Modelling the performance of a syngas fueled engine: Effect of excess air and CO2 as combustion diluents.

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Bio-syngas, derived from the pyrolysis and gasification of a biomass feedstock, can be used as an alternative fuel in reciprocating engines. This mixture of CO, H_2, CH_4, and CO_2 is less calorific than traditional fuels and has different in-cylinder combustion properties, ultimately resulting in a power derating for syngas engines when compared to traditional engines of a similar size. Nevertheless, syngas remains an attractive renewable, low-carbon fuel suitable for use in spark-ignited (SI) internal combustion engines. The current study investigates syngas engine performance across varying air/fuel ratios and the effect of substituting CO_2 for excess air.

In the present work, a representative time-dependent, thermodynamic model of a four-stroke internal combustion engine is derived (eqs. 1 – 3). Fuel combustion is sub-modelled using a Weibe function tuned specifically for the combustion characteristics of syngas mixtures of predominantly H_2, CO, and CO_2 in air (eq. 4). Instantaneous heat lost to the cylinder wall is calculated using a convective heat transfer model with the heat transfer coefficient determined using a Nusslet-Reynolds correlation (eq. 5).

$$\frac{dP}{d\theta} = \frac{\gamma - 1}{V} \cdot \left(\Delta H_c \cdot \frac{dX_b}{d\theta} - \frac{dQ_L}{d\theta}\right) - \frac{\gamma \cdot P}{V} \cdot \frac{dV}{d\theta} + \frac{\gamma - 1}{V}\left(\frac{\gamma_0\, T_0\, R}{\gamma_0 - 1}\right) \cdot \frac{dm}{d\theta} \tag{1}$$

$$V(\theta) = V_c + \frac{V_c\,(r_c - 1)}{2}\left(1 - \cos(\theta) + \epsilon \cdot \frac{\sin^2(\theta)}{2}\right) \tag{2}$$

$$\frac{dV}{d\theta} = \frac{V_c\,(r_c - 1)}{2}\sin(\theta) \cdot (1 + \epsilon\cos(\theta)) \tag{3}$$

$$X_b(\theta) = 1 - \exp^{-2.23\left(\frac{\theta - \theta_s}{\theta_d}\right)^{1.71}} \tag{4}$$

$$h_c = a\frac{B^{b-1}}{k}\left(\frac{\rho\,\overline{U}_p}{\mu}\right)^b \tag{5}$$

When validated against experimental data (Shivapuji & Dasappa, 2013), this model can predict brake power output and in-cylinder pressure profiles, as highlighted in Table 1. Additionally, the model predicts the overall energy distribution between the brake power, friction losses, jacket cooling load, and exhaust sensible heat as shown in Table 2 (Shivapuji, 2015).

Table 1: Experimental and modelled values for brake power output at different engine loads (Shivapuji & Dasappa, 2013)

Measured brake power (experiment)	27.32 kW	16.41 kW	5.23 kW
Calculated brake power (simulation)	27.25 kW	16.83 kW	5.30 kW
Relative error	0.26%	2.56%	1.34%

Table 2: Experimental and modelled energy distribution for an engine with 123.5 kW input energy rate (Shivapuji, 2015)

	Brake power	Cooling load	Exhaust sensible heat	Friction losses
Measured output (experiment)	22.1%	31.9%	26.2%	14.8%
Calculated output (simulation)	20.0%	32.5%	30.8%	15.6%

Engine simulation was set up by maintaining a constant mass flow of the syngas mixture while varying the mass flow air from sub-stoichiometric conditions to an excess air-fuel ratio of $\lambda=2.15$, representing the lean limit for syngas combustion (Dasappa, Sridhar & Paul, 2011). Intake conditions were set to constant temperature and pressure conditions of 298 K and 100 kPa, respectively.

To simulate the effect of using CO_2 as a substitute diluent instead of excess air, a mass flow stream of CO_2 was introduced into the intake mixture while a stoichiometric airflow was maintained. The mass flow rate of CO_2 was increased for successive iterations of the model. For comparison to excess air cases, an equivalent λ parameter is calculated for the CO_2 cases such that the mass flow of air remains at the stoichiometric flow rate while the CO_2 mass flow substitutes the equivalent mass flow of excess air. Equation (6) illustrates the calculation.

$$\lambda_{eq} = \left(1 + \frac{\dot{m}_{CO_2}}{\dot{m}_{air,st}}\right) \tag{6}$$

Under stoichiometric combustion conditions, the peak cylinder temperature is calculated to be 1941.4 K. For engine operation at excess air of $\lambda=2.0$, peak temperature is limited to 1519.1 K. This falls to a peak temperature of 1470.3 K for an equivalent CO_2 dilution of $\lambda_{eq}=2.0$.

By substituting CO_2 for excess air as the engine diluent gas, the indicated mean effective pressure at high λ ratios remains slightly higher, indicating more available work per engine cycle. For $\lambda_{(eq)}=1.50$, this translates to a 1% increase in IMEP with a trend that rises to 2% for $\lambda_{(eq)}=2.15$.

Additionally, lower cylinder cooling loads are required and exhaust gas temperatures remain higher under the CO_2 diluent regime. For example, at $\lambda_{(eq)}=2.0$ the cooling load reduces from 37.48 kW in excess air to 36.56 kW in CO_2 while exhaust temperatures are 629.7 K and 688.2 K, respectively. This is attributable to the increased heat capacity of the exhaust gases since they have a higher CO_2:N_2 ratio. Such an effect is beneficial for potential down stream processes like removal of pollutants or CHP applications.

References:

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