

## Thermodynamic analysis of spark-ignition engine using a gas mixture model for the working fluid

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### SUMMARY

This paper presents thermodynamic analysis of spark-ignition engine. A theoretical model of Otto cycle, with a working fluid consisting of various gas mixtures, has been implemented. It is compared to those which use air as the working fluid with variable temperature specific heats. A wide range of engine parameters were studied, such as equivalence ratio, engine speed, maximum and outlet temperatures, brake mean effective pressure, gas pressure, and cycle thermal efficiency. For example, for the air model, the maximum temperature, brake mean effective pressure (BMEP), and efficiency were about 3000 K, 15 bar, and 32%, respectively, at 5000 rpm and 1.2 equivalence ratio. On the other hand, by using the gas mixture model under the same conditions, the maximum temperature, BMEP, and efficiency were about 2500 K, 13.7 bar, and 29%. However, for the air model, at lower engine speeds of 2000 rpm and equivalence ratio of 0.8, the maximum temperature, BMEP, and efficiency were about 2000 K, 8.7 bar, and 28%, respectively. Also, by using the gas mixture model under these conditions, the maximum temperature, BMEP, and efficiency were about 1900 K, 8.4 bar, and 27%, i.e. with insignificant differences. Therefore, it is more realistic to use gas mixture in cycle analysis instead of merely assuming air to be the working fluid, especially at high engine speeds. Copyright © 2007 John Wiley & Sons, Ltd.

KEY WORDS: temperature-dependent specific heats; gas mixture; SI engine; Otto cycle

### 1. INTRODUCTION

In most models of air-standard power cycles, the air–fuel mixture and combustion products are approximated as ideal gases. In such cases air is assumed to be the working fluid with constant specific heats without taking into consideration the temperature dependence of the specific heats of the working fluid (Akash, 2001; Al-Sarkhi *et al.*, 2002; Ge *et al.*, 2005a; Chen *et al.*, 1998; Hou, 2004; Jafari and Hannani, 2006; Ozsoysal, 2006; Parlak, 2005a, b). However, due to the high rise in combustion temperature this assumption becomes less realistic. Although

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air-standard power cycle analysis gives only approximation of the actual conditions and outputs (Pulkrabek, 2004), it would be very useful to compare the performance of air-standard power cycles using more sound assumptions. In a recent study the effect of the heat loss on the overall engine efficiency was conducted by using constant specific heats for cylinder gases (Karamangil *et al.*, 2006). The convective heat transfer coefficient for cylinder side and coolant side of gasoline engines was determined at various engine parameters. These parameters include engine speed, inlet pressure, inlet temperature, excess air, and compression ratio. The in-cylinder convective heat transfer was determined by using two different models which are Anand's model and the Woshni model.

Other studies presented the effect of temperature-dependent specific heats on various air standard cycles such as Otto, Diesel, and Miller (Al-Sarkhi *et al.*, 2006a, b; Ge *et al.*, 2005b). However, the model used for temperature-dependent specific heats was linear. Linear models can be applied with moderate temperature changes. However, for large changes in temperature, more accurate models are needed. 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 (Abu-Nada *et al.*, 2006). In the above-mentioned models, air was used as the working fluid to perform their thermodynamic analyses. Such models are used for comparison reasons in order to show the effect of varying engine parameters, conditions, fluid properties, etc. Therefore, the objective of this work is to model the working fluid as a gas mixture, with temperature-dependent specific heats, and to examine the effect of this model on the engine performance parameters. Moreover, the results are compared to the air with temperature-dependent specific heats.

## 2. THERMODYNAMIC PROPERTIES OF AIR-FUEL MIXTURE AND COMBUSTION PRODUCTS

An equation describing the variation of air specific heats for the temperature range 300–3500 K is found in literature (Sonntag *et al.*, 1998). The equation 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). It is presented in the following equation:

$$C_p = 2.506 \times 10^{-11} T_g^2 + 1.454 \times 10^{-7} T_g^{1.5} - 4.246 \times 10^{-7} T_g + 3.162 \times 10^{-5} T_g^{0.5} + 1.3303 - 1.512 \times 10^4 T_g^{-1.5} + 3.063 \times 10^5 T_g^{-2} - 2.212 \times 10^7 T_g^{-3} \quad (1)$$

The results obtained from the above equation are in agreement with those reported elsewhere (Burcat and Ruscic, 2005; Heywood, 1988); see Figure 1. It is found from Equation (1) that specific heat at constant pressure increases with temperature from about 1.0 kJ kg<sup>-1</sup> K at 300 K to about 1.3 kJ kg<sup>-1</sup> K at 3000 K. Definitely, such difference should be taken into consideration. Similarly, the specific heat ratio, *k*, decreases from 1.40 to about 1.28 within the same temperature range.

In real life SI engines the combustion products have temperature-dependent specific heats. The most common combustion products are CO<sub>2</sub>, CO, H<sub>2</sub>O, N<sub>2</sub>, O<sub>2</sub>, and H<sub>2</sub>. The specific heats of these species have different dependence on temperature. Some species specific heats are strongly dependent on temperature, while others are less dependent. Thus, it is more accurate to calculate the specific heat of the mixture as a summation of individual species specific heats

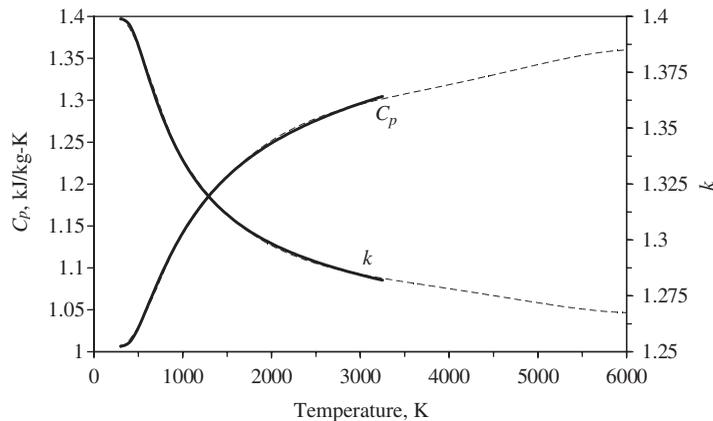


Figure 1.  $C_p$  and  $k$  versus temperature for air (thick lines representing results of Equation (1); thin lines representing data obtained from Burcat and Ruscic, 2005).

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:  $\text{CO}_2$ ,  $\text{CO}$ ,  $\text{H}_2\text{O}$ ,  $\text{N}_2$ ,  $\text{O}_2$ , and  $\text{H}_2$ . The temperature-dependent specific heat for these combustion product species takes the general form (Ferguson and Kirkpatrick, 2001):

$$\frac{C_p}{R_g} = a_1 + a_2 T + a_3 T^2 + a_4 T^3 + a_5 T^4 \quad (2)$$

The constants  $a_1$ – $a_5$  for all combustion species are given in Table I. Also, the specific heat of the fuel is assumed to be temperature dependent and it takes the following form (Heywood, 1988):

$$\tilde{C}_p = -0.55313 + 181.62 \left( \frac{T}{1000} \right) - 97.787 \left( \frac{T}{1000} \right)^2 + 20.402 \left( \frac{T}{1000} \right)^3 - 0.03095 \left( \frac{T}{1000} \right)^{-2} \quad (3)$$

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

The gas constant for the mixture is calculated as follows:

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

The molar mass of the mixture is determined as

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

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

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

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

$$y_f = \frac{1}{1 + 4.76 a_s / \Phi} \quad (7)$$

Table I. Coefficients used in Equation (2)\*. Coefficients for species temperature-dependent specific heats (I)  $T \leq 1000$  and (II)  $1000 < T < 3200$  K.

Species	$a_1$	$a_2$	$a_3$	$a_4$	$a_5$
<b>I</b>					
CO <sub>2</sub>	0.2400779E1	0.8735096E-2	-0.6660708E-5	0.2002186E-8	0.632740E-15
H <sub>2</sub> O	0.40701275E1	-0.1108450E-2	0.4152118E-5	-0.296374E-8	0.807021E-12
N <sub>2</sub>	0.36748261E1	-0.1208150E-2	0.2324010E-5	-0.6321756E-9	-0.225773E-12
O <sub>2</sub>	0.36255985E1	-0.1878218E-2	0.7055454E-5	-0.6763513E-8	0.215560E-11
CO	0.37100928E1	-0.1619096E-2	0.3692359E-5	-0.2031967E-8	0.239533E-12
H <sub>2</sub>	0.30574451E1	0.267652E-2	-0.5809916E-5	0.5521039E-8	-0.181227E-11
<b>II</b>					
CO <sub>2</sub>	0.4460800E1	0.3098170E-2	-0.1239250E-5	0.2274130E-9	-0.155259E-13
H <sub>2</sub> O	0.27167600E1	0.294513E-2	-0.802243E-6	0.102266E-9	-0.484721E-14
N <sub>2</sub>	0.289631E1	0.151548E-2	-0.572352E-6	0.998073E-10	-0.652235E-14
O <sub>2</sub>	0.362195E1	0.736182E-3	-0.196522E-6	0.362015E-10	-0.289456E-14
CO	0.298406E1	0.148913E-2	-0.578996E-6	0.103645E-9	-0.693535E-14
H <sub>2</sub>	0.3100190E1	0.511194E-3	0.526442E-7	-0.349099E-10	0.369453E-14

\*Taken from Ferguson and Kirkpatrick (2001).

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

where  $\Phi$  is the equivalence ratio and is given as  $\Phi = AF_s/AF$  and  $a_s$  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,

$$y_a = 1 - y_f \quad (9)$$

$$x_a = 1 - x_f \quad (10)$$

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

$$C_{p_{\text{mix}}} = C_{p_a}x_a + C_{p_f}x_f \quad (11)$$

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

$$C_{p_{\text{mix}}} = \sum_{i=1}^n C_{p_i}x_i \quad (12)$$

where  $i$  goes for CO<sub>2</sub>, CO, H<sub>2</sub>O, N<sub>2</sub>, O<sub>2</sub>, and H<sub>2</sub>. The mass fraction  $x_i$  is given as

$$x_i = \frac{n_i M_i}{m_{\text{mix}}} \quad (13)$$

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

$$m_{\text{mix}} = \sum_{i=1}^n n_i M_i \quad (14)$$

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:

$$C_{p_{\text{mix}}} = C_{p_{\text{air-fuel}}}(1 - x_b) + C_{p_{\text{products}}}(x_b) \quad (15)$$

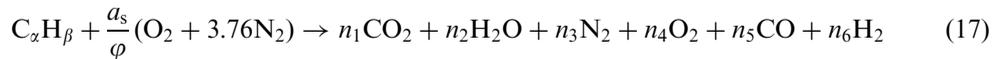
where  $x_b$  is evaluated from the Weibe function and represents the burned fraction of the fuel.

Finally, the specific heat ratio is calculated as

$$k = \frac{C_{p_{\text{mix}}}}{C_{v_{\text{mix}}}} = \frac{C_{p_{\text{mix}}}}{C_{p_{\text{mix}}} - R_{\text{mix}}} \quad (16)$$

### 3. COMBUSTION REACTIONS

By considering the existence of only six species ( $\text{CO}_2$ ,  $\text{H}_2\text{O}$ ,  $\text{N}_2$ ,  $\text{O}_2$ ,  $\text{CO}$ , and  $\text{H}_2$ ), in the combustion products, the chemical reaction for burning one mole of hydro-carbon fuel is written as



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

$$n_1 = \alpha, \quad n_2 = \frac{\beta}{2}, \quad n_3 = 3.76 \frac{a_s}{\Phi}, \quad n_4 = a_s \left( \frac{1}{\Phi} - 1 \right), \quad n_5 = 0, \quad n_6 = 0 \quad (18)$$

However, for  $\Phi > 1$  (Rich Mixture), it is assumed that there is no  $\text{O}_2$  in the combustion products. Thus, the numbers of moles for the combustion products are given as

$$n_1 = \alpha - n_5, \quad n_2 = \frac{\beta}{2} - d_1 + n_5, \quad n_3 = 3.76 \frac{a_s}{\Phi}, \quad n_4 = 0, \quad n_5 = n_5, \quad n_6 = d_1 - n_5 \quad (19)$$

where  $n_5 = -b_1 + \sqrt{b_1^2 - 4a_1c_1/2a_1}$

$$a_1 = 1 - K \quad (20)$$

$$b_1 = \frac{\beta}{2} + K\alpha - d_1(1 - K) \quad (21)$$

$$c_1 = -\alpha d_1 K \quad (22)$$

$$d_1 = 2a_s \left( 1 - \frac{1}{\Phi} \right) \quad (23)$$

The equilibrium constant  $K$  (T) in Equation (15) is a curve fit of JANAF table for the temperature range  $400 < T < 3200$  K (Ferguson and Kirkpatrick, 2001).

$$K = \exp \left( 2.743 - \frac{1.761}{(T/1000)} - \frac{1.611}{(T/1000)^2} + \frac{0.2803}{(T/1000)^3} \right) \quad (24)$$

In deriving the previous relations for rich mixture, an equilibrium reaction is assumed to take place between the species  $\text{CO}_2$ ,  $\text{H}_2\text{O}$ ,  $\text{CO}$ , and  $\text{H}_2$ . The equilibrium reaction is given as (Ferguson and Kirkpatrick, 2001)



#### 4. THERMODYNAMIC ANALYSIS

For a closed system and a small change of the process, the first law of thermodynamics is simply written as

$$\delta Q - \delta W = dU \quad (26)$$

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

$$\delta Q_{\text{in}} - \delta Q_{\text{loss}} - (PdV) = dU \quad (27)$$

For an ideal gas the equation of state is expressed as

$$PV = mR_g T_g \quad (28)$$

By differentiating Equation (28), we can get

$$PdV + VdP = mR_g dT_g \quad (29)$$

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

$$dU = mC_v dT_g \quad (30)$$

By substituting Equation (30) into Equation (29)

$$dU = \frac{C_v}{R_g} (PdV + VdP) \quad (31)$$

By substituting Equations (27) and (31) into Equation (26), the following equation is obtained:

$$\delta Q_{\text{in}} - \delta Q_{\text{loss}} - (PdV) = \frac{C_v}{R_g} (PdV + VdP) \quad (32)$$

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

$$Q_{\text{in}} = m_f LHV \quad (33)$$

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

$$Q(\theta) = Q_{\text{in}} x_b \quad (34)$$

where  $x_b$  is the burned fraction of the fuel and is expressed as (Ferguson and Kirkpatrick, 2001)

$$x_b = 1 - \exp\left(-5\left(\frac{\theta - \theta_s}{\Delta\theta}\right)^3\right) \quad (35)$$

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

$$Q_{\text{loss}} = \frac{h_{\text{cg}} A_h}{\omega} (T_g - T_w) d\theta \quad (36)$$

By substituting Equations (33)–(36) into Equation (32) followed by differentiation with respect to crank angle ( $\theta$ ), the following equation is obtained:

$$\frac{dP}{d\theta} = \frac{k-1}{V} \left( Q_{in} \frac{dx_b}{d\theta} - \frac{h_{cg} A_h}{\omega} (T_g - T_w) \frac{\pi}{180} \right) - k \frac{P}{V} \frac{dV}{d\theta} \quad (37)$$

Equation (37) can be solved by using explicit finite difference technique with second-order accurate differentiation. The result is given as

$$P(\theta) = \frac{4}{3} P(\theta - \Delta\theta) - \frac{1}{3} P(\theta - 2\Delta\theta) + \frac{k-1}{3V} Q_{in} (3x_b(\theta) - 4x_b(\theta - \Delta\theta) + x_b(\theta - 2\Delta\theta)) + \frac{2(k-1)}{3} \frac{h_{cg} A_h(\theta)(T_g - T_w)}{3V} \frac{1}{\omega} - \frac{2kP(\theta - \Delta\theta)}{3V(\theta)} \left( \frac{V(\theta + \Delta\theta) - V(\theta - \Delta\theta)}{2\Delta\theta} \right) \quad (38)$$

where  $dP/d\theta$  is expressed as

$$\frac{dP}{d\theta} = \frac{3P(\theta) - 4P(\theta - \Delta\theta) + P(\theta - 2\Delta\theta)}{2\Delta\theta} \quad (39)$$

The instantaneous cylinder volume, area, and displacement are given by the slider crank model as (Pulkrabek, 2004)

$$V(\theta) = V_c + \frac{\pi D^2}{4} x(\theta) \quad (40)$$

$$A_h(\theta) = \frac{\pi D^2}{4} + \frac{\pi DS}{2} (R + 1 - \cos(\theta) + (R^2 - \sin^2(\theta))^{1/2}) \quad (41)$$

$$x(\theta) = (\ell + R) - (R \cos(\theta) + (\ell^2 - \sin^2(\theta))^{1/2}) \quad (42)$$

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

$$T_g = \frac{P(\theta)V(\theta)}{mR_g} \quad (43)$$

The convective heat transfer coefficient in Equation (37)  $h_{cg}$  is given by the Woschni model as (Ferguson and Kirkpatrick, 2001; Stone, 1999; Woschni, 1967)

$$h_{cg} = 3.26D^{-0.2} P^{0.8} T_g^{-0.55} w^{0.8} \quad (44)$$

The velocity of the burned gas and is given as

$$w(\theta) = \left( 2.28 \bar{U}_p + C_1 \frac{V_d T_{gr}}{P_r V_r} (P(\theta) - P_m) \right) \quad (45)$$

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 the same position to obtain  $P$  without combustion (pressure values in cranking). The value of  $C_1$  is given as

For compression process:  $C_1 = 0$ .

For combustion and expansion processes:  $C_1 = 0.00324$ .

Table II. Engine and operational specifications used in simulation.

Fuel	C <sub>8</sub> H <sub>18</sub>
Compression ratio	8.3
Cylinder bore (m)	0.0864
Stroke (m)	0.0674
Connecting rod length (m)	0.13
Crank radius (m)	0.0337
Number of cylinders	1
Clearance volume (m <sup>3</sup> )	5.41 × 10 <sup>-5</sup>
Swept volume (m <sup>3</sup> )	3.95 × 10 <sup>-4</sup>
Engine speed (rpm)	2000–5000
Inlet pressure (bar)	1
Equivalence ratio	1
Ignition timing	–25° BTDC
Duration of combustion	70°
Wall temperature (K)	400

By which the average piston speed is calculated from

$$\bar{U}_p = \frac{2NS}{60} \quad (46)$$

Engine and operational specifications used in present simulation are given in Table II.

## 5. SOLUTION METHODOLOGY

Equation (38) is solved for each crank angle for the range of  $-180 \leq \theta \leq 180^\circ$  using a step size  $\Delta\theta = 1^\circ$ . The values of  $\theta = \pm 180^\circ$  correspond to bottom dead centre (BDC), whereas the value of  $\theta = 0$  corresponds to top dead centre (TDC). The heat addition in Equation (38) is only valid for  $\theta_s < \theta < (\theta_s + \Delta\theta)$ , i.e. during the period of combustion. In solving Equation (38), note 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 (43). Once the value of the initial temperature is obtained, then the temperature-dependent property  $C_p$  is calculated. For crank angle less than  $\theta_s$ , 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 (11). During combustion, i.e.  $\theta_s < \theta < (\theta_s + \Delta\theta)$ , Equation (15) is used to calculate the specific heat. However, for  $\theta > \theta_s$ , the number of moles for the product species for lean and rich mixture are calculated by using Equations (18) and (19), respectively. Then the specific heats for the combustion products are calculated by using Equation (12). After that, the gas constant for the mixture is determined by applying Equations (4)–(7). After calculating  $C_{p_{mix}}(T)$  for the mixture, the value of  $C_{v_{mix}}(T)$  is then determined from the relation  $[C_{v_{mix}}(T) = C_{p_{mix}}(T) - R_{mix}]$ . Thereafter, the value of  $k$  is calculated as  $k_{mix}(T) = C_{p_{mix}}(T)/C_{v_{mix}}(T)$ .

After getting the required gas mixture properties the new corrected temperature is calculated by using Equation (43). Then, the heat transfer coefficient is calculated using the Woschni model

given by Equation (44). At this point a new value of the corrected pressure is calculated by applying Equation (38). This value is compared to a previous iteration. 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 less than 0.0001. The previous mentioned procedure is repeated for each value of crank angle  $\theta$  from  $\theta = -180$  to  $+180$ .

## 6. RESULTS AND DISCUSSION

Figure 2 represents cylinder pressure in order to examine the validity and sensitivity of the presented model. It shows the variation of cylinder pressure *versus* volume for SI engine running at 3000 rpm. Air with variable specific heats is compared to gas mixture running at a stoichiometric air–fuel mixture. The deviation between the two models is obvious. Higher values of pressure are obtained when air is assumed as the working fluid. However, when air–fuel mixture model is used the same trend is reported but the values are lower. For example, a maximum pressure of 53 bar is reported when the working fluid used is assumed as air, and a maximum pressure of about 46 bar is reported when the gas mixture model is implemented. The reason for such a difference can be explained as follows: during combustion process species with high values of specific heats are generated. These species include  $\text{CO}_2$  and  $\text{H}_2\text{O}$  besides the existence of heated unburned fuel. These components absorb some of the heat generated during combustion. This absorbed heat will be reflected as a temperature and pressure increase in the combustion chamber. This difference is very important from engineering point of view absorbed because the absorbed heat is not transformed into useful work. Such physics is not evident when air is used as the working fluid because the temperature-dependent specific heat of the air is much lower than that of the mentioned combustion species. Therefore, the use of air as the working fluid leads to an over-estimation of the power produced by the engine.

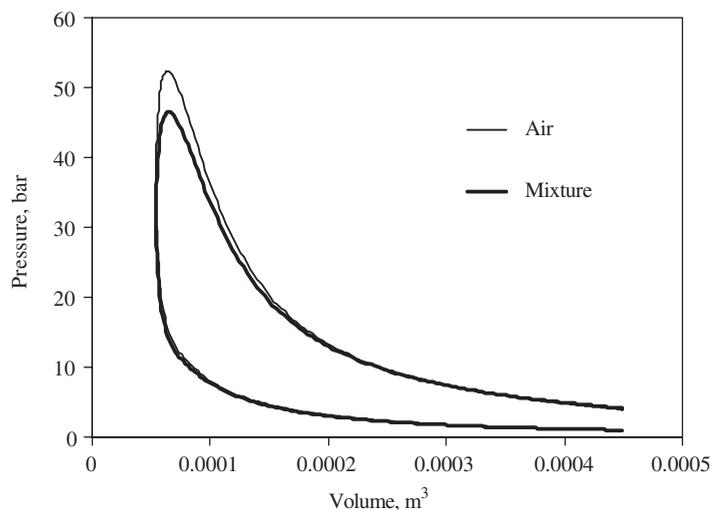


Figure 2. Variation of cylinder pressure *versus* volume for SI engine using air and mixture specific heats models running at 3000 rpm at  $\Phi = 1$ .

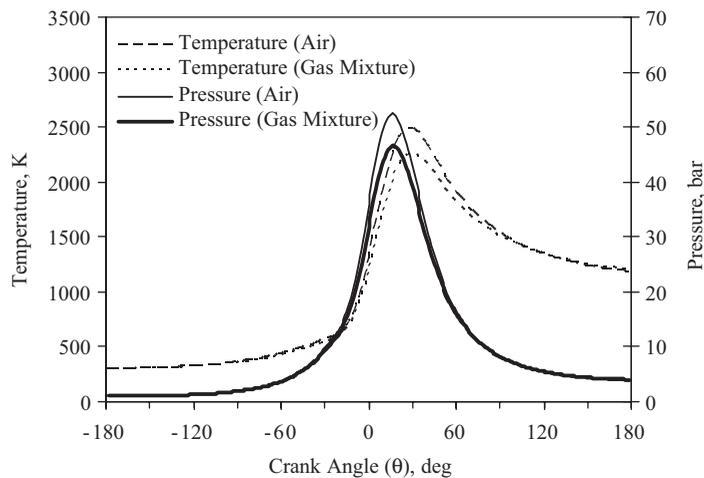
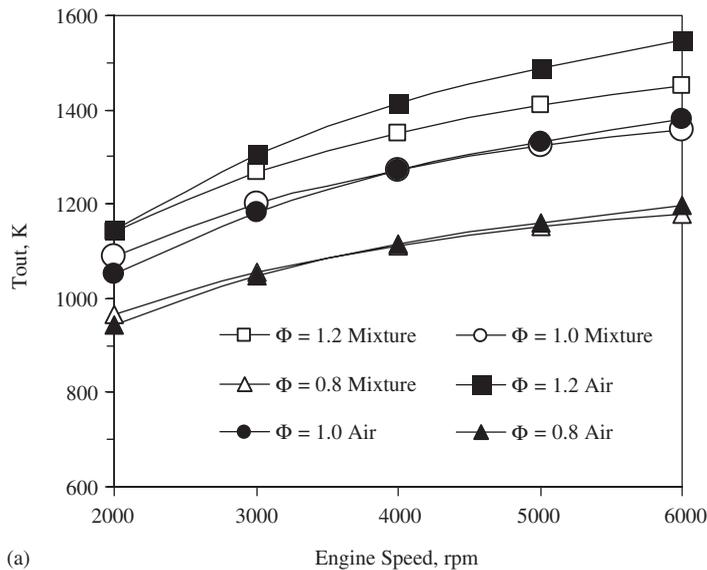


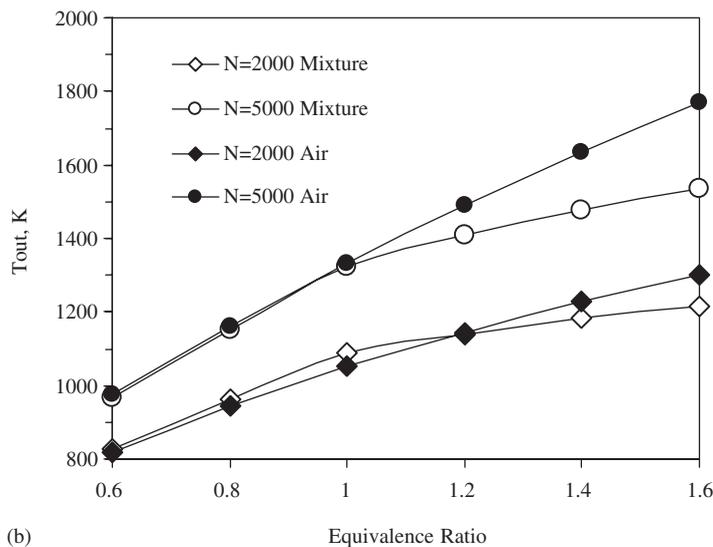
Figure 3. Variation of gas temperature and cylinder pressure *versus* crank for SI engine using air and mixture specific heat models running at 3000 rpm and  $\Phi = 1$ .

In order to study the effect of temperature, Figure 3 is presented. It shows variation of gas temperature *versus* crank angle using air and gas mixture as the working fluids running at engine speed of 3000 rpm at a stoichiometric air–fuel mixture. The gas temperature is presented for the two models. The temperature is over-estimated when air as the working fluid is assumed. The reported maximum temperature is about 2500 and 2300 K for air and gas mixture models, respectively. This affects heat transfer calculations between the engine and the coolant fluid. Most importantly, this increases the temperature of cylinder parts such as spark plug electrodes and exhaust valves. The increase in temperature of these parts increases the chances of occurrence of detonation.

The effects of engine speed and equivalence ratio on the gas outlet temperature, using air and gas mixture models, are presented in Figure 4. Figure 4(a) presents outlet gas temperature *versus* engine speed at equivalence ratios of rich ( $\Phi > 1$ ), stoichiometric ( $\Phi = 1$ ), and lean ( $\Phi < 1$ ) mixtures. Whereas Figure 4(b) presents outlet gas temperature *versus* equivalence ratio at engine speeds of 2000 rpm (low speed) and 5000 rpm (high speed). Higher outlet temperatures are obtained at higher engine speeds and higher equivalence ratios. It is interesting to note that the effect of air and gas mixture models is very significant on the gas outlet temperature, especially when rich mixtures are considered. Higher engine speeds produce more significant deviations between the two models. Similarly, the effects of engine speed and equivalence ratio on the maximum gas temperature, using the two models, are presented in Figure 5. Figure 5(a) presents the maximum gas temperature *versus* engine speed at equivalence ratios of rich, stoichiometric, and lean mixtures. On the other hand, Figure 5(b) presents maximum gas temperature *versus* equivalence ratio at engine speeds of 2000 rpm (low speed) and 5000 rpm (high speed). Higher values of maximum temperatures are obtained at higher engine speeds and higher equivalence ratios. Again, as noted previously, the effect of the gas model is very significant on the reported maximum gas temperatures.



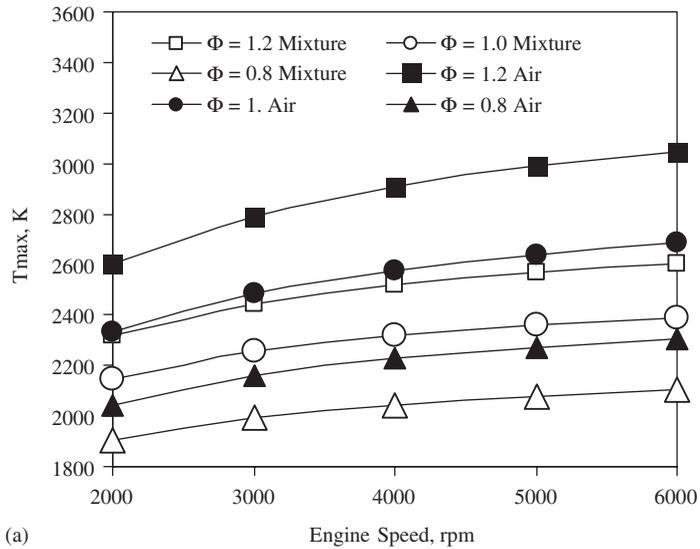
(a)



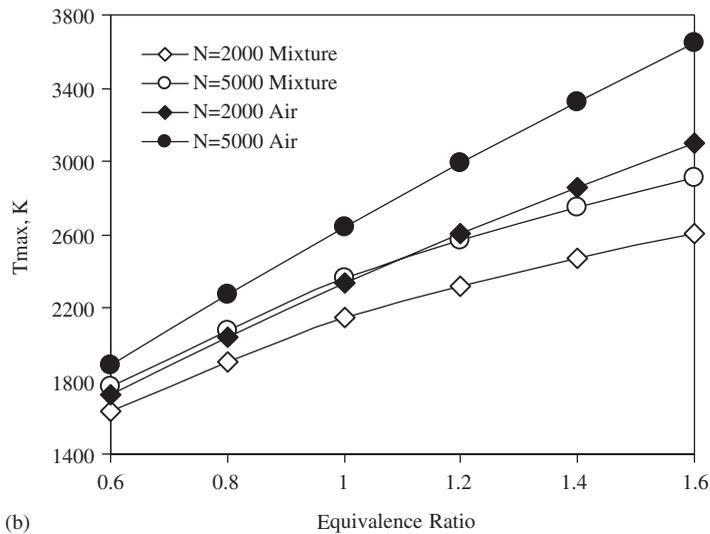
(b)

Figure 4. (a) Outlet gas temperature *versus* engine speed at various equivalence ratios using air and mixture specific heat models and (b) outlet gas temperature *versus* equivalence ratio at different engine speeds using air and mixture specific heat models.

The effects of engine speed and equivalence ratios on brake mean effective pressure (BMEP), using both gas models, are presented in Figure 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 engine speeds and lower air–fuel ratios. Like



(a)



(b)

Figure 5. (a) Maximum gas temperature *versus* engine speed at various equivalence ratios using air and mixture specific heat models and (b) maximum gas temperature *versus* equivalence ratio at different engine speeds using air and mixture specific heat models.

previous findings, simulated BMEP is significantly lower when the gas mixture model is used instead of air.

Finally, the effect of the gas mixture model on the cycle efficiency was investigated. The results are presented in Figure 7. It was found that higher values of thermal efficiencies are produced at high engine speeds and at low equivalence ratios. Also, it was found that at low engine speeds

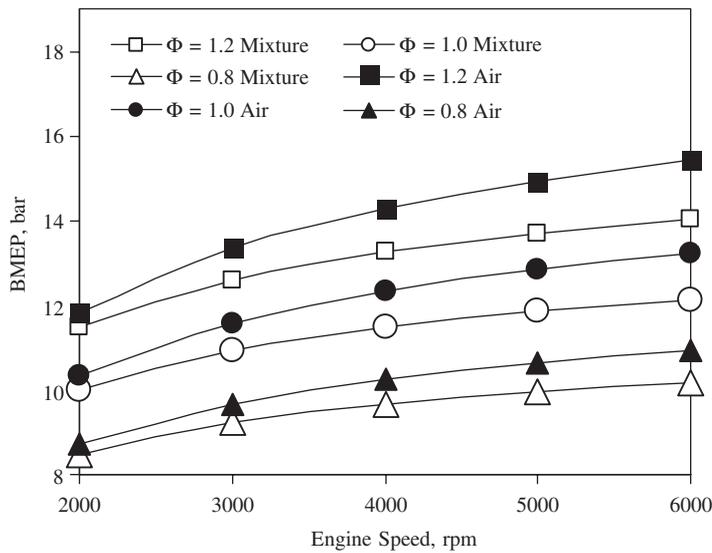


Figure 6. Brake mean effective pressure *versus* engine speed at various equivalence ratios using air and mixture specific heat models.

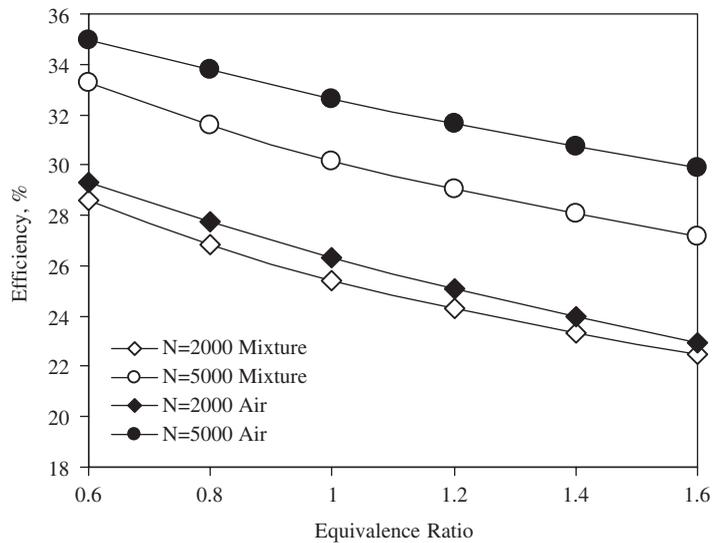


Figure 7. Efficiency *versus* equivalence ratio at different engine speeds using air and mixture specific heat models.

the effect of the gas mixture model was insignificant on efficiency. However, when the engine runs at high speed, the difference becomes more pronounced. A relative drop of 7% in efficiency results at 5000 rpm when the gas mixture model is used.

## 7. CONCLUSION

The effect of using a gas mixture model instead of air as the working fluid for the analysis of SI engines was investigated. The investigation covered in-cylinder pressure and temperature, BMEP and efficiency under a wide range of engine speeds (ranging from 2000 to 6000 rpm) and equivalence ratios (ranging from 0.6 to 1.6).

It was found that the use of air as the working fluid results in significant overestimation of the maximum pressure and temperature at high engine speeds and rich mixtures. This variation in temperature and pressure calculations has a direct effect on power and efficiency calculations. Moreover, it can influence the estimation of heat losses, exhaust emissions and detonation properties. However, the variation between the two models becomes less significant at low engine speeds and lean mixtures. An overestimation in the calculated values of BMEP and efficiency was also evident when air is used as the working fluid in comparison to the values obtained from the gas mixture model. Although this variation is almost negligible at low engine speeds and lean mixtures, it becomes increasingly significant as the engine speed increases and when rich mixtures are used. For example, at 5000 rpm and 1.2 equivalence ratio the efficiency value drops from 32 to 29% when using the gas mixture model instead of air.

Therefore, it is more realistic to use the gas mixture model instead of air as the working fluid for the analysis of spark ignition engines especially when running at high engine speeds and/or rich mixtures. This is expected to provide valuable guidelines for researchers and designers regarding performance evaluation and development of spark ignition engines.

## NOMENCLATURE

$a_s$	= number of moles of air at stoichiometric condition, dimensionless
$A$	= heat transfer area, $m^2$
AF	= air–fuel ratio, dimensionless
AF <sub>s</sub>	= air–fuel ratio for stoichiometric condition, dimensionless
$C_p$	= constant pressure specific heat, $kJ\ kg^{-1}\ K$
$C_v$	= constant volume specific heat, $kJ\ kg^{-1}\ K$
$D$	= cylinder diameter, m
$h_{cg}$	= heat transfer coefficient for gases in the cylinder, $W\ m^{-2}\ K$
$k$	= specific heat ratio, dimensionless
LHV	= lower heating value, $kJ\ kg^{-1}$
$\ell$	= connecting rod length, m
$m$	= 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, $kJ\ kg^{-1}\ K$
$R_{mix}$	= gas constant for the mixture, $kJ\ kg^{-1}\ K$

$R_u$	= universal gas constant = $8314.5 \text{ J kmol}^{-1} \text{ K}$
$T$	= gas temperature in the cylinder, K
$T_i$	= inlet temperature, K
$T_w$	= cylinder temperature, K
$U$	= internal energy, kJ
$U_p$	= piston speed, $\text{ms}^{-1}$
$V$	= cylinder volume, $\text{m}^3$
$V_c$	= clearance volume, $\text{m}^3$
$V_d$	= displacement volume, $\text{m}^3$
$X$	= distance from top dead centre, m
$x$	= mass fraction
$y$	= mole fraction
$x_b$	= burned fraction of the fuel, dimensionless
$w$	= average cylinder gas velocity, $\text{m s}^{-1}$
$\alpha$	= number of carbon atoms in the fuel
$\beta$	= number of hydrogen atoms in the fuel
$\theta$	= angle, degree
$\theta_s$	= start of combustion or heat addition, degree
$\Delta\theta$	= duration of combustion, degree
$\Phi$	= equivalence ratio

#### Abbreviation

$a$	= air
BDC	= bottom dead centre
$f$	= fuel
mix	= mixture
TDC	= top dead centre

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