Performance of diesel engine using gas mixture with variable specific heats model

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A thermodynamic, one-zone, zero-dimensional computational model for a diesel engine is established in which a working fluid consisting of various gas mixtures has been implemented. The results were compared to those which use air as the working fluid with variable specific heats. Most of the parameters that are important for compression ignition engines, such as equivalence ratio, engine speed, maximum temperature, gas pressure, brake mean effective pressure and cycle thermal efficiency, have been studied. Furthermore, the effect of boost pressure was studied using both the gas mixture and dependent temperature air models. It was found that the temperature dependent air model overestimates the maximum temperature and cylinder pressure. For example, for the air model, the maximum temperature and cylinder pressure were about 1775 K and 93·5 bar respectively at 2500 rev min^{-1}, and the fuel/air equivalence ratio Φ=0·6. On the other hand, when the gas mixture model is used under the same conditions, the maximum temperature and cylinder pressure were 1685 K and 87·5 bar respectively. This is reflected on the brake mean effective pressure and cycle thermal efficiency, which were both overestimated in the case of using the temperature dependent air model. The conclusions obtained in this study are useful when considering the design of diesel engines.

Keywords: Diesel Engine, Compression-Ignition Engine Simulation, Gas Mixture Model, Temperature Dependent Specific Heats

List of symbols

a constant used in equation (35)
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
BMEP brake mean effective pressure, bar
C_1 constant used in equation (33)
C_p constant pressure specific heat, cal g^{-1} mol^{-1} K^{-1}
C_v constant volume specific heat, kJ kg^{-1} K^{-1}
D cylinder diameter, m
h heat transfer coefficient for gases in the cylinder, W m^{-2} K^{-1}
k specific heat ratio, dimensionless
LHV lower heating value, kJ kg^{-1}
ℓ connecting rod length, m
m mass of cylinder contents, kg
m_d constant used in equation (35)
m_f mass of fuel in the cylinder, kg
m_p constant used in equation (35)
M molar mass
N engine speed, rev min^{-1}
p pressure inside cylinder, bar
p_i inlet pressure, bar
p_r reference state pressure
Q heat transfer, kJ
Q_d integrated energy release for diffusion combustion phases
Q_{in} heat added from burning fuel, kJ
Q_{loss} heat losses, kJ
Q_p integrated energy release for premixed combustion phases
R crank radius, m
R_g gas constant, kJ kg^{-1} K^{-1}
S engine stroke, m
T_g gas temperature in the cylinder, K
T_{gr} reference state gas temperature
T_i inlet temperature, K
T_w cylinder temperature, K
U internal energy, kJ
U_p piston speed, m s^{-1}
V cylinder volume, m^3
V_c clearance volume, m^3
V_d displacement volume, m^3
V_r reference state volume
X distance from top dead centre, m
w average cylinder gas velocity, m s^{-1}

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W work done, kJ
$\Delta\theta$ duration of combustion, °
θ angle, °
θ_d duration of the diffusion combustion phases, °
θ_p duration of the premixed combustion phases, °
Φ equivalence ratio

Introduction

The rapid development of computer technology has encouraged the use of simulation techniques to quantify the effect of the fundamental processes in the engine systems. The main reason for the growth in engine simulation arises from the economic benefits. Using computer models, large savings are possible in expensive experimental work. Obviously, models cannot replace real engine testing, but they are able to provide good estimates of engine performance and can thus help in selecting the best options for further development. In most models, air standard power cycles are used as a basis for analysing the actual conditions in real engines. In such cases, the working fluid being air is treated as a perfect gas with constant specific heats without taking into consideration the temperature dependence of the specific heats or the gas mixture of the working fluids.[1–9]

In order to deal with cycle calculations on more realistic basis, it is necessary to deal with certain fundamental aspects of the behaviour of the working fluid in real cycles, both under non-reacting and reacting conditions. Although air standard power cycle analysis gives only approximation of the actual conditions and outputs,[9] it would be very useful to study the cycle using variable specific heats and gas mixture model for the working fluid. In practical cycles, the heat capacities of the working fluids are variable and function of both temperature and gas mixture of the working fluid. This variation has a great influence on the performance of the cycles.

Several researchers studied the performance of air standard power cycles using more realistic assumptions. In the last few years, several authors used linear temperature specific heats model in their work.[10–13] These models can be applied with moderate temperature changes. However, for large changes in temperature, more accurate models are needed. For example, Zhao and Chen[14] analysed the performance of an irreversible diesel heat engine taking into account the temperature dependant heat capacities of the working fluid, the irreversibilities resulting from non-isentropic compression and expansion and heat leak losses through the cylinder wall. Wu *et al.*[15] carried out a numerical simulation of combustion characteristics for a closed diesel engine with different intake gas contents under different engine speeds and equivalent ratios.

Recently, several papers concerning variable specific heats,[16–18] variable specific heat ratio[19,20] and mathematical modelling[21–23] using finite time thermodynamics have been published.

In recent studies carried out by the current authors, spark ignition engine simulations were conducted taking into account the effect of heat loss, friction, rates of heat release, temperature dependant specific heats and gas mixture model on the overall engine performance.[24–27] The simulation code used in these studies employs a thoroughly validated thermodynamic, one-zone, zero-dimensional computational model.

The major goal of the present article is to model diesel engines using a gas mixture with temperature dependent specific heats as the working fluid. Furthermore, the effect of this model on the engine performance will be examined. Moreover, the results obtained from the gas mixture model will be compared to those obtained when air with temperature dependent specific heats is used as the working fluid.

Theoretical model

Thermodynamics properties of air–fuel mixture and combustion products

In real life compression ignition engines, the combustion products have temperature dependent specific heats. 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 is 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[20]

$$\frac{c_p}{R_g}=a_1+a_2T+a_3T^2+a_4T^3+a_5T^4 \qquad (1)$$

The constants a_1 through a_5 for all combustion species are given in Table 1.[28] Furthermore, the specific heat of

Table 1 Coefficients for species temperature dependent specific heats

Species	a_1	a_2	a_3	a_4	a_5
$T \leqslant 1000$ K					
CO_2	0.2400779×10	0.8735096×10^{-2}	-0.6660708×10^{-5}	0.2002186×10^{-8}	0.632740×10^{-15}
H_2O	0.40701275×10	-0.1108450×10^{-2}	0.4152118×10^{-5}	-0.296374×10^{-8}	0.807021×10^{-12}
N_2	0.36748261×10	-0.1208150×10^{-2}	0.2324010×10^{-5}	-0.6321756×10^{-8}	-0.225773×10^{-12}
O_2	0.36255985×10	-0.1878218×10^{-2}	0.7055454×10^{-5}	-0.6763513×10^{-8}	0.215560×10^{-11}
CO	0.37100928×10	-0.1619096×10^{-2}	0.3692359×10^{-5}	-0.2031967×10^{-8}	0.239533×10^{-12}
H_2	0.30574451×10	0.267652×10^{-2}	-0.5809916×10^{-5}	0.5521039×10^{-8}	-0.181227×10^{-11}
$1000<T<3200$ K					
CO_2	0.4460800×10	0.3098170×10^{-2}	-0.1239250×10^{-5}	0.2274130×10^{-9}	-0.155259×10^{-13}
H_2O	0.27167600×10	0.294513×10^{-2}	-0.802243×10^{-6}	0.102266×10^{-9}	-0.484721×10^{-14}
N_2	0.289631×10	0.151548×10^{-2}	-0.572352×10^{-6}	0.998073×10^{-10}	-0.652235×10^{-14}
O_2	0.362195×10	0.736182×10^{-3}	-0.196522×10^{-6}	0.362015×10^{-10}	-0.289456×10^{-14}
CO	0.298406×10	0.148913×10^{-2}	-0.578996×10^{-6}	0.103645×10^{-9}	-0.693535×10^{-14}
H_2	0.3100190×10	0.511194×10^{-3}	0.526442×10^{-7}	-0.349099×10^{-10}	0.369453×10^{-14}

the fuel is assumed to be temperature dependent, and it takes the following form[29]

$$\widetilde{C}_p = -0.55313 + 181.62\left(\frac{T}{1000}\right) - 97.787\left(\frac{T}{1000}\right)^2 + 24.402\left(\frac{T}{1000}\right)^3 - 0.03095\left(\frac{T}{1000}\right)^{-2} \quad (2)$$

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

The gas constant for the mixture is calculated as follows

$$R_{mix} = \frac{R_u}{M_{mix}} \quad (3)$$

where R_u is the universal gas constant. The molar mass of the mixture is determined as

$$M_{mix} = \sum_{i=1}^{n} y_i M_i \quad (4)$$

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

$$M_{mix} = y_a M_a + y_f M_f \quad (5)$$

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

$$y_f = \frac{1}{1 + 4{\cdot}76(a_s/\Phi)} \quad (6)$$

$$x_f = \frac{1}{1 + (AF_s/\Phi)} \quad (7)$$

where Φ is the fuel/air 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 and the mass fraction for the air are obtained respectively

$$y_a = 1 - y_f \quad (8)$$

$$x_a = 1 - x_f \quad (9)$$

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

$$C_{p_{mix}} = C_{p_a} x_a + C_{p_f} x_f \quad (10)$$

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

$$C_{p_{mix}} = \sum_{i=1}^{n} C_{p_i} x_i \quad (11)$$

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

$$x_i = \frac{n_i M_i}{m_{mix}} \quad (12)$$

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

$$m_{mix} = \sum_{i=1}^{n} n_i M_i \quad (13)$$

The gases within the combustion chamber consist of two main parts: gases, which have the air–fuel mixture properties and gases that take the properties of the combustion products. Thus, it is very reasonable to estimate the specific heat for the mixture as follows

$$C_{p_{mix}} = C_{p_{air-fuel}}(1 - x_b) + C_{p_{products}}(x_b) \quad (14)$$

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

$$k = \frac{C_{p\,mix}}{C_{v\,mix}} = \frac{C_{p\,mix}}{C_{p\,mix} - R_{mix}} \quad (15)$$

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 1 mol of hydrocarbon fuel is written as

$$C_\alpha H_\beta + \frac{a_s}{\phi}(O_2 + 3.76N_2) \rightarrow n_1 CO_2 + n_2 H_2O + n_3 N_2 n_4 O_2 + n_5 CO + n_6 H_2 \quad (16)$$

This chemical reaction is applicable for lean, stoichiometric or rich mixtures. Diesel engine air utilisation is generally limited to lean conditions. Higher equivalence ratios (stoichiometric or rich conditions) cause excessive smoke emissions. For $\Phi \leqslant 1$ (stoichiometric and lean mixtures), the numbers of moles of the combustion products are given as

$$n_1 = \alpha; \quad n_2 = \frac{\beta}{2}; \quad n_3 = 3.76\frac{a_s}{\Phi}; \quad n_4 = a_s\left(\frac{1}{\Phi} - 1\right); \quad n_5 = 0; \quad n_6 = 0 \quad (17)$$

Thermodynamic analysis

For a closed system, the first law of thermodynamics is written as

$$\delta Q - \delta W = dU \quad (18)$$

Using the definition of work, the first law can be expressed as

$$\delta Q_{in} - \delta Q_{loss} - (p\,dV) = dU \quad (19)$$

For an ideal gas, the equation of state is expressed as

$$pV = mR_g T_g \quad (20)$$

By differentiating equation (20), the following equation is obtained

$$p\,dV + V\,dp = mR_g\,dT_g \quad (21)$$

In addition, for an ideal gas, the change in internal energy is expressed as

$$dU = d(mC_v T_g) \quad (22)$$

Using the chain rule of differentiation, equation (22) is rearranged as

$$mR_g\,dT_g = \frac{R_g}{C_v}(dU - mT_g\,dC_v) \quad (23)$$

By substituting equation (23) into equation (21) and solving for the change in internal energy

$$dU=\frac{C_v}{R_g}(pdV+Vdp)+mT_g dC_v \quad (24)$$

Furthermore, by substituting equation (24) into equation (19), the first law is written as

$$\delta Q_{in}-\delta Q_{loss}-pdV=\frac{C_v}{R_g}(pdV+Vdp)+mT_g dC_v \quad (25)$$

Dividing equation (25) by $d\theta$

$$\frac{\delta Q_{in}}{d\theta}-\frac{\delta Q_{loss}}{d\theta}-p\frac{dV}{d\theta}=\frac{C_v}{R_g}\left(p\frac{dV}{d\theta}+V\frac{dp}{d\theta}\right)+mT_g\frac{dC_v}{d\theta} \quad (26)$$

Expressing the gradient of the specific heat as

$$\frac{dC_v}{d\theta}=\frac{dC_v}{dk}\frac{dk}{d\theta} \quad (27)$$

Noting that

$$\frac{R_g}{C_v}=k-1 \quad (28)$$

Plugging equation (28) into equation (27), then the gradient of the specific heat is expressed as

$$\frac{dC_v}{d\theta}=-\frac{R_g}{(k-1)^2}\frac{dk}{d\theta} \quad (29)$$

Substituting equation (29) into equation (26), the final form of the governing equations is

$$\frac{dp}{d\theta}=\frac{k-1}{V}\left(\frac{dQ_{in}}{d\theta}-\frac{dQ_{loss}}{d\theta}\right)-k\frac{p}{V}\frac{dV}{d\theta}+\frac{p}{k-1}\frac{dk}{d\theta} \quad (30)$$

In equation (30), the rate of the heat loss $\frac{dQ_{loss}}{d\theta}$ is expressed as

$$\frac{dQ_{loss}}{d\theta}=hA(\theta)(T_g-T_w)\left(\frac{1}{\omega}\right) \quad (31)$$

The convective heat transfer coefficient h in equation (31) is given by the Woschni model as[28,30,31]

$$h=3{\cdot}26D^{-0{\cdot}2}p^{0{\cdot}8}T_g^{-0{\cdot}55}w^{0{\cdot}8} \quad (32)$$

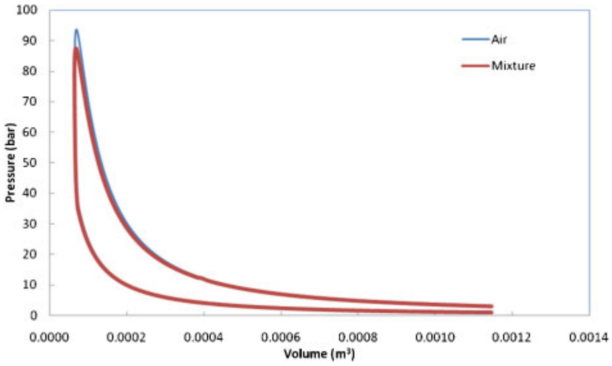
The velocity of the burned gas and is given as

$$w(\theta)=2{\cdot}28\overline{U_p}+C_1\frac{V_d T_{gr}}{p_r V_r}[p(\theta)-p_m] \quad (33)$$

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). Engine and operational specifications used in present simulation are given in Table 2. The value of C_1 is given as: for compression process, $C_1=0$, and for combustion and expansion processes, $C_1=0{\cdot}00324$. The average piston speed $\overline{U_p}$ is calculated from

$$\overline{U_p}=\frac{2NS}{60} \quad (34)$$

On the other hand, the rate of the heat input $dQ_{in}/d\theta$ (heat release) can be modelled using a dual Weibe function[28,32]



1 Rate of heat release model N=2500 rev min^{-1}, Φ=0·6, θ_p=10°, θ_d=60° and injection is −8°

$$\frac{dQ_{in}}{d\theta}=a\left(\frac{Q_p}{\theta_p}\right)m_p\left(\frac{\theta}{\theta_p}\right)^{m_p-1}\exp\left[-a\left(\frac{\theta}{\theta_p}\right)^{m_p}\right]+a\left(\frac{Q_d}{\theta_d}\right)m_d\left(\frac{\theta}{\theta_d}\right)^{m_d-1}\exp\left[-a\left(\frac{\theta}{\theta_d}\right)^{m_d}\right] \quad (35)$$

where p and d refer to premixed and diffusion phases of combustion. The parameters θ_p and θ_d represent the duration of the premixed and diffusion combustion phases. In addition, Q_p and Q_d represent the integrated energy release for premixed and diffusion phases respectively. The constants a, m_p and m_d are selected to match experimental data. For the current study, these values are selected as 6·9, 4 and 1·5 respectively.[28,32] It is assumed that the total heat input to the cylinder by combustion for one cycle is

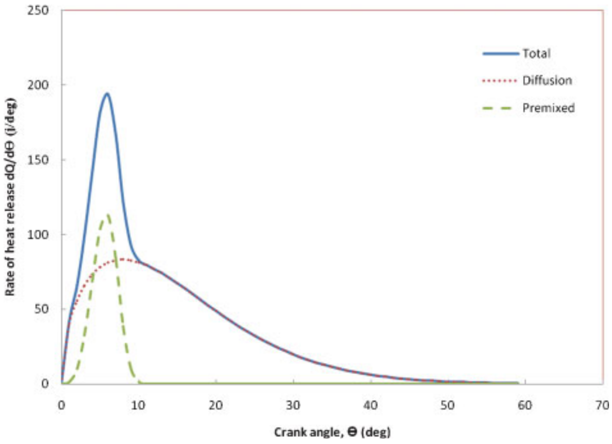
$$Q_{in}=m_f \text{LHV} \quad (36)$$

where 20% of this amount is assumed to take place in the premixed phase and the rest in the diffusion phase. Figure 1 shows a plot for the rate of heat addition as a function of crank angle.

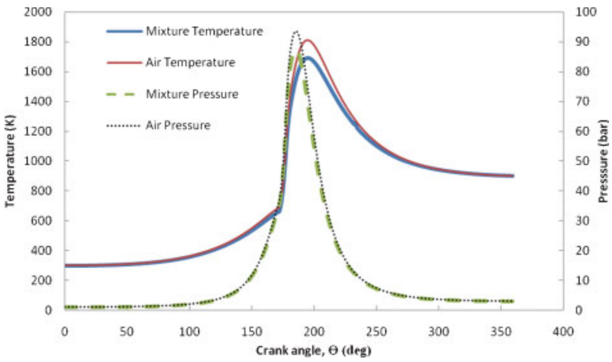
Equation (30) is discretised using a second order finite difference method to solve for the pressure at each crank angle θ. For full details of discretisation and numerical implementation, the reader is referred to Refs. 24–27. Once the pressure is calculated, the temperature of the gases in the cylinder can be calculated using the equation of state as

Table 2 Engine and operational specifications used in simulation

Fuel	$C_{14{\cdot}4}H_{24{\cdot}9}$
Compression ratio	18.
Cylinder bore, m	0·105
Stroke, m	0·125
Connecting rod length, m	0·1
Number of cylinders	1
Clearance volume, m^3	$6{\cdot}367\times10^{-5}$
Swept volume, m^3	$1{\cdot}082\times10^{-3}$
Engine speed, rev min^{-1}	1000–5000
Inlet pressure, bar	1
Equivalence ratio	0·2–1·2
Injection timing	−24 to −8°
Duration of combustion	60°
Duration of premixed combustion	8°
Wall temperature, K	400



2 Variation in cylinder pressure versus volume for compression ignition (CI) engine using variable air and mixture specific heats running at 2500 rev min^{-1} and Φ=0·6



3 Variation in gas temperature and cylinder pressure versus crank angle for CI engine using variable air and mixture specific heat models running at 2500 rev min^{-1} and Φ=0·6

$$T_{\mathrm{g}}=\frac{p(\theta)V(\theta)}{mR_{\mathrm{g}}} \quad (37)$$

The instantaneous cylinder volume, area and displacement are given by the slider crank model as[33]

$$V(\theta)=V_{\mathrm{c}}+\frac{\pi D^2}{4}x(\theta) \quad (38)$$

$$A_{\mathrm{h}}(\theta)=\frac{\pi D^2}{4}+\frac{\pi DS}{2}\left\{R+1-\cos(\theta)+\left[R^2-\sin^2(\theta)\right]^{1/2}\right\} \quad (39)$$

$$x(\theta)=(\ell+R)-\left\{R\cos(\theta)+\left[\ell^2-\sin^2(\theta)\right]^{1/2}\right\} \quad (40)$$

The thermal efficiency is defined as

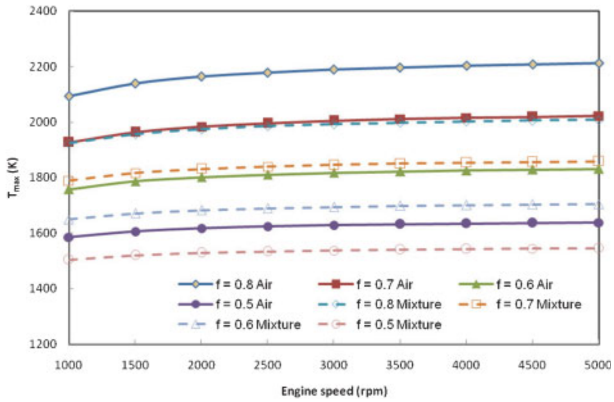
$$\eta=\frac{W_{\mathrm{net}}}{Q_{\mathrm{in}}} \quad (41)$$

While the brake mean effective pressure is defined as

$$\mathrm{BMEP}=\frac{W_{\mathrm{net}}}{V_{\mathrm{d}}} \quad (42)$$

Results and discussion

The rate of heat release model used in this work is a dual Wiebe function (Fig. 1). Three combustion phases are



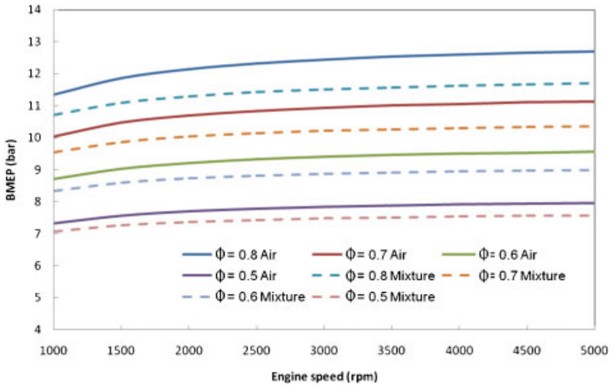
4 Maximum gas temperature versus engine speed at various equivalence ratios using variable air and mixture specific heat models

observed, namely, premixed combustion phase, diffusion combustion phase and late combustion phase.

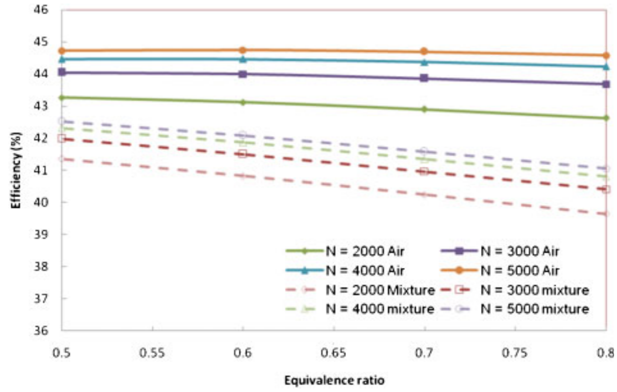
The variation in cylinder pressure is presented in Fig. 2 to examine the sensitivity and validity of the presented model. It shows the comparisons of in-cylinder pressures versus volume using air and gas mixture specific heats models at 2500 rev min^{-1} and Φ=0·6. Both models have similar trends, but the magnitude of the pressure is higher in the case of air as the working fluid. For example, the maximum reported pressures were about 93·5 and 87·5 bar using air and gas mixture specific heat models respectively. The use of air model overestimates the pressure inside the cylinder.

Figure 3 illustrates the influence of air and mixture specific heat models on the gas temperature and cylinder pressure. It shows variation in gas temperature and cylinder pressure versus crank angle using air and gas mixture as the working fluids running at 2500 rev min^{-1} and equivalence ratio of 0·6. Using gas mixture results in lower gas temperature and cylinder pressure. For example, the maximum reported temperature and pressure are 1775 and 1685 K, and 93·5 and 87·5 bar for air and gas mixture respectively. The reason is that air has a lower specific heat than air mixture does. During combustion, species with high values of specific heats are generated like CO_2 and H_2O, besides the existence of heated unburned fuel. These components absorb some of the heat generated during combustion, so that the temperature is higher inside the cylinder when air is used as a working fluid.

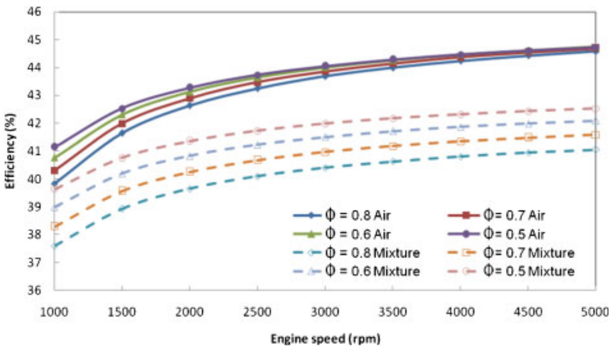
The effect of engine speed on the maximum gas temperature and BMEP are presented in Figs. 4 and 5 respectively. Figure 4 presents the maximum gas temperature versus the engine speed at equivalence ratios of 0·5, 0·6, 0·7 and 0·8. Higher maximum temperatures are obtained at higher equivalence ratios. For higher equivalence ratio, more fuel is burned in the cylinder, and therefore, more heat is released that leads to higher gas temperatures. Again, as noted previously, the effect of gas mixture model is very significant on the reported maximum gas temperature. The maximum gas temperature difference resulted from air and gas mixture model increases for high equivalent ratios. It can also be observed that the effect of equivalence ratio is more significant than the effect of engine speed. Figure 5



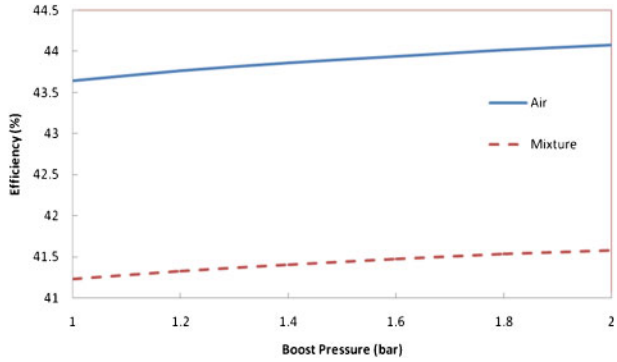
5 Brake mean effective pressure versus engine speed at various equivalence ratios using variable air and mixture specific heat models



7 Efficiency versus equivalence ratios at various engine speed using variable air and mixture specific heat models



6 Efficiency versus engine speed at various equivalence ratios using variable air and mixture specific heat models



8 Efficiency versus boost pressure using variable air and mixture specific heat models

presents BMEP versus engine speed at various equivalence ratios using air and gas mixture specific heat models. Similarly, the effect of gas mixture model is very significant on BMEP especially at high equivalence ratios. It is obvious that the difference in BMEP using the air and gas mixture specific heat models decreases with low equivalence ratios. From BMEP consideration, it is desirable to have high equivalence ratio to achieve high values of BMEP. However, diesel engine air utilisation is generally limited to $\Phi<0{\cdot}7$. Higher equivalence ratios cause excessive smoke emissions.

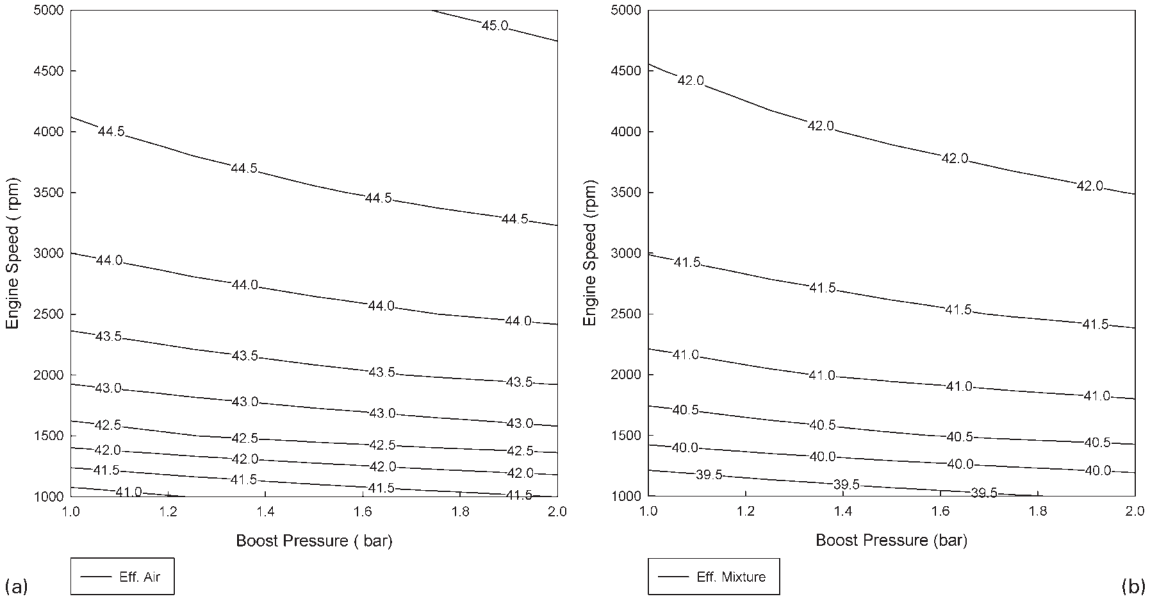
The effect of the gas model on cycle efficiency was investigated and demonstrated in Figs. 6 and 7. It is clear that the higher thermal efficiencies were reached at high engine speeds and low equivalence ratios. The cycle efficiency difference between the air and gas mixture specific heat models becomes more pronounced with the increase in both equivalence ratio and engine speed. In addition, it was found that for high engine speeds, the efficiency becomes independent of equivalence ratio when air model is used, which is non-realistic.

In order to study the effect of boost pressure, Fig. 8 is presented. It shows the variation in cycle efficiency versus boost pressure using air and gas mixture as working fluids at engine speed of 2500 rev min^{-1} and equivalence ratio of 0·6. Although they have similar trends, the efficiency is overestimated when air model is used as working fluid. The increase in efficiency by increasing the boost pressure agrees with the results obtained by Al-Hinti *et al.*[34] Finally, contour efficiency plots for both air and gas mixture specific heat models are generated for various boost pressures and engine speeds. They are presented in Fig. 9. These plots show that cycle efficiency increases with engine speed and boost pressure. Furthermore, it is clear that the effect of engine speed is dominant over that of boost pressure, and as stated previously, the air model overestimates the cycle efficiency.

Conclusions

In the present work, a diesel cycle model, assuming a gas mixture as the working fluid, has been investigated numerically. The results were compared to those obtained using variable temperature specific heat model in which air is used as the working fluid. The investigation covered the in-cylinder pressure and temperature, BMEP, cycle efficiency for different engine speeds and equivalence ratios and boost pressure. It was clear from the results obtained that the use of air as the working fluid overestimates the maximum temperature and pressure in the cylinder. The results from this research are compatible with those in the open literature for spark ignition engines.

There are significant effects of the gas mixture model on the performance of the cycle; therefore, it is more



a air; *b* mixture

9 Interpolated contour efficiency plots for both air and gas mixture specific heat models for various boost pressures and engine speeds

realistic to use the gas mixture model instead of air as the working fluid for the analysis of CI engines. This should be considered in practical cycle analysis.

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