engine efficiency is obtained.

Conclusions

The investigation of using a gas mixture model instead of simply air as the working fluid for the analysis of a SI engine with friction was performed. The results were compared to those obtained from a frictionless engine.



9 Contour plots for efficiency including friction



10 Contour plots for efficiency with no friction

The study included the following parameters: cylinder pressure, gas temperature, BMEP and efficiency under a wide range of engine speeds (2000–6000 rev min^{-1}) and equivalence ratios (0·6–1·6). It was determined that assuming SI engines without taking friction into consideration results in an underestimation of the contribution of friction on the various engine parameters. Contour plots for efficiency of both models with and without friction were generated for various ranges of speed and conditions of fuel–air mixture. They demonstrate that cycle efficiency increases with speed when friction is not considered. However, when friction is considered in model cycle efficiency decreases at higher engine speed, which is more realistic. Thus, a maximum value of the efficiency can be located for the SI engine.

As a result, it is more realistic to use the gas mixture model instead of air as the working fluid for the analysis of SI engines and to consider friction especially when running at high engine speeds. This is expected to provide valuable guidelines for researchers and designers regarding performance evaluation and development of SI engines.

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