CHAPTER 6

Results and Discussion

6.1 Coefficient of Variation

A COV is a measure of cyclic variability that occurs during early stage of combustion and around peak pressure. Figure 6.1 shows COV of BMEP as a function of engine speed and load with bath tub and cavity combustion chambers at a 14:1 CR. For each speed, the ignition timing was adjusted to maximum brake torque (MBT) timing. The COV of BMEP in both combustion chambers was found to vary between 1.75 and 3.0%. Minimum COV of bath tub and cavity combustion chambers occurred at 1300 rpm and 1500 rpm, respectively. At higher engine speeds, the COV of both combustion chambers was found to increase, but remained small. Increase in COV was due to difference in cycle-to-cycle combustion process caused by variations in mixture motion in the cylinder, the mixing of air-producer gas and residual gas in cylinder for each cycle (Heywood, 1989). In comparison of operation loads, full load appeared to show higher COV than part load.

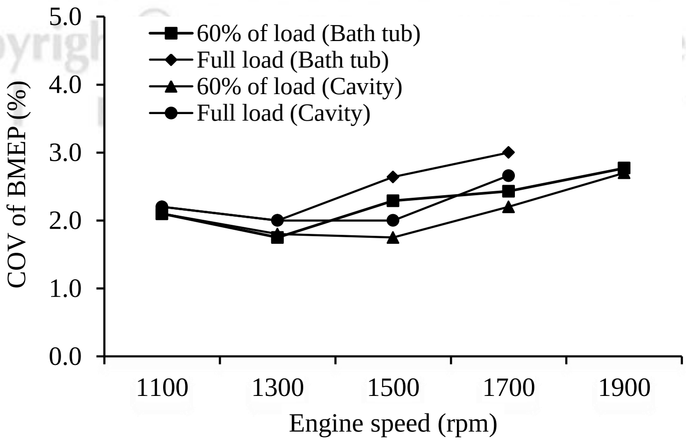


Figure 6.1 COV of the producer gas engine at various engine speeds

6.2 Performance of the Modified Engine

6.2.1 Effect of load

Figure 6.2 shows the variation in engine torque of the small producer gas engine at 1500 rpm with engine loads, CR and combustion chamber type. Maximum engine torques of bath tub and cavity combustion chambers were obtained at 15.38 Nm and 15.34 Nm, respectively. Both occurred at 14:1 CR on full load. For all CRs and both combustion chambers, the brake torques were similar between 20–60 % of load. Increasing load from 60 to 80% at medium CR increased brake torque significantly. The main reason for the increase in torque was that the work in expansion stroke power exceeded the work in the compression stroke, compared to low CR (Raman et al, 2013). At a higher CR, engine torque was low due to abnormal combustion, leading to knocking (Wise, 2005).

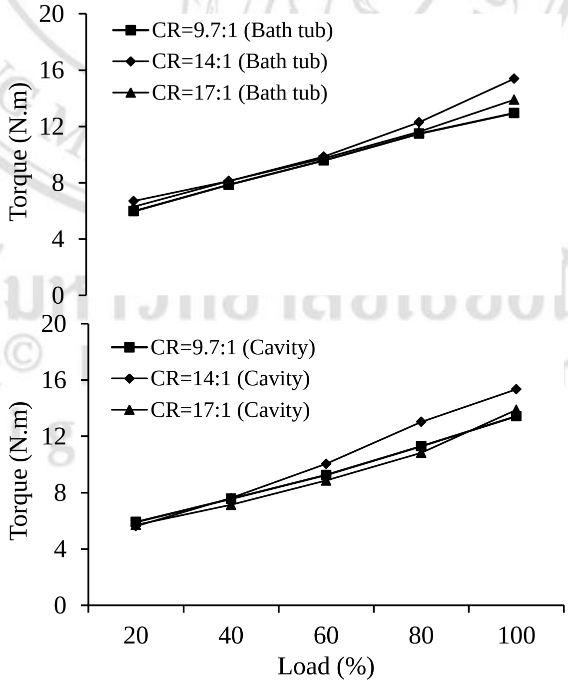


Figure 6.2 Effect of load and CR on the engine torque

Figure 6.3 shows the effect of loads on the brake power for each CR and combustion chamber considered. The engine brake power was found to increase as engine load increased for all CRs and combustion chambers. The maximum brake power of both combustion chambers occurred at 14:1 CR on full load. Both combustion chambers were equal and achieved 2.41 kW.

Figure 6.4 shows brake thermal efficiency as a function of engine load and combustion chamber for different CRs. The thermal efficiency of both combustion chambers tended to increase with engine load. This may be attributed to better combustion of the relatively rich gas-air mixture at high loads. The thermal efficiency of the medium CR was slightly higher than those at low and high CRs. Reduction of brake thermal efficiency was due to higher producer gas flow rates and poor combustion. The thermal efficiency of cavity combustion chamber was higher than bath tub combustion chamber due to low fuel consumption rate and complete combustion in the cavity combustion chamber. With the medium CR of cavity and bath tub combustion chambers, the maximum brake thermal efficiency of 20.0% and 18.6% were obtained at full load, respectively.

Figure 6.5 shows effect of load and CR on brake specific fuel consumption. The charcoal to gas conversion rate was arrived at by measuring the gas flow rate and fuel consumption rate. Increasing load of the engine led to reduced specific charcoal consumption rate, for both combustion chambers. The cavity combustion chamber used less charcoal, compared to the bath tub. The specific charcoal consumption rate of the small producer gas engine with cavity and bath tub combustion chambers were 0.87 kg/kWh and 0.94 kg/kWh, respectively. The use of low and high CRs consumed more fuel than medium CR. Generally, the specific fuel consumption rate of the producer gas engine was in a range of 1.2–2 kg/kWh (Dasappa et al, 2011), (Aung, 2008).

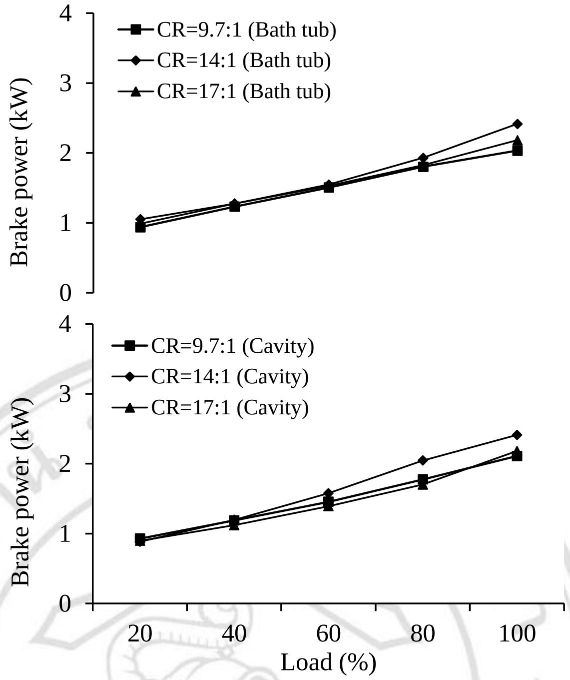


Figure 6.3 Effect of load and CR on the brake power

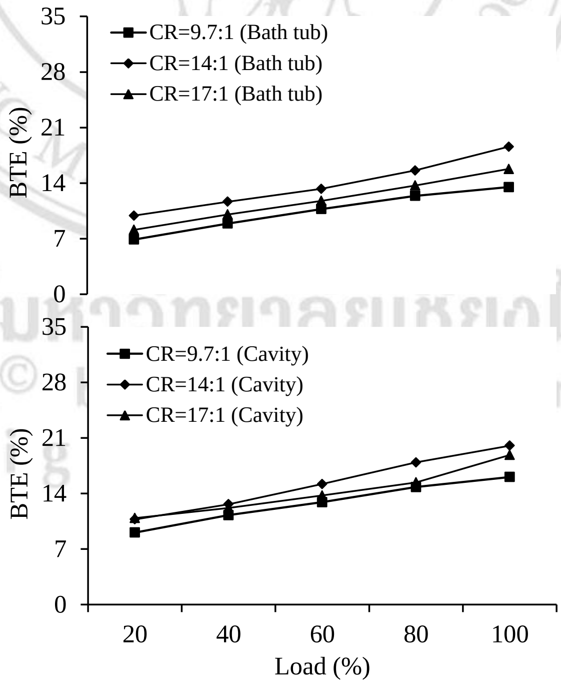


Figure 6.4 Effect of CR and load on the BTE

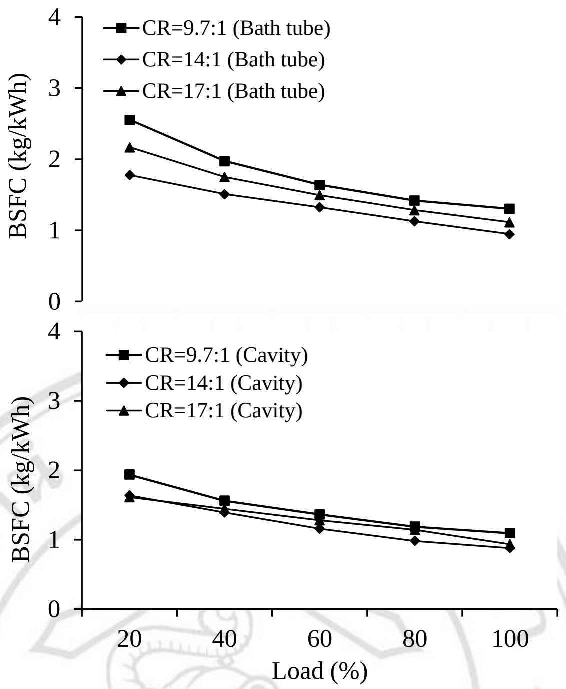


Figure 6.5 Effect of CR and load on the BSFC

6.2.2 Effect of engine speed

Figure 6.6 shows the variation in engine torque of the small producer gas engine at full load, with varying engine speed, CR and combustion chamber type. The engine torque increased with engine speed for all CR and both combustion chambers. At medium CR with both combustion chambers, higher engine torque was observed. The use of a bath tub combustion chamber in low and high CRs had an effect on engine operation at high speed. It was observed that, at a low compression ratio (CR = 9.7:1), the small producer gas engine was able to be gradually loaded and stabilized up to 1500 rpm. Acceleration was good, and the engine power increased with increasing engine speed. The engine slowed and became unstable when trying to increase to 1700–1900 rpm. The observed deceleration may be due to reduction in energy density, compared to gasoline. Producer gas's energy is less dense than gasoline (Sridhar et al, 2000, Shah et al, 2010). Low CR of the small producer gas engine may cause lower pressure inside the combustion chamber (Kassaby et al, 2013), and affect flammability of the producer gas (Ando et al, 2005). Lower volumetric efficiency may be reduced for gas fuel operations, compared to conventional liquid fuels (Mustafi et al, 2006). In the high compression ratio (CR = 17:1), the small producer gas engine produced knocking which started between 1700 - 1900 rpm and 80–100% of full load and heavy knocking at full load and 1900 rpm. However, the small engine operated well between 1100 – 1500 rpm. Knocking in engine may result from increasing the compression ratio, load and engine speed, leading to an increase in gas density, temperature, and ignition lag in the combustion chamber (Heywood, 1989). Comparing maximum engine torque, the bath tub combustion chamber provided higher engine torque than the cavity combustion chamber. The maximum engine torques of bath tub and cavity combustion chamber were 18.61 Nm and 18.05 Nm, respectively, with both occurred at 1700 rpm and medium CR.

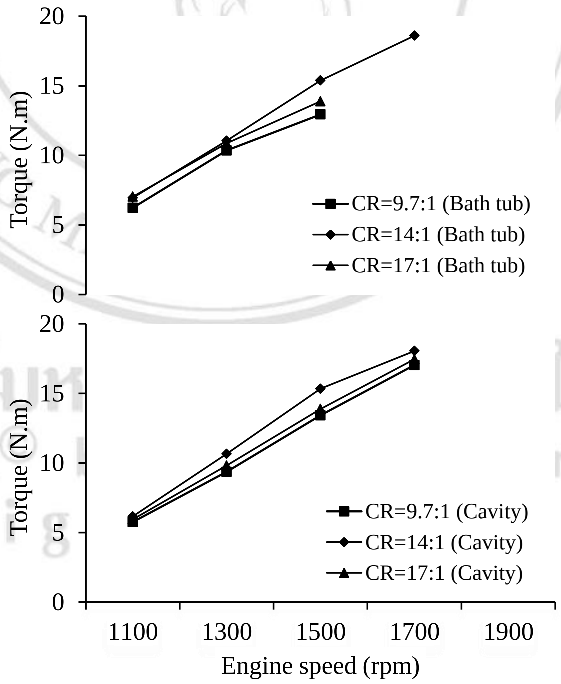


Figure 6.6 Effect of engine speed on the engine torque

Figure 6.7 shows the variation of brake power at full load on varying engine speed, CR and combustion chamber type. The brake power of small producer gas engine increased with engine speed. The maximum brake power at medium compression ratio and with a bath tub combustion chamber was higher than that from the cavity combustion chamber. The maximum brake powers of bath tub and cavity combustion chamber were 3.31 kW and 3.10 kW, respectively, with both occurring at 1700 rpm

Figure 6.8 shows brake thermal efficiency as a function of engine speed and combustion chamber for different CRs. Increasing the engine speed of producer gas engine led to increased thermal efficiency. Using the cavity combustion chamber in the small producer gas engine, it has a higher thermal efficiency than bath tub combustion chamber. At medium CRs of both combustion chambers have a high brake thermal efficiency. The maximum thermal efficiency of cavity and bath tub combustion chambers was 23.9% and 18.7% respectively at 1700 rpm.

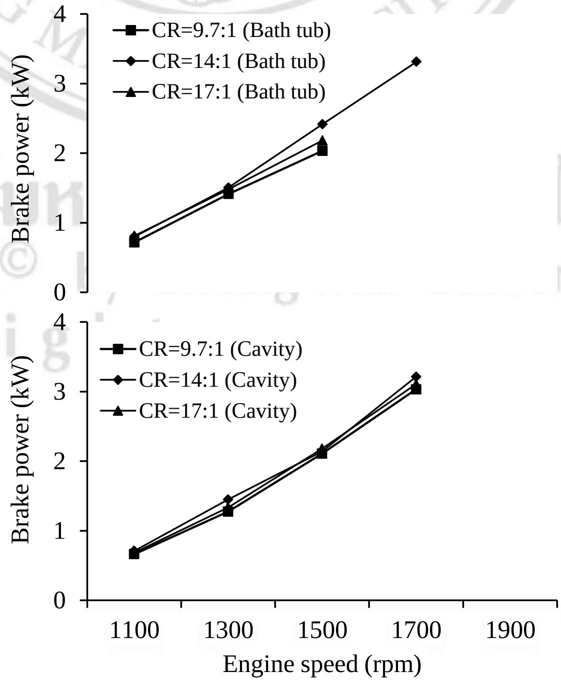


Figure 6.7 Effect of engine speed on the brake power

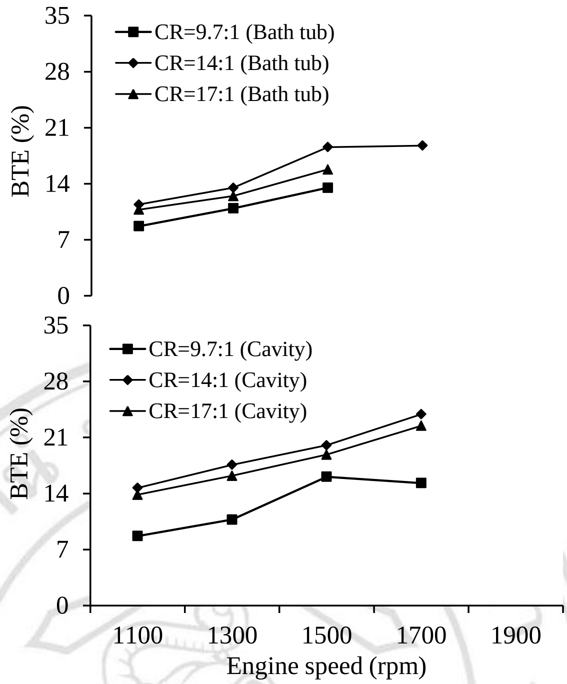


Figure 6.8 Effect of engine speed on the BTE

Figure 6.9 shows the brake specific fuel consumption as a function of engine speed and combustion chambers for different CRs. Increasing engine speed reduced fuel consumption. The medium CR and cavity combustion chamber had the lowest fuel consumption. The minimum brake specific fuel consumption of cavity and bath tub combustion chambers of 0.74 kg/kWh and 0.94 kg/kWh was obtained at 1700 rpm respectively.

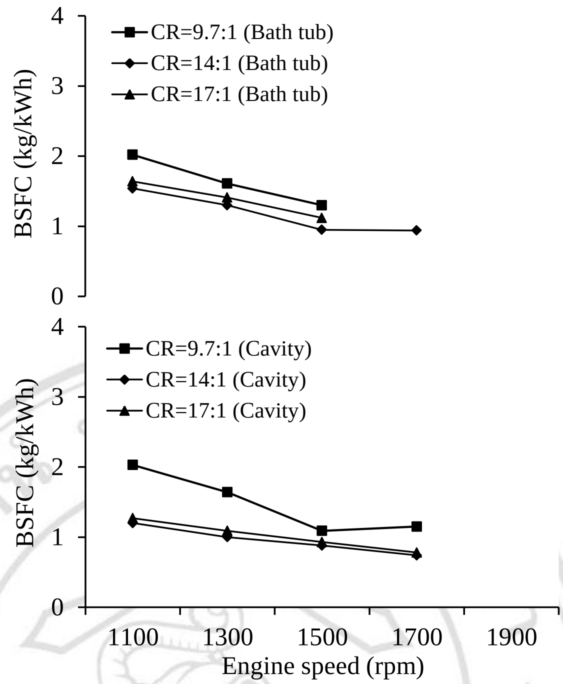


Figure 6.9 Effect of engine speed on the BSFC

6.2.3 Effect of ignition timing

Figure 6.10 shows the effect on a small producer gas engine of changing ignition timing on an engine under full load. The results show that BMEP tended to increase with appropriate advance of ignition timing that mostly depended on engine speed and load. Except at 1500 rpm on full load, the small engine exhibited deceleration when adjusted to lower than 35˚ BTDC ignition timing. Retarding ignition timing, the air-fuel mixture in the cylinder will burn as the piston is moving down, leading to decreasing pressure and performance (Kakaee et al, 2011), (Zareei et al, 2013). When advancing ignition timing, the mixer in the cylinder will burn while the piston is moving up in compression stroke. In the cavity combustion chamber, the result of retarding and advancing ignition timing was similar to the bath tub combustion chamber. The cavity combustion chamber needed the ignition timing to be advanced more than the bath tub combustion chamber at all engine speeds. The best ignition timing found in the bath tub combustion chamber was 25˚ BTDC at 1100 rpm, 30˚ BTDC at

1300 rpm, 35° BTDC at 1500 rpm, 40° BTDC at 1700 rpm. At 1900 rpm, the engine showed knocking. The best ignition timing cavity combustion chamber was 35° BTDC at 1100 rpm, 40° BTDC at 1300 rpm, 45° BTDC at 1500 rpm, 45° BTDC at 1700 rpm. At 1900 rpm the engine showed knocking in both combustion chambers. The maximum BMEP of bath tub and cavity combustion chamber were 443.10 kPa and 379.13 kPa, respectively and that occurred at 1700 rpm. Using producer gas in small engines adjusted to suitable ignition timing, high BTE can be obtained. Adjusting the ignition timing caused the combustion process in the cylinder to directly affect the power output and fuel consumption.

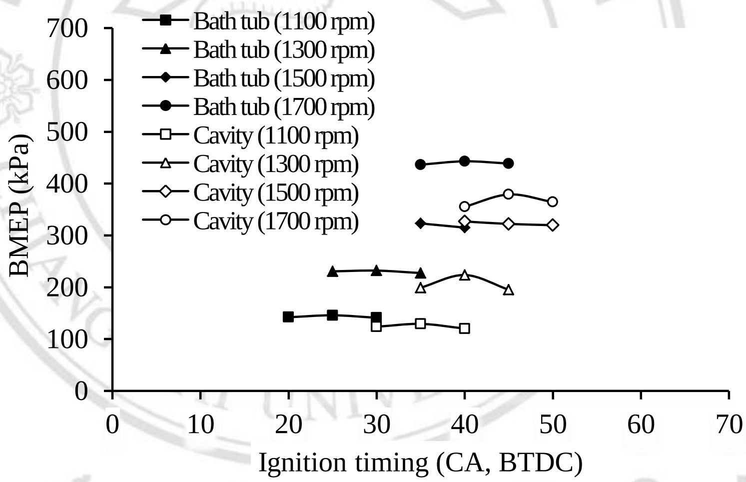


Figure 6.10 Effect of ignition timing on the brake mean effective pressure

Using producer gas in small engines adjusted to suitable ignition timing, high BTE can be obtained. Adjusting the ignition timing caused the combustion process in the cylinder to directly affect the power output and fuel consumption. Figure 6.11 shows BTE as a function of ignition timing and engine speed at full load of the small producer gas engine using the bath tub and cavity combustion chambers. For the bath tub combustion chamber, the maximum BTE was 18.8% but lower than the cavity combustion chamber of 23.9%, this was achieved at 1700 rpm on full load for both combustion chambers. Comparison with a similar magnitude in medium and large engines was made. Typical thermal efficiency of large producer gas engines was in a range of 18-24 % (Dasappa et al, 2011, Sridhar et al, 2000, Raman et al, 2013).

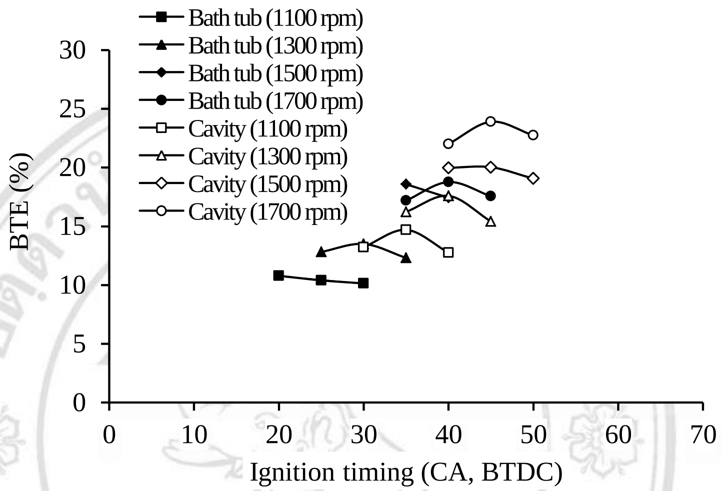


Figure 6.11 Effect of ignition timing on BTE

Figure 6.12 shows the variation of BSFC with adjusted ignition timing, engine speed and load of the small producer gas engine in the bath tub combustion chamber. The BSFC rate tended to decrease with increasing ignition timing. In comparison between different loads and engine speeds, the minimum BSFC rate occurred on full load operation and 1700 rpm of engine speed. Increasing engine speed tended to decrease BSFC rate. The lowest BSFC rate for bath tub combustion chamber was 0.93 kg/kWh while the cavity combustion chamber was 0.74 kg/kWh.

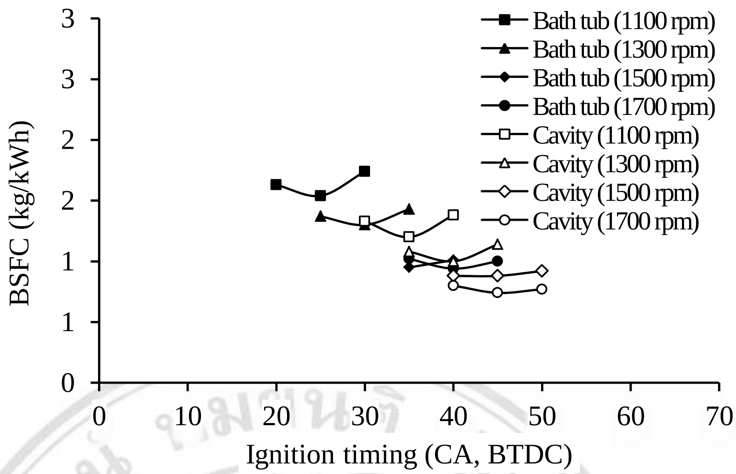


Figure 6.12 Effect of ignition timing on the BSFC

6.2.4 Optimum ignition timing

With the small producer gas engine, to increase the engine speed it was necessary to increase the ignition timing to a higher than existing conventional engines. Figure 6.13 shows the summary of optimum ignition timing of the small producer gas engine in bath tub and cavity combustion chambers at each engine speed on part and full loads. The ignition timing tended to increase with engine speed because the air-producer gas mixture in cylinder was more turbulent due to faster moving gas. The burning time became shorter at higher engine speeds. So, it was necessary to increase the burn duration by altering the timing. The timing of the bath tub combustion chamber, at 1100 rpm, to give maximum power output occurred between 20° to 25° BTDC. The engine speed of 1500 rpm was important because most applications use this speed. The best power output was between 32.5° to 37.5° BTDC for 1500 rpm. It should be noted that when adjusted to 40° BTDC of ignition timing advance, the power output was reduced. For the cavity combustion chamber, at 1100 rpm, maximum power output was between 30° to 35° BTDC, and 40° to 45° BTDC in 1500 rpm. At 1900 rpm, the small producer gas engine was unable to operate at full load due to deceleration and knocking. Whereas, adjusting timing at 60% of load or lower, the engine had good acceleration stability. Therefore, the best power output of bath tub combustion chamber occurred between 40˚ to 45˚ BTDC, and cavity combustion chamber was in a range of 50˚ to 55˚ BTDC.

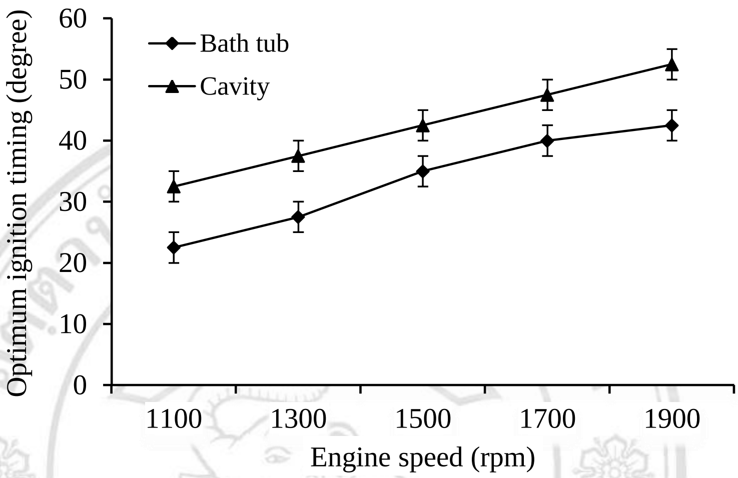


Figure 6.13 Optimum ignition timing of the small producer gas engine

6.3 Emission of Small Producer Gas Engine

CO emission of producer gas engine is generally lower than gasoline engines. (Munoz et al, 2000), (Saridemir et al, 2012). Figure 6.14 shows the variation of CO emissions from producer gas engine in bath tub and cavity combustion chambers with varying load and engine speed. Increasing load led to slightly reduced CO emission. Reduction of CO emission at a high load was due to more complete combustion, while increase in CO emission at high engine speed was caused by incomplete combustion. In a comparison of both combustion chambers, the use of bath tub combustion chamber showed a higher CO emission than cavity combustion chamber. For the cavity combustion chamber’s squish area, the burning velocity was higher, causing complete combustion and thus reduced CO emission (Munoz et al, 2000). Minimum CO emission of 0.28% was achieved at 1500 rpm on full load and with the cavity combustion chamber. The maximum

CO emission of 1.9% was observed at lowest engine speed with bath tub combustion chamber.

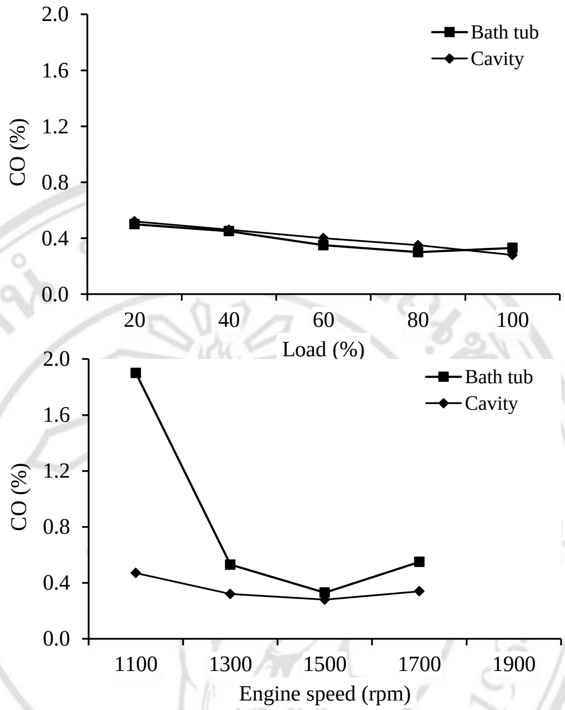
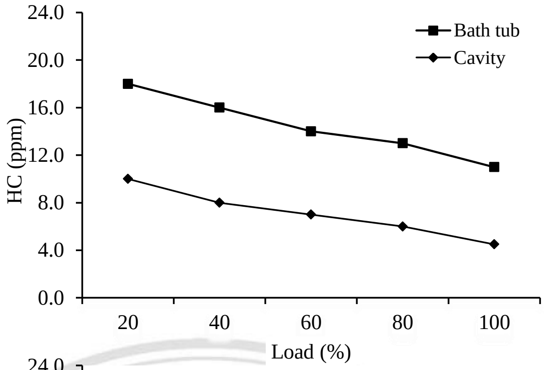


Figure 6.14 CO emission of the small producer gas engine

Figure 6.14 shows the hydrocarbon emissions of both combustion chamber with varying load and engine speed. The HC emissions tended to decrease with increasing load and engine speed. The use of the cavity combustion chamber produced less HC emissions than the bath tub combustion chamber. Reduction of HC emissions may be due to efficiency losses at low loads and incomplete combustion in the engine. Minimum HC emission of 3.5 ppm was achieved at 1700 rpm on full load and occurred in the cavity combustion chamber. The maximum HC emission of 20 ppm was obtained at 1100 rpm in bath tub combustion chamber. However, a comparison against the gasoline engine, the HC emissions of the producer gas engine was lower (Saridemir et al, 2012).



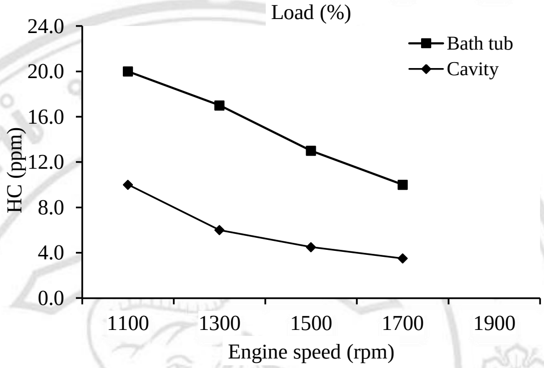


Figure 6.15 HC emission of the small producer gas engine

Figure 6.16 shows the noise of small producer gas engine with varying loads and engine speed. The noise of both combustion chambers increased with engine load and speed (Yasar, 2010). The use of bath tub combustion chamber was noisier than the cavity combustion chamber. The average increase in noise level was less than 3%. The maximum noise in the bath tub and the cavity combustion chamber were 96.3 and 94.5 dB, respectively.

Finally, the smoke density of small producer gas engine was very low. Measurement of smoke density could only be done when changing the load and engine speed.

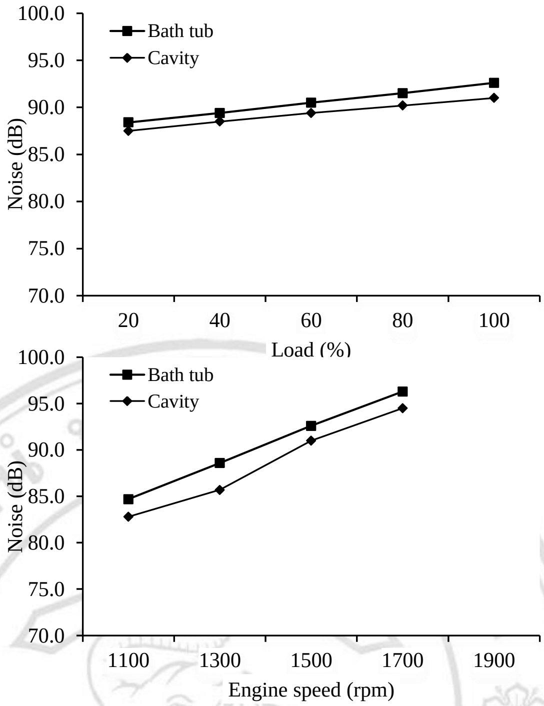


Figure 6.16 Noise produced by the small producer gas engine

6.4 Comparison with Diesel Engines

Figure 6.17 shows the engine torque of producer gas engine at varying load and engine speed, compared to the original diesel engine. The engine torque of the producer gas engine and diesel engine tended to be similar with change of load. At higher engine speeds above 1500 rpm there was increased engine torque in diesel engine. Increase in engine torque in the diesel engine was due to a plentiful supply of oxygen available, leading to more complete combustion (Lekpradit et al, 2009). Maximum engine torque, for producer gas engine, of 18.61 Nm was achieved at 1700 rpm in bath tub combustion chamber on full load, while the diesel engine obtained 39.2 Nm at 1900 rpm on full load.

Figure 6.18 shows the brake power of the producer gas engine at varying loads and engine speeds compared to the original diesel engine. For a fixed speed, the brake power of producer gas engine and diesel engine had similar value. Maximum brake power for the producer gas engine of 3.31 kW was obtained at 1700 rpm, while the diesel engine was 7.80 kW at 1900 rpm.

Figure 6.19 shows the brake thermal efficiency of the producer gas engine at varying loads and engine speeds, compared to the original diesel engine. The BTE of diesel engine increased with load and engine speed. But, the BTE of diesel engine was higher than the producer gas engine. The BTE of diesel engines increased steadily over 1900 rpm while the highest thermal efficiency of the producer gas engine was at speed of 1700 rpm. Over this speed, knocking was occurred. Maximum BTE for the producer gas engine was 23.9% while the diesel engine was obtained 31.02 kW at 1900 rpm. Comparison of specific fuel consumption of both engines could not happen, due to different fuel types.

Figure 6.20 shows BSEC of producer gas engine at varying load and engine speed compared to the original diesel engine. The BSEC decreased with increasing load and engine speed. The BSEC of diesel engine was lower than the producer gas engine at all loads and engine speeds. The minimum BSEC of the diesel engine occurred at higher engine speeds at full load where 11.60 MJ/kWh was obtained. For the producer gas engine, the minimum BSEC occurred in cavity combustion chamber on full load at 1700 rpm where 15.07 MJ/kWh was obtained.

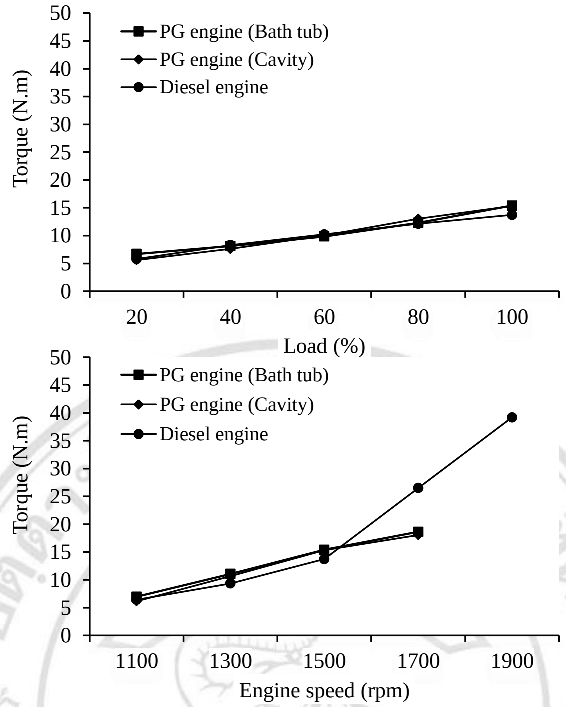


Figure 6.17 Comparing engine torque of producer gas engine with the diesel engine

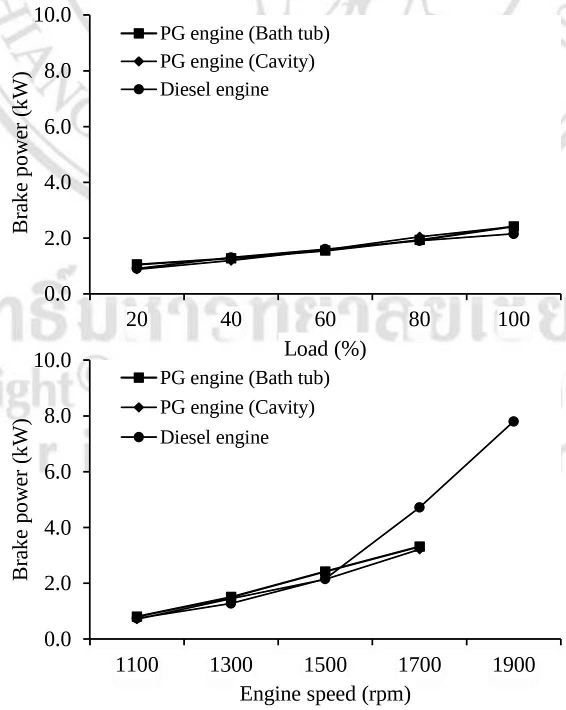


Figure 6.18 Comparing brake power of producer gas engine with the diesel engine

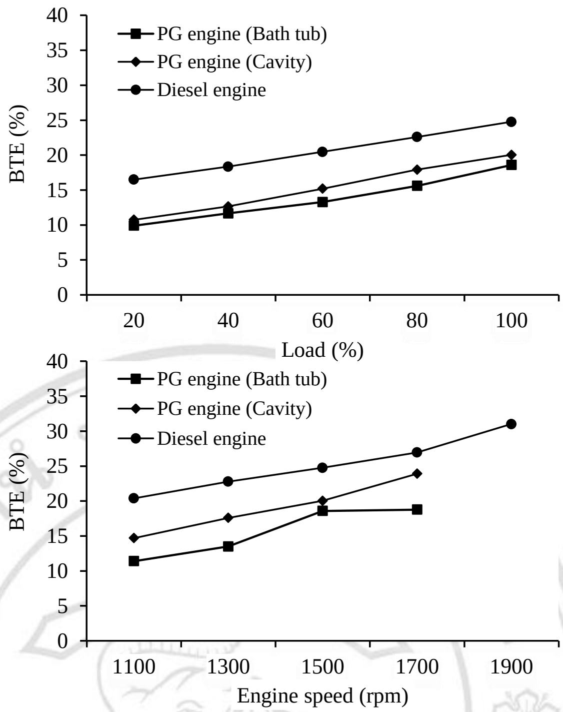


Figure 6.19 Comparing BTE of producer gas engine with the diesel engine

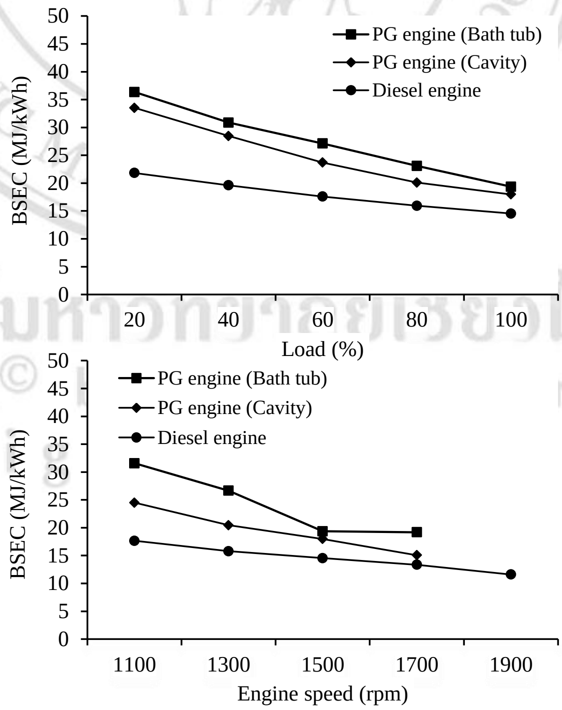


Figure 6.20 Comparing BSEC of producer gas engine with the diesel engine

Figure 6.21 shows the comparison of the CO emissions of the producer gas engine with original diesel engine at varying load and engine speed. CO emission of the producer gas engine was found to be higher than the diesel engine for all loads and speeds. Higher CO in the exhaust was due to insufficient oxygen for combustion. CO emission from the producer gas engine were slightly reduced with increasing load and speed, while CO from diesel engine was stable when loads were in the range of 60-100%. Reduction of CO emission at high load was due to more complete combustion. Minimum CO emission of diesel engine of 0.01% was achieved at full load while the small producer gas engine in cavity and bath tub combustion chamber on full load was obtained 0.28% and 0.33% respectively. However, CO emission of small producer gas engine was less than gasoline engine by ten times. (Saridemir et al, 2012).

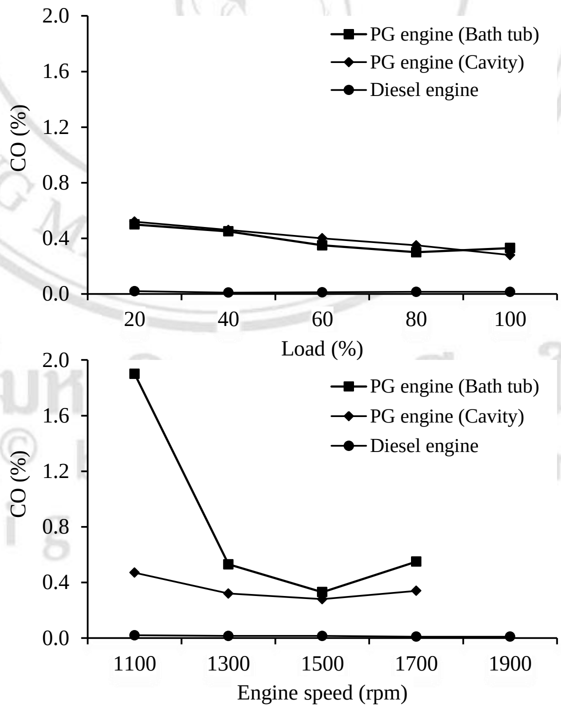


Figure 6.21 Comparison CO of producer gas engine with the diesel engine

Figure 6.22 shows the comparison of hydrocarbon emissions of the producer gas engine with original diesel engine at varying loads and engine speeds. Hydrocarbon emissions of diesel engine were less than the small producer gas engine at all engine speeds and loads, while slightly increased HC emissions on high load, due to the incomplete combustion, a consequence of lean flame-out (Hussain et al, 2012). Considering various engine speeds, the HC emissions of the diesel engine were slightly reduced with increasing engine speed. Minimum HC emission of the diesel engine, of 3 ppm, was achieved at full load while the small producer gas engine in cavity and bath tub combustion chamber on full load was 3 ppm and 10 ppm respectively. Comparing the gasoline engine, and the producer gas engine, the HC emission of the producer gas engine was ten times less than the gasoline engine.

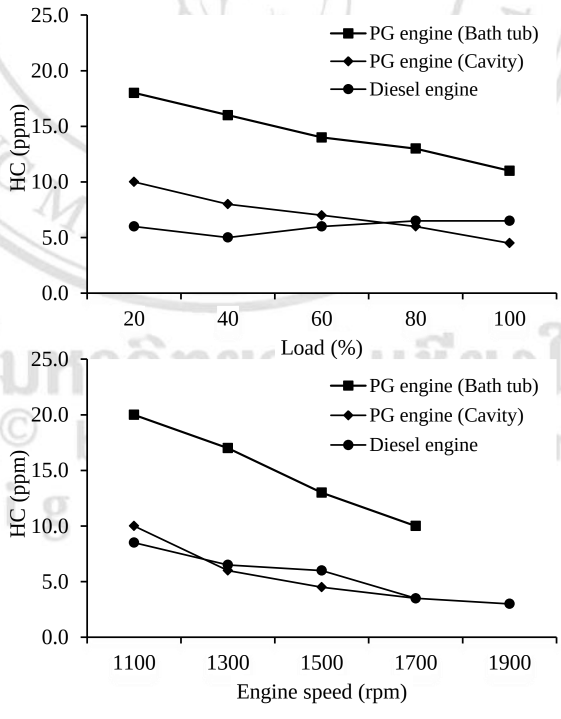


Figure 6.22 Comparison HC of producer gas engine with the diesel engine

Figure 6.23 shows comparison of the smoke density of the producer gas engine with the original diesel engine at varying loads and engine speeds. Smoke density of diesel engine is higher than the small producer gas engine significantly and increases steadily with increasing loads and engine speed. Between 1100-1300 rpm of diesel engine, smoke density is constant and rapidly increases between 1300-1500 rpm and after that the trend is constant again. However, the smoke density of small producer gas engine was relatively constant and very low. The rise of smoke density on increase load and engine speed was due to incomplete combustion of fuel in cylinder, a consequence of insufficient oxygen and less time for combustion (Hussain et al, 2012). The highest smoke density of diesel engine and small producer gas engine of 11.60% and 1.9% were obtained respectively.

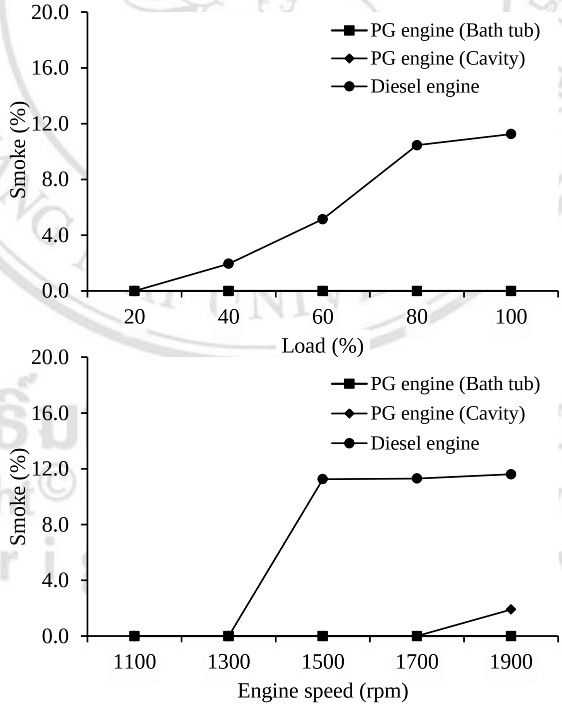


Figure 6.23 Comparison of smoke density of producer gas engine with the diesel engine

Figure 6.24 shows noise comparison of the small producer gas engine with original diesel engine on varying loads and engine speeds. The noise of diesel engine is constant with increase load while the producer gas engine, noise increased with engine load. In low load and low engine speed, the noise of diesel engine is higher than producer gas engine, but the diesel's noise is less than producer gas engine with high load and engine speed. The noise of diesel engine and producer gas engine is similar between of 94.5-96.7 dB.

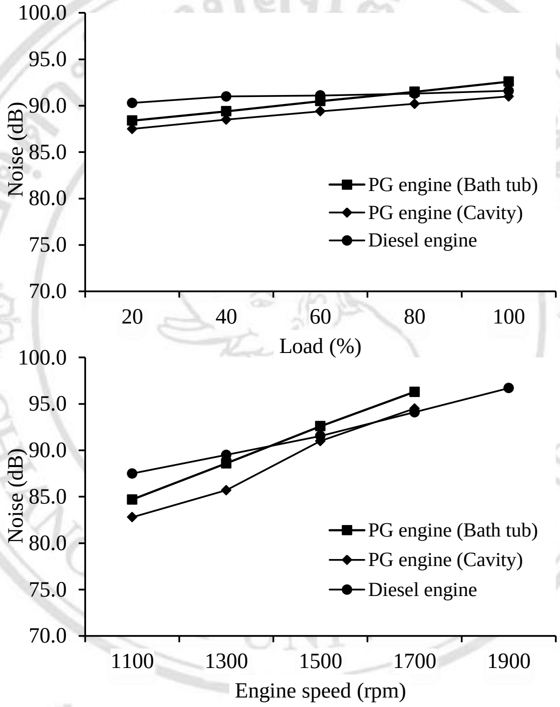


Figure 6.24 Comparison of the noise of producer gas engine with the diesel engine

6.5 Operation Limit of Small Producer Gas Engine

Table 6.1 and 6.2 shows operation limit of the small producer engine with bath tub and cavity combustion chambers. The bath tub combustion chamber can operate at low CR (9.7:1) and remain stable up to 1500 rpm after that the engine decelerated. The cause of the engine deceleration was due to the characteristics of an unsuitable fuel and its low volume efficiency. At medium CR (14:1), the small producer engine was able to operate at almost all experimental conditions, except at 1900 rpm on full load, where the engine knocked. At higher CR (17:1), the producer gas engine was able to operate in a range of 1100-1500 rpm. Between of 1700-1900 rpm, the engine was unable to operate at 80% and full loads, due to knocking. The cavity combustion chamber, the use of low CR (9.7:1) could be operated at all loads and engine speeds. At medium CR (14:1), the engine could operate with either combustion chamber. At higher CR (17:1), the producer gas engine was able to operate at all experiments conditions, except at 1900 rpm on 80-100% of load. The engine was found to knock with the cavity combustion chamber. For comparison of operation limits with both combustion chambers, the cavity combustion chamber had a wider range of applications. Squish area helped to increase burning velocity and reduced knock eventually.

Table 6.1 Limitation of producer gas engine in bath tub combustion chamber

Compression ratio	Load (%)	Engine operation				
		1100 rpm	1300 rpm	1500 rpm	1700 rpm	1900 rpm
	20	✓	✓	✓	x	x
	40	✓	✓	✓	x	x
9.7:1	60	✓	✓	✓	x	x
	80	✓	✓	✓	x	x
	100	✓	✓	✓	x	x
	20	✓	✓	✓	✓	✓
	40	✓	✓	✓	✓	✓
14:1	60	✓	✓	✓	✓	✓
	80	✓	✓	✓	✓	✓
	100	✓	✓	✓	✓	xx
	20	✓	✓	✓	✓	✓
	40	✓	✓	✓	✓	✓
17:1	60	✓	✓	✓	✓	✓
	80	✓	✓	✓	xx	xx
	100	✓	✓	✓	xx	xx

(✓) OK, (x) Erratic, (xx) Knocking

Table 6.2 Limitation of producer gas engine in cavity combustion chamber

Compression ratio	Load (%)	Engine operation				
		1100 rpm	1300 rpm	1500 rpm	1700 rpm	1900 rpm
9.7:1	20	✓	✓	✓	✓	✓
	40	✓	✓	✓	✓	✓
	60	✓	✓	✓	✓	✓
	80	✓	✓	✓	✓	✓
	100	✓	✓	✓	✓	✓
14:1	20	✓	✓	✓	✓	✓
	40	✓	✓	✓	✓	✓
	60	✓	✓	✓	✓	✓
	80	✓	✓	✓	✓	✓
	100	✓	✓	✓	✓	xx
17:1	20	✓	✓	✓	✓	✓
	40	✓	✓	✓	✓	✓
	60	✓	✓	✓	✓	✓
	80	✓	✓	✓	✓	xx
	100	✓	✓	✓	✓	xx

(✓) OK, (x) Erratic, (xx) Knocking

6.6 Prediction of Small Producer Gas Engine's Performance

In this study, a small producer gas engine model was developed to estimate torque, brake power, thermal efficiency and specific fuel consumption. The simulated results were compared against the experiment. They are shown in Figures 6.25-6.26. At low engine speeds, the estimated values were similar to the experimental results. At higher speeds, there were small differences between engine speeds of 1500-1900 rpm. This may be attributed to deviation in the producer gas flow rate when entering the cylinder. The producer gas flow rate was derived empirically from the sum of fuel consumption and volumetric efficiency. However, the deviations were likely due to other factors such as pressure and temperature in cylinder in the combustion process, etc. The average errors of brake power, engine torque, BTE and BSFC were -3.30%, -3.32%, -6.50 and 3.07%, respectively. Therefore, it was concluded that the developed mathematical model gave good agreement and can be applied to the small producer gas engine under the same conditions. For comparison, the use of the thermodynamic model to internal combustion engines is summarized in Table 6.3. The model validates the three engines were four stroke SI engine operated on gasoline and gasoline/ethanol blend. The mean errors of both engines were in a range of -0.12–7.63 %. This mathematical modeling was acceptable when compared to the experimental results. The mathematic modeling of this work can predict performance of the SI engine operated on producer gas engine well.

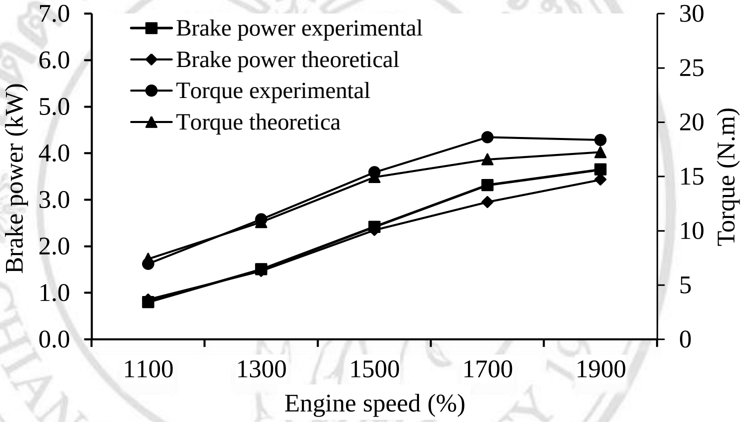


Figure 6.25 Theoretical and experimental results of brake power and torque

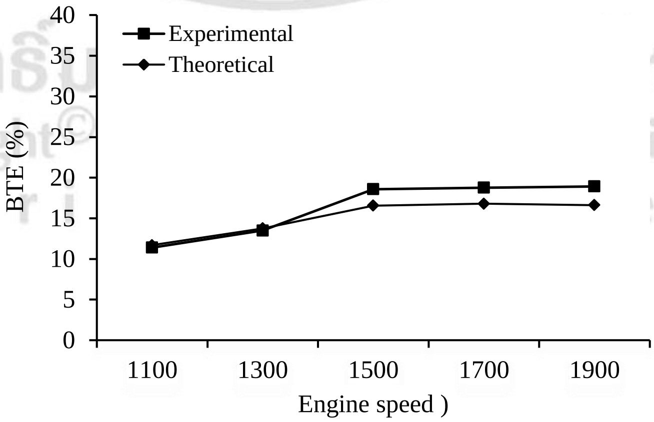


Figure 6.26 Comparison between theoretical and experimental results of BTE

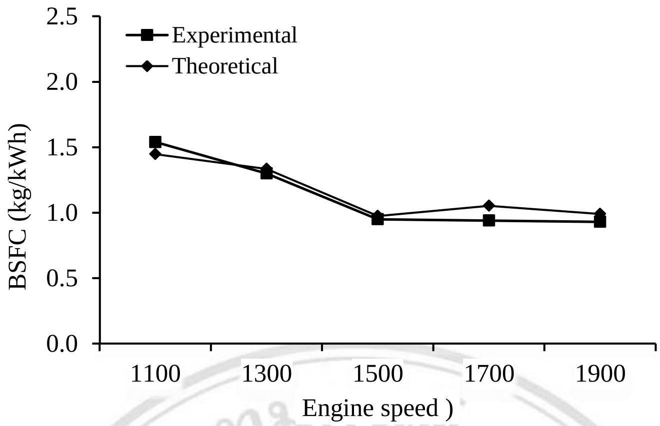


Figure 6.27 Comparison between theoretical and experimental results of BSFC

Table 6.3 Mean percentage error of the thermodynamics model with a SI engine

Comparison research	Mean percentage error (%)			
	Brake power	Torque	BTE	BSFC
This work	-3.30	-3.32	-6.50	3.07
Hatte et al.,2012	7.63	-	0.06	-0.12
Sitthiracha,2006	-2.74	-3.14	-	-
Chaudhari et al., 2014	23.08	-	21.83	-

6.7 Fuel Cost of Small Producer Gas Engine

Fuel cost of producer gas and diesel engines to produce electricity from various biomasses is shown in Table 6.4. The biomass used were charcoal and longan wood. The data came from experimental engine based on various loads, engine speed, spark timing and combustion chamber types. The suitable engine speed for producer gas engine was 1700 rpm at full load. Spark timing of bath tub and cavity combustion chamber were 40° BTDC and 45° BTDC, respectively, at full load. The price of diesel, charcoal and longan wood were 30.74 Baht/liter, 10 Baht/kg and 1.5 Baht/kg, respectively. The fuel cost of diesel engine was higher than the producer gas engine in a range of 1.15-1.47 times with charcoal, and 5.34-6.81 folds with longan wood. The use of charcoal produced producer gas to reduce tar removal processes, but was rather expensive compared to longan wood. With the use of charcoal as a fuel, tar removal was needed, similarly to longan wood. On the other hand, the use of wood may alter some results. Therefore, it can be seen that the fuel cost depend on the type of biomass. The fuel cost for a generator with diesel was 10.90 Baht/kWh, while the charcoal cost was between of 7.4-9.4 Baht/kWh. The use of longan wood lowered fuel costs within a range of 1.6-2.0 Baht/kWh.

Table 6.4 Fuel costs of producer gas fueled and diesel fueled generator with charcoal and longan fuel

Item	Diesel	Bath tub (Charcoal)	Cavity (Charcoal)	Bath tub (Longan wood)	Cavity (Longan wood)
Compression ratio	21:1	14:1	14:1	14:1	14:1
Engine speed (rpm)	1700	1700	1700	1700	1700
Spark timing (Degree)	25	40	45	40	45
Load (%)	100	100	100	100	100
Brake power (kW)	4.71	3.31	3.21	3.31	3.21
Brake torque (N.m)	26.49	18.61	18.05	18.61	18.05
Brake thermal efficiency (%)	26.95	18.77	23.9	18.77	23.90
Fuel consumption (kg/hr)	1.38	3.1	2.36	3.42	4.5
Heating value (kJ/kg)	42500	28550	28550	20590	20590
Gas flow rate (m^3/hr)	-	13.67	10.42	13.67	10.42
Gas calorific (MJ/Nm^3)	-	4646.17	4646.17	4646.17	4646.17
Cold gas efficiency (%)	-	71.4	71.4	68.5	68.5
Brake specific fuel consumption (kg/kWh)	0.293	0.94	0.74	1.36	1.07
Fuel price (Baht/kg)	30.74	10	10	1.50	1.50
Fuel cost (Baht/kWh)	10.90	9.4	7.40	2.04	1.60