CHAPTER 1

Introduction

1.1 Background and Significance of the Problem

1.1.1 Overview of energy situation

Energy is necessary. Any activity that occurs on this planet requires energy. With less energy in the world, the effect on humanity is a lowering of the quality of life. This is inevitable. Scarcities of fossil fuels causes oil prices to rise and coupled with exploding population have resulted in serious crisis. The mean crude oil reserve to production ratio is 53.3 years for the whole world as shown in Figure 1.1 (BP Statistical Review of World Energy, 2014). South and Central America have the longest oil reserves, more than 100 years, in contrast to Asia Pacific. The depletion of oil is due to population growth and industrial development. By extrapolating the world population from the year 2010 to 2080, it is expected to rise from 6 billion to over 10.8 billion (Sheffield, 1998). There will be an increase in fuel consumption because of industrial development over this period. From evaluation of oil consumption between of 1988 and 2013, fuel consumption was expected to increase more than 38% worldwide, shown in Figure 1.2. South and Central America, Africa and Asia Pacific regions are likely to increase oil consumption, while North America and Europe decrease. The Middle East usage of oil is uncertain (BP Statistical Review of World Energy, 2014). With research and development of renewable energy sources, the solution to replace fossil fuel needs serious studies.

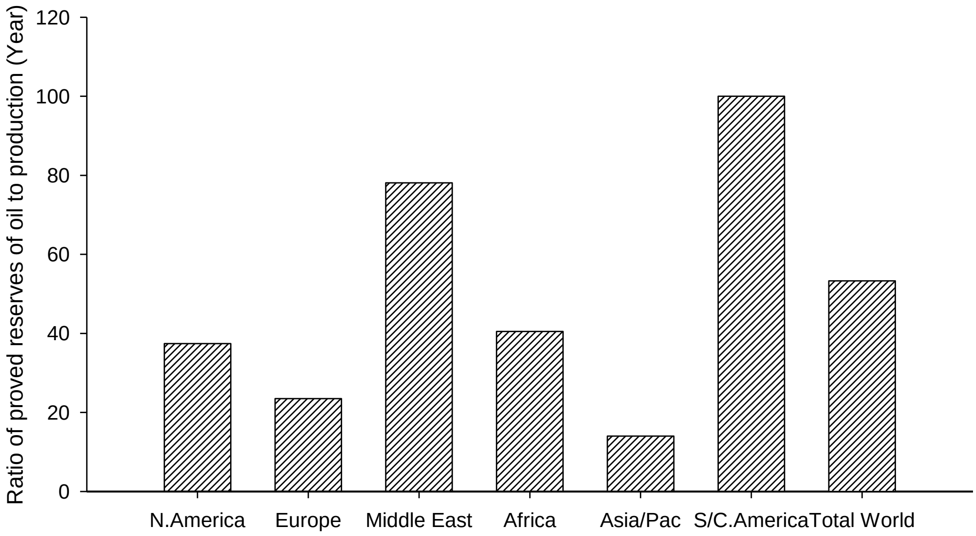

Figure 1.1 Ratio of proved reserve of oil to production in the world

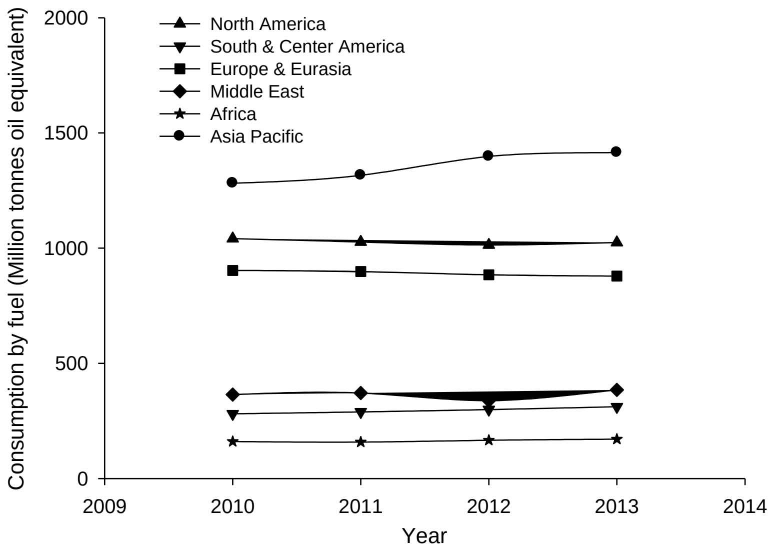

Figure 1.2 Oil consumption/region of the world

Thailand has limited domestic oil production and reserves. Our oil consumption relies on imported oil from overseas. With the world oil crisis, Thailand will be affected inevitably. Foreign oil imports are calculated at 85% of total oil consumed in Thailand. Most oil imports are from the Middle East, more than 60% and valued at 39,855 million US$. From analysis of energy consumption by economic sectors in Thailand, the transportation and industrial sectors are the largest energy consumers with more than 35%: commercial, agriculture and residential use 7.2, 5.2 and 15.1%, respectively. Because of high energy consumption in the country, the Thai government is promoting and motivating renewable energy, such as hydro power, wind energy, solar, biomass and biofuel (DEDE, 2012). With ten years of renewable energy development, the government aims to increase renewable energy by 25% of total energy consumption shown in Figure 1.3. Measures and motivation by the government to use renewable energy include research and development, adder cost, working capital and energy service company fund (ESCO). The first measure supports research and development of high potential renewable energies such as solar, hydro, wind, biomass agricultural residues and solid waste. Adder cost measure is incremental to the purchase price of renewable energy generation at higher than normal prices and power producers would receive for selling electricity to the power utilities (Tongsopit et al., 2013). Currently, the project has seen great interest from investors. Working capital measure reduce the initial investment in new renewable energy technologies; low-interest loans to include measures to promote privilege investments in domestic (BOI); for new renewable energy investment. Energy service company fund measure is used to promote investment and risk assessment.

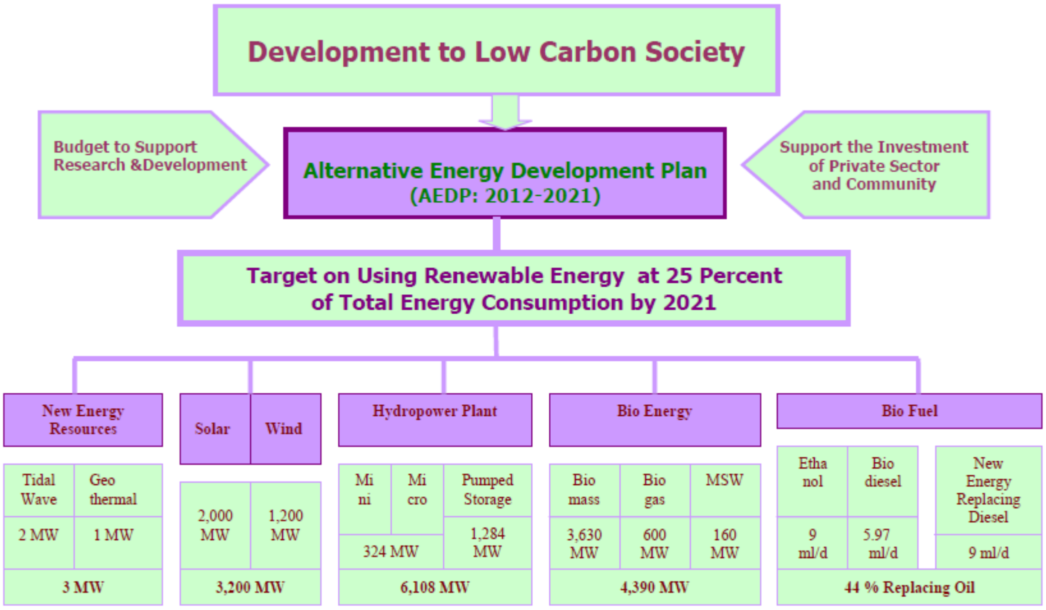

Figure 1.3 Strategic of renewable energy development in Thailand (DEDE, 2012)

1.1.2 Agricultural residues potential in Thailand

Thailand is an advancing agro-industrial country and a major agricultural products exporter. Processing of agriculture's products uses labor-saving devices such as small agricultural engines. Power and speed of the engines used are mostly in the range of 2.2-10.4 kW and 1000-2000 rpm, respectively. Most work includes pumping, power generation and plowing on farms, with high diesel and gasoline consumption. In 2005, fuel oil consumption in rice production in Thailand was more than 1,200 million liter, or 125 L/ha (Liter/hectare) for 9.92 million hectare of rice production. Expenses were calculated to be more than 36 billion Bath/year and considered to be high cost for a developing country (Chaichana, 2004). However, Thailand has energy potential from agricultural residues. The biomass waste comes from production of: sugarcane, paddy, oil palm, coconut, cassava, maize, groundnut, cotton, soybean, sorghum and wood residues such as longan wood, mango wood, rubber wood, etc. Thailand has energy potential from biomass of about 44 million tons or equivalent to 612.89 Peta Joule (PJ) of energy (Sajjakulnukit et al., 2005). However, some biomass is burnt as it is uneconomical to collect, producing a pollution problem throughout the country, especially in the north of Thailand. Therefore, if Thailand can convert agricultural residues into to alternative energy, it can reduce fuel oil imports and pollution.

1.1.3 Biomass conversion technology

Biomass converted to producer gas via gasification is of great interest because the fuel gas can be used directly in engines, producing less emission and has applications for heat usage. The advantage of gasification is that using producer gas is potentially more efficient than direct combustion/burning of the original fuel. Gasification is an irreversible thermo-chemical process, by which feedstock is thermally decomposed. The reactions are conducted at high temperatures, between 500-1400°C at atmospheric pressure. The reactions use air, pure oxygen, or steam mixing with another gases. The end products are in gaseous form. The resultant producer gas is composed of hydrogen (H_2), carbon monoxide (CO), methane (CH_4), carbon dioxide (CO_2) and nitrogen (N_2) with a mean calorific value of about 4.0-6.0 MJ/Nm^3. The use of pure oxygen and steam if used in gasification process produces higher concentration of CO and H_2 and less N_2 content, and leading to high calorific value of gas which may be more than 15 MJ/Nm^3. At the same time, care is needed to enable flame stability during combustion (Hernandez et al., 2012).

1.1.4 Utilization of producer gas in IC engines

Producer gas has been utilized in two different types of existing four stroke engines. Firstly, spark ignition engines operated on gas as gas engines and can be classified into two types; low and high compression engines. The first type with low compression ratio (CR) has very low thermal efficiency and the power output is de-rated by more than 30% during the operation (FAO, 1986). Reduction of power and thermal efficiency of the engine is due to lower energy density of the producer gas, compared to that of gasoline and diesel fuels. The engine in high CR is converted from diesel engines and operated as spark ignition gas engines powered by gas. The efficiency and power of high CR engine has performance close to conventional engine. Secondly, compression ignition engines are diesel engines operated on gas and diesel as dual-fuel engines. Second type engine uses diesel for pilot ignition. The power output and thermal efficiency are lower than diesel engines. This engine can decrease diesel consumption by more than 70% with appropriate tuning and advanced injection timing (Tongorn, 2007). Besides, the emissions are lower, compared to diesel engines. However, the use of diesel for pilot ignition will be higher than spark ignition engine with 100% producer gas. Comparison of both engine types, the high CR spark ignition (SI) engine is attractive and has high possibility. However, the engine needs to be modified. Prior studies of this engine were shown to be successful. Effect of engine performance parameters are unclear, such as combustion chamber configuration, engine speed, ignition timing, and load. Most researchers believed that knocking on engine may increase with increasing CR. There are not many higher CR, producer gas engine. Thus, it is of great interest to study producer gas engines at high CR.

Therefore, in this research, effects of CR, geometry of engine combustion chamber, engine speed, ignition timing, and load on performance of a producer gas engine namely; engine torque, power, specific fuel consumption, specific energy consumption, thermal efficiency, emissions, and economics are investigated and compared with those from a typical diesel engine.

1.2 Literature Reviews

1.2.1 Producer gas for SI engines with low CR

Parke et al. (1981) reported work on gasoline engine operation on producer gas obtained from corn stove gasification in a fluidized bed. Natural gas and gasoline fuelled engine performance data were included for comparison. Results indicated that maximum engine power occurred with a lean mixture and producer gas fuelling with de-rating of 34%, compared to gasoline operation. The authors suggested supercharging to enhance the engine with a lesser de-rating in a supercharged mode Parke et al. (1995).

Martin et al. (1981) reported work using charcoal gas and producer gas on an SI engine at a CR of 7:1 with a de-rating of 50% and 40%, respectively. However, the same authors claimed 20% de-rating when worked with producer gas at a CR of 11:1. The authors presented a CR barrier of 14:1 and 11:1 for charcoal and producer gas, respectively.

Jenkins et al. (1988) reported work using producer gas on small SI engine at 6.3 of CR, L-head engine, 206 cc displacement, wedge type combustion chamber and the nominally rate power of 3.7 kW at 3600 rpm. The experiments used producer gas and gasoline fuel. It was found that the maximum brake power in optimal spark timing was decreased by more than 55 % against gasoline operation. However, the thermal efficiency of both fuels were likely similar. Experiment analysis, the brake power was increased with CR of 7.3:1, Increasing CR was related with geometry of combustion chamber.

Munoz et al. (1999) reported work using forestry residues derived producer gas on an SI engine, as shown in Figure 1.4. The engine used in this work was not specially modified for producer gas. The engine was a Honda GX

270, 6.6 kW, single cylinder, naturally aspirated suitable for use with gasoline and liquid petroleum gas (LPG) or natural gas at a CR of 8.2:1. It was found that the engine torque and power with producer gas were approximately half those gasoline. On the other hand, it must be emphasized that as a result of the unsuitability of the gas dosage equipment mentioned above, it has not been possible to obtain the maximum power. The power was reduced because of the lack of ability to supply the power of gas required.

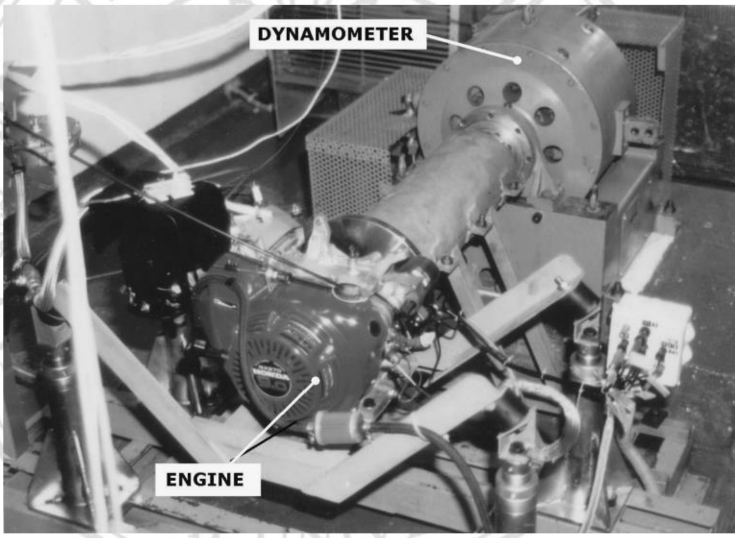

Figure 1.4 Small SI engine operate on forestry residues producer gas (Munoz et al., 1999)

Ando et al. (2005) reported using syngas on an SI engine compared to the natural gas driven engine, as shown in Figure 1.5. Two simulated low-BTU gases, obtained from one-step high temperature gasification (hydrogen rich) and two-step pyrolysis/reforming gasification (methane rich), were tested in a small-scale spark ignition engine was changed from 8.47 to 9.4 and 11.9 of CR. The base engine was a 296 cc displacement, single-cylinder gasoline engine. The hydrogen rich low-BTU gas driven engine showed similar thermal efficiency, about 23%, although output was lower and wider equivalence ratio range for stable engine operation. On the other hand, the methane rich low-BTU gas engine showed narrower equivalence ratio range for stable operation. The test results showed engine performance to be more affected by combustion characteristics than the heating value of the fuel gas. For better engine performance, hydrogen rich fuel gas may be desirable.

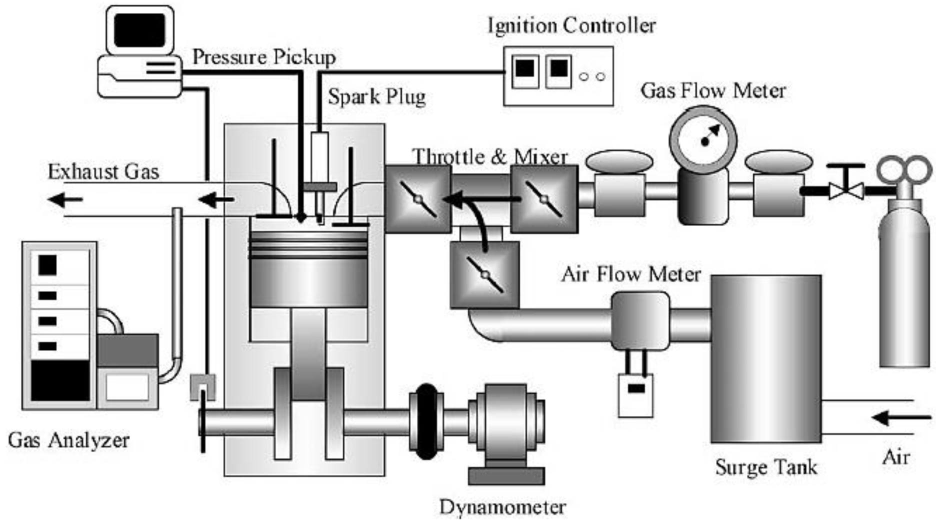

Figure 1.5 Experimental setup of SI engine operate on low-BTU gas (Ando et al., 2005)

Ahrenfeldt (2007) reported work using wood chip producer gas on an SI engine, shown in Figure 1.6. The engine was converted from diesel engine, in line 3 cylinder, 3.1 L, 20 kW of rate power, bowl-in-piston, naturally aspirated and thermal efficiency, performance, emission involve another parameters were evaluated. The experiment were operated more than 3200 h. The producer gas was generated from two stage downdraft gasifier and with 6 MJ/Nm^3 of gas heating value. The engine was run on producer gas smoothly. The overall electric efficiency was 25%, based on low heating value. Adjustments of spark timing made no impact on power output and efficiency, but level of NO_x and CO.

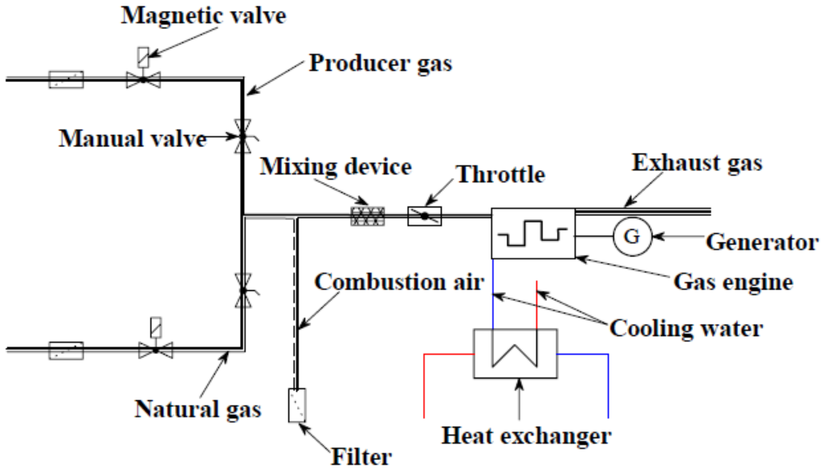

Figure 1.6 Schematic diagram of experimental on 20 kW producer gas engine (Ahrenfeldt, 2007)

Shah et al. (2010) reported work using hardwood chips and producer gas on an SI engine at a CR of 9:7, shown in Figure 1.7. The engine was 5.5 kW, naturally-aspirated, single cylinder, four stroke, spark-ignited engine driven alternating current generator, designed to operate electrical lighting, appliances, tools and motor loads. The overall efficiencies for gasoline and syngas operation were similar at their respective maximum electrical power outputs. The maximum electrical power output for syngas operation (1392 W) was lower, compared to gasoline (2451 W). The maximum power output in syngas operation was only 1.8 times lower than for gasoline operation, while the density of syngas (1.7 kg m^{-3}) was 423 times lower than density of gasoline (720 kg m^{-3}).

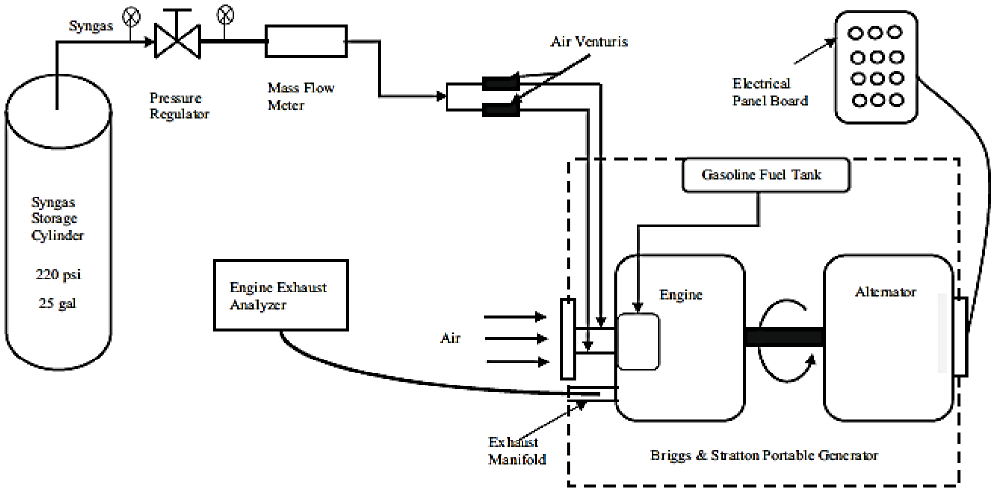

Figure 1.7 Experimental setup of 5.5 kW SI engine on producer gas fueled (Shah et al., 2010)

Dasappa et al. (2011) studied use of producer gas on 100 kW SI engine coupled to a generator at a CR of 8.5, shown in Figure 1.8. The engine was Cummins GTA-855-G, six cylinders, 4-stroke, turbo-charged with after cooler gas engine and was not specially modified for use on producer gas. The gasification system can be operated at the rated condition of 135 kg/h continuously. Measured producer gas compositions show CO and H_2 in the range of 18±1%, CH_4 1.8±0.4%, CO_2 9±1%. The composition would result in a gas calorific value of about 4.5±0.3 MJ/kg. They found that specific fuel consumption at 1.36 kg/kWh when the measured calorific value of biomass used at 15 MJ/kg. The overall efficiency of conversion of biomass to electricity was about 18%. The engine efficiency was 25%, lower by about 6 points compared with natural gas operation. The main reasons for the reduction in efficiency were related to the higher exhaust temperatures, properties of producer gas and also combustion processes with producer gas inside the engine cylinder.

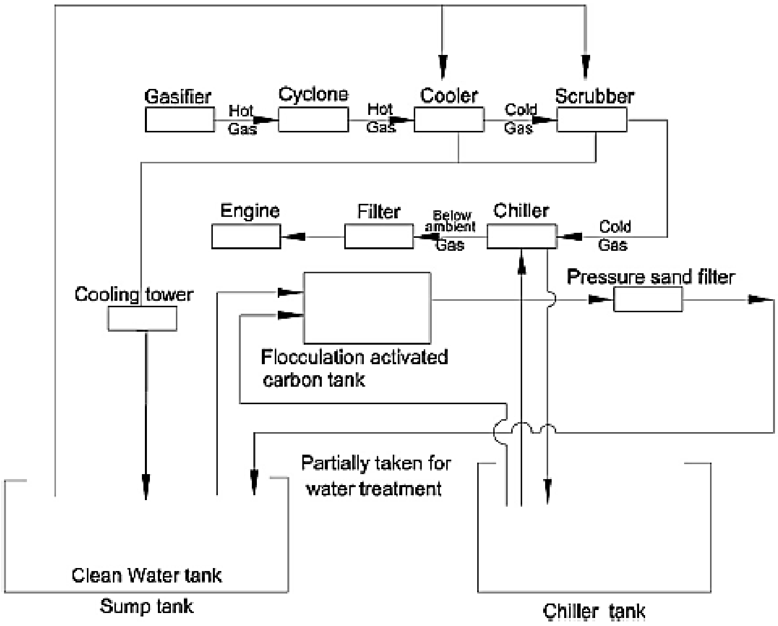

Figure 1.8 Schematic diagram of 100 kW biomass gasification plant (Dasappa et al., 2011)

Lee et al. (2003) reported work using syngas combined with landfill gas (LFG) on an SI engine. The main component of landfill gas is methane (CH_4) and component of syngas included H_2, CO, and CO_2. The engine used in the study was Honda GC 160E-QHA engine, a single-cylinder, four-stroke, a CR of 8.5.1, 3.5 kW. They found that 10% of syngas addition also increased the engine efficiency (Kohn et al., 2011). The maximum efficiencies were 0.73 % and 1.1 % higher than the addition of H_2 and CO, respectively.

Mustafi et al. (2006) studied the use of syngas obtained from the processing of ‘‘Aqua-fuel” in a Ricardo E6 gasoline SI engine. The engine used in our study was a single-cylinder, four-stroke, naturally-aspirated. Syngas used in our study was produced by biomass gasification. They found that syngas produced about 20 and 30% lower engine power output than natural gas and gasoline, respectively at a CR of 8.1. At higher CR of 11.1, the brake torque for syngas improved, compared to the other fuels. When operating at around 1500 rpm, syngas produced about 18% less torque output than natural gas. This was a considerable improvement over the 30% less torque at the lower CR. However, the brake specific fuel consumption remained almost the same at this operating condition at CR of 8:1.

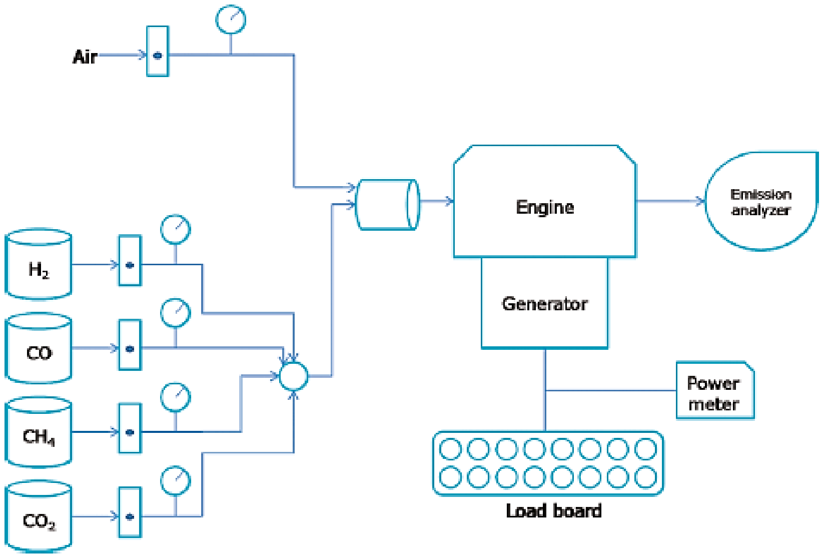

Figure 1.9 Experimental setup of engine operate on syngas mixed landfill gas (Lee et al., 2003)

1.2.2 Producer gas for dual fuel engine

Uma et al. (2004) reported work using producer gas in a diesel engine on dual fuel mode, as shown in Figure 1.10. The engine was "Kirloskar" direct injected, six cylinders, vertical and four stroke engines with mechanical injector, 77:2 kW, 6.614 L and CR 15:1. Experimental investigations in diesel alone and dual fuel mode were carried out at different loads of 10; 20; 30 and 40 kW. Three experiments were conducted at each load. In each experiment, emission parameters such as CO, NO_x, SO_2, HC and particulates and parameters related to thermal performance of engine such as fuel consumption and diesel replacement were measured. They achieved the maximum diesel replacement in a range of 67-86% at 30 kW of load and low sulfur dioxide, hydrocarbons and oxides of nitrogen, compared, to the diesel mode.

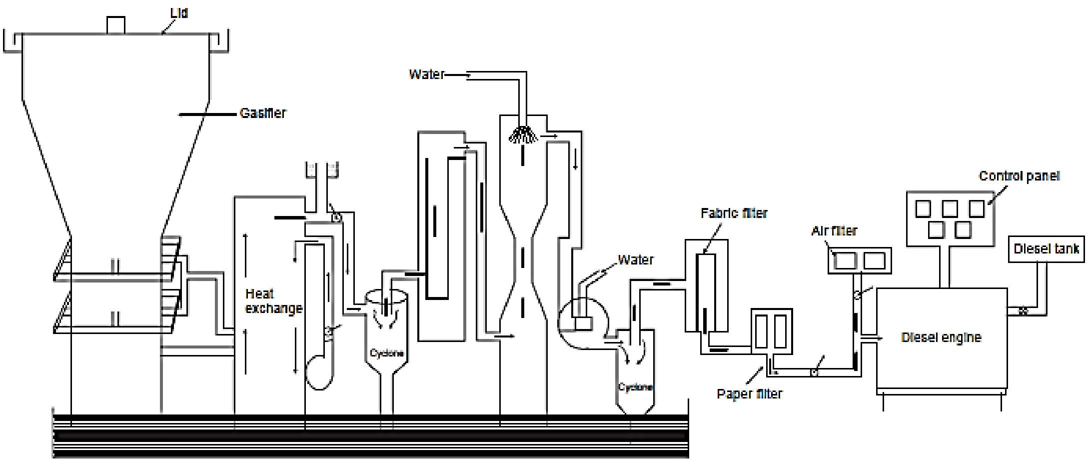

Figure 1.10 Experimental setup of diesel engine operate on producer gas (Uma et al., 2004)

Singh et al. (2006) reported work using producer gas in a diesel engine on dual fuel mode, shown in Figure 1.11. The engine was “Kirloskar” three cylinders, naturally aspirated, four strokes, direct CR 17:1 and 23 kW at 1500 rpm of constant engine speed. Performance of dual engine was evaluated in terms of diesel replacement, brake thermal efficiency and exhaust gas composition. Experiments were initially carried out on the engine at five loads 21, 42, 63, 84 and 98% of rated capacity. The maximum percentage diesel replacement was of 63, 68 and 22 at 63, 84 and 98 % of load, respectively. The brake powers were decreased between 0.3-0.9%. The emission was lower compared to diesel.

Ramadhas et al. (2007) presented results from a producer gas engine of 5.5 kW single-cylinder, four stroke, direct injection, naturally aspirated, as shown in Figure 1.12. Performance of dual engine was evaluated in terms of brake thermal efficiency, specific energy consumption, pilot fuel savings and exhaust gas composition. They found that the specific energy consumption reported for both wood chips and coir pith as fuel was 18 MJ/kWh, when compared the diesel specific energy consumption of about 15 MJ/kWh. They reported a maximum of 72 % pilot fuel savings at 50 % of the rated load. Exhaust emission was found to be higher in the case of dual fuel mode of operation, compared to neat diesel operation.

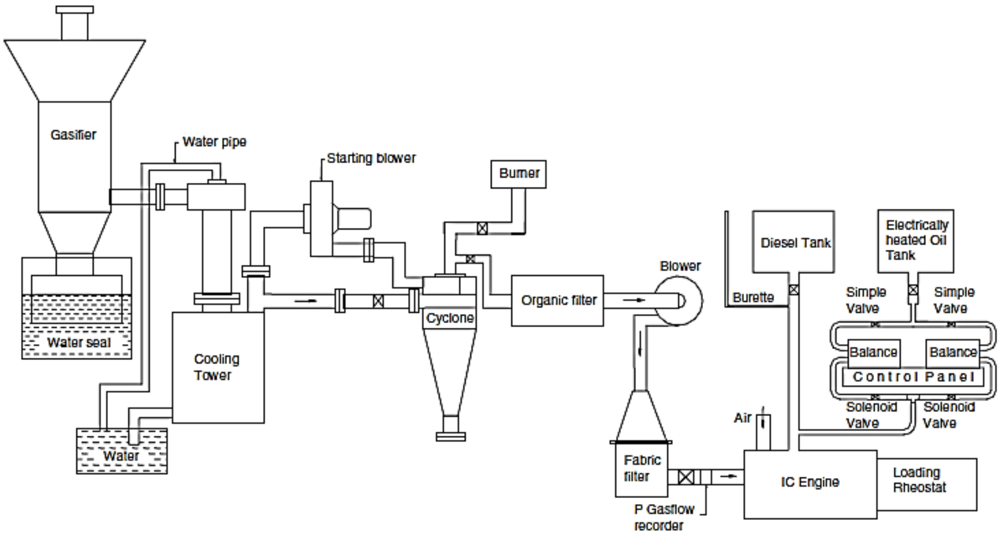

Figure 1.11 Schematic diagram of gasifies system by Singh et al. (2006)

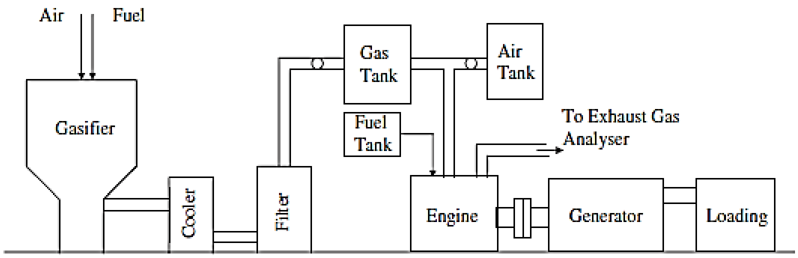

Figure 1.12 Experimental setup of 5.5 kW diesel producer gas engine (Ramadhas et al., 2007)

Banapurmath et al. (2008) reported work using producer gas in a diesel engine on dual fuel mode. The fuel testing of the engine was biodiesel alone and biodiesel-producer gas. The engine tests were conducted on 5.2 kW a four-stroke, single cylinder direct injection and CR of 17.5:1, as shown in Figure 1.13. The engine was always operated at its rated speed of 1500 rpm. Performance of dual engine was evaluated in terms of brake thermal efficiency, Honge oil substitution, exhaust gas analysis and effect of injection timing with brake power. The brake thermal efficiency with dual fuel operation at injection timings of 19.1, 23.1 and 27.1 BTDC were 17.25%, 18.25% and 19.00%, respectively, as compared to 28.5%, 27% and 26% for neat Honge oil operation. Substitution was higher at lower loads and found to decrease with load. The trend was similar for all injection timings. Maximum Honge oil substitution up to 58% was observed at an injection timing of 27.1 BTDC. The CO emission on dual fuel mode was higher than biodiesel mode, but the NO_x and smoke opacity were lower at various brake powers.

Figure 1.13 Overall view of 5.2 kW diesel engine on producer gas fueled (Banapurmath et al., 2008)

Dasappa et al. (2011) reported test results on dual fuel with producer gas and diesel on a 68.4 kW, Isuzu engine of 6.494 l, direct injection, six cylinders, naturally aspirated and CR of 17.5:1, as shown in Figure 1.14.

The engine was coupled to a 75 kVA alternator to generate 190–240V electricity at 60 Hz. Performance testing of the engine in diesel and duel fuel modes was carried out to assess fuel consumption, diesel replacement and exhaust gas emissions. Measurements were made on the diesel consumption at various loads between about 8 - 60 kWe. They found that the maximum diesel replacement was about 88.5% at 29.2 kWe of load. Diesel consumption was in a range between 1.21-3.89 l/h. Overall efficiency of about 22 % was obtained. The CO emission on dual fuel mode was higher than diesel mode, but the NO_x emission was lower at various loads.

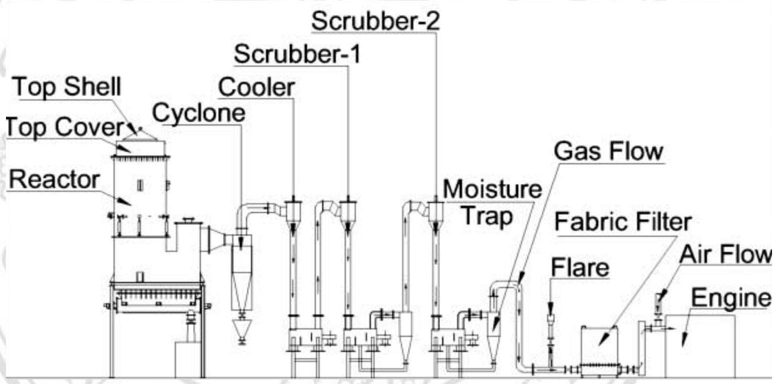

Figure 1.14 Schematic diagram of biomass gasification plant on dual fuel mode (Dasappa et al., 2011)

Hassan et al. (2011) reported work using producer gas in a diesel engine on supercharged dual fuel mode, shown in Figure 1.15. The experiments were carried out on a 4.9 kW, a four-stroke, single cylinder, direct injection and CR of 19.1. Performance of dual engine was evaluated in terms of brake thermal efficiency, specific energy consumption, diesel displacement and exhaust emissions with brake mean effective pressures. They found that the maximum brake thermal efficiency of diesel dual fuel was about 21.38% and 24.11% for the pure diesel fuel operations. The values of specific energy consumption at 80% load in diesel fuel and producer gas-diesel operation were 15.17 and 17.67 MJ/kWh, respectively. The maximum diesel displacement about was 70.42% at 50% of load. The CO emission on dual fuel mode was higher than diesel mode, but the NO_x emission was lower at various loads.

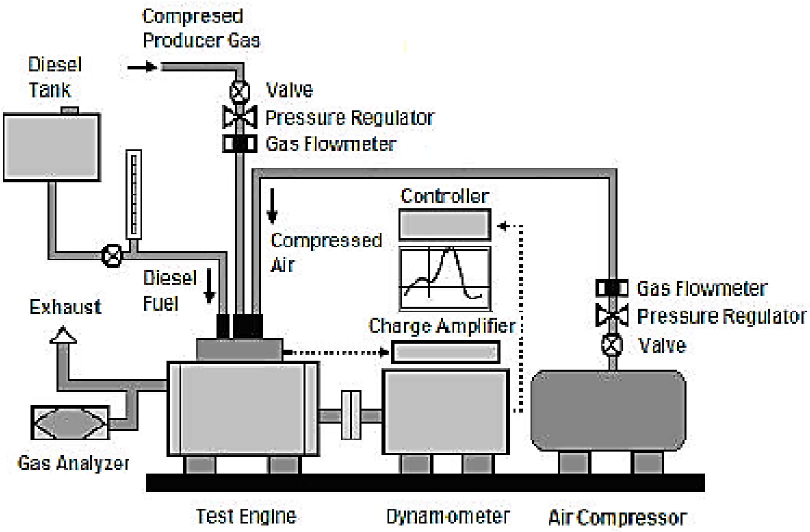

Figure 1.15 Overall view of experimental setup by Hassan et al. (2011)

Dussadee et al. (2011) reported test results on dual fuel with producer gas and diesel on 46.4 kW, as shown in Figure 1.16. The experiments were carried out on “Perkins 1103A” four cylinders, direct injection and CR of 17.25:1. Performance of dual engine was evaluated in terms of brake thermal efficiency, specific energy consumption, diesel displacement and exhaust emissions with various loads between about 9 - 20 kWe at 1500 rpm of constant engine speed. The producer gase was from wood gasifier. The maximum brake thermal efficiency of diesel dual fuel was about 20.88% and 29.62% for the pure diesel fuel operations at 20 kWe of load. The diesel displacement about was 60.72% on 20% of load. The specific energy consumption of diesel dual fuel was about 20.88% and 12.15% for the pure diesel fuel operations. The CO emission on dual fuel mode was higher than diesel mode but the NO_x and smoke opacity emission was lower at various loads.

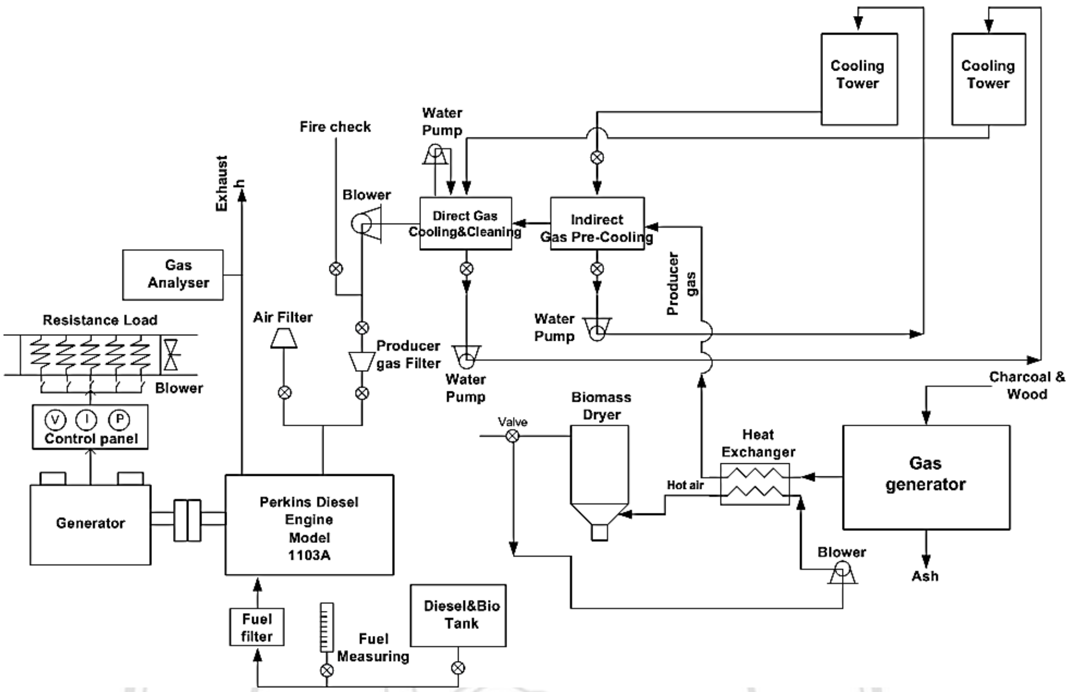

Figure 1.16 Overall view of experimental setup by Dussadee et al. (2011)

Lekpradit et al. (2009) carried out a study to advancing the injection timing of a diesel engine on dual fuel mode. The experiments were carried out on a 50 kW Cummins diesel engine, four cylinders, 3.9 L, direct injection, and CR of 16.5:1, as shown in Figure 1.17. The diesel and producer gas engine was adjusted at standard timing injection 12° of BTDC and advanced timing injection adjusted at 17° of BTDC. Performance of the dual engine was evaluated in terms of brake thermal efficiency, specific energy consumption, diesel displacement and exhaust emissions with brake mean effective pressures. They have found that increasing advanced injection timing affected lower specific diesel consumption. Conversely, the overall efficiency and diesel replacement were increased. The maximum brake thermal efficiency of 53.24% was achieved in dual fuel engine in advanced timing injection mode. The diesel replacement rate went up to 76.5% at 40% of full load. Finally, The CO emission on dual fuel mode was higher than diesel mode, but the NO_x and emission was lower at various loads.

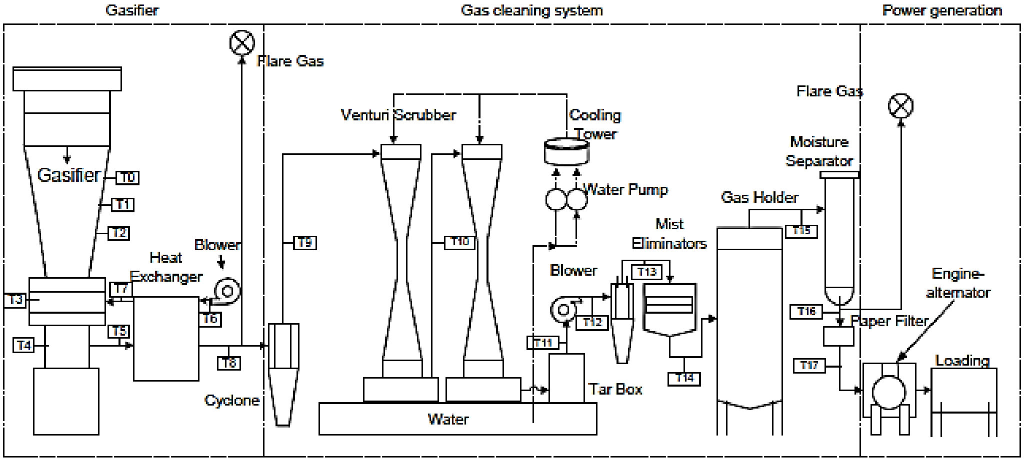

Figure 1.17 Diesel producer gas engine and biomass gasifier system (Lekpradit et al., 2009)

Das et al. (2011) reported work using producer gas in a diesel engine on dual fuel mode, shown in Figure 1.18. Experiments were conducted to study the performance of a diesel engine, four stroke, single cylinder, 5.25 kW with respect to its thermal efficiency, specific fuel consumption and diesel substitution by use of diesel alone and dual fuel mode at 1600 rpm of engine speed with variable engine loads. The gas producer system developed for a 5.25 kW diesel engine was found to perform satisfactorily by using three types of biomass such as wood chips, pigeon pea stalks and corn cobs. The average value of thermal efficiency on dual fuel mode was found to be slightly lower than that of diesel mode. The specific diesel consumption was found to be 60 to 64 % less in dual fuel mode, than that in diesel mode for the same amount of energy output. Average diesel substitution of 64% was observed with pigeon pea stalks, followed by corn cobs (63%), and wood chips (62%).

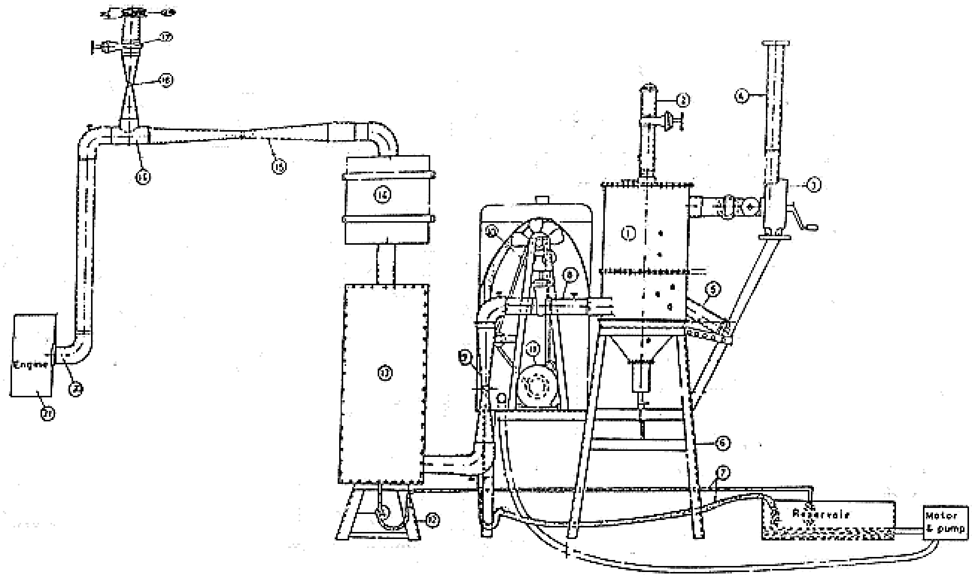

Figure 1.18 Overall view of experimental setup by Das et al. (2011)

Sahoo et al. (2011) reported experimental work based on the performance evaluation of a base diesel engine using syngas under dual fuel mode without varying any of the existing engine design and operating parameters. Schematic of experimental setup is shown in Figure 1.19. In this study, effects of simulated syngas with different compositions of H_2/CO ratios on engine performance brake thermal efficiency, diesel substitution and emissions were examined. The volumetric proportions of H_2 and CO in syngas considered for dual fuel study were 50:50, 75:25 and 100:0. The test was accessed from the operation of a four-stroke, 1500 rpm of constant speed, direct injection and CR of 17.5:1, diesel engine at varying engine loads of 20, 40, 60, 80 and 100%. The maximum diesel replacement was found to be 72.3% for 100% H_2 syngas mode. At the same engine load, the thermal efficiency was found to be 16.1% for 50% H_2 syngas. It increased to 18.3% and 19.8% when H_2 content was increased to 75% and 100%, respectively. CO and HC emission levels were recorded for 25 and 50% CO fraction syngas fuels due to their CO content in the fuel compositions. At part-loads (20% and 40% loads), all the tested ranges of syngas modes resulted in a poor performance including higher emission levels.

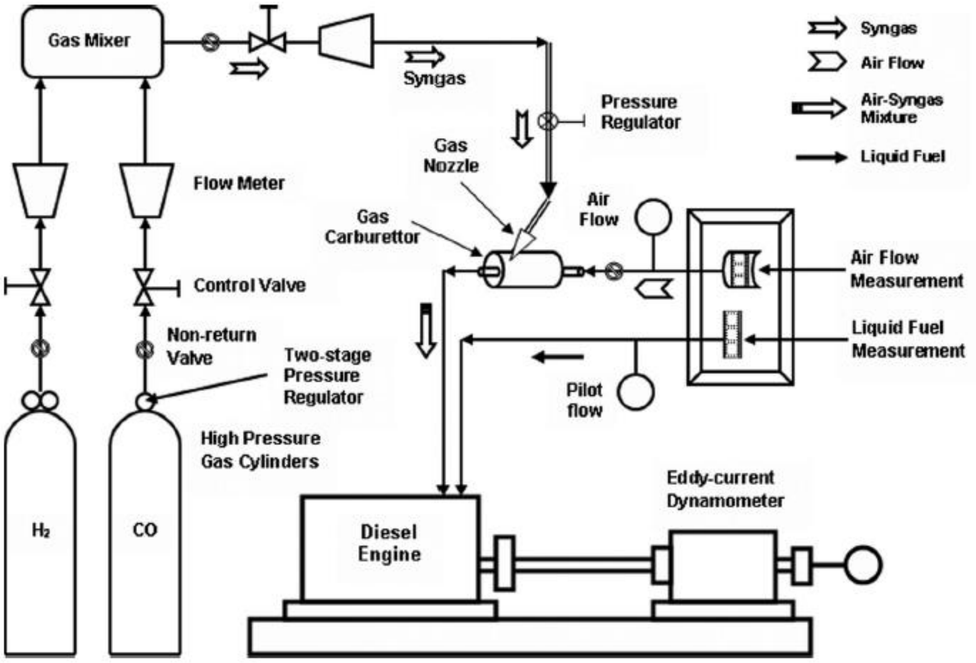

Figure 1.19 Experimental setup of diesel fuel mode by Sahoo et al. (2011)

1.2.3 Producer gas for SI engines with high CR

Ramachandra. (1993) reported working on producer gas fuelled engine at high CR (16.5:1) for water pumping application without any sign of knock but engine power was reduced by 20%, compared to diesel oil. Thermal efficiency of producer gas engine was 19%.

Shashikantha et al. (1994) and Parikh et al., (1995) reported work on a producer gas engine converted from a naturally aspirated diesel engine at CR of 11.5. The reason for limiting the CR was cited to be the knocking tendency; however, no experimental evidence was provided in support it. The work reported a gas engine converted from a diesel engine with a modified combustion chamber. The modified combustion chamber of Hesselman (shallow W) shape was claimed to enhance the in-cylinder turbulence by suppression of swirl and promotion of squish effect. With the above modification, a power output of 16 kW was reported in gas mode against a rated output of 17 kW in diesel mode. The maximum thermal efficiency was claimed at 32%, which was close to the results in compression ignition (with diesel) mode at an output of 15 kW. It is quite surprising to note the conversion efficiencies to be same, when the CRs were widely different. The authors also claimed that an optimum ignition timing was 35° BTDC, compared to 22° BTDC for natural gas on the same engine. With the producer gas claiming to contain about 24.1% H_2, 21.5% CO and 2.1% CH_4 the burning velocities ought to be higher than natural gas. In contradiction to the reported ignition timing, the claimed gas composition would require the ignition timing to be located close to TDC. Therefore, it was not clear as to how such a large output was obtained at advanced ignition timing when all logics point towards retarded ignition timing (as also demonstrated in this work).

Nay Zar Aun. (2008) reported work on a producer gas engine converted from a naturally aspirated, diesel engine. The experiments were carried out on "Yandown /YSD 2100" twin cylinder, 26.5 kW with diesel fuel. The work was undertaken on a producer gas engine converted from a diesel engine with a converted CR from 18:1 to 10.1:1 and modified the piston crown, as shown in Figure 1.20. In this study, performance of producer gas engine was evaluated in terms of power and torque output with engine speed compared diesel mode. It was found that the power and torque output of engine reduced by 40%, compared to diesel mode and 32-35° BTDC of optimum ignition timing. At full throttle, no load condition, the engine can run nearly 2000 rpm and the engine were coupled to generator. At engine speed of 1800 rpm, the engine can be used for 300 of 40 W lamps in a village.

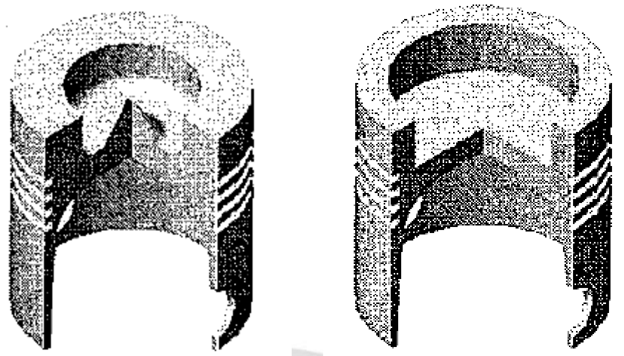

Figure 1.20 Piston crown modification for SI producer gas engine
(Nay Zar Aun, 2008)

Shidhar et al. (2005) reported experimental work on producer gas fuelled spark ignition engine converted from diesel engine, and two types of natural gas engine. The work was reported on a gas engine converted from a diesel engine "Kirloskar RB-33", 3 Cylinder, bowl in piston chamber, shown in Figure 1.21. In this study, the engine was tested and verified at the highest CR of 17:1 in order to establish knock-less performance by capturing the pressure-crank angle trace. Subsequently, the engine was tested at varying CRs so as to arrive at an optimum CR for maximum brake power and efficiency. The two natural gas engines on producer gas fuel Greaves, TBD, V12, 240 – 258kW, diluted natural gas at a CR of 12:1, and Cummins, G743G, inline, 6 Cylinder, 84 kW, with natural gas at a CR of 10:1, respectively. In this study, the engine was tested at CR for maximum brake power and thermal efficiency. It was found that for the producer gas engine converted from a diesel engine, the maximum brake power and thermal efficiency were 17.5 kW and 30.7%, respectively at CR of 17:1, 6° BTDC of optimum ignition timing and CR of 11.5:1. The thermal efficiency of engine was reduced due to heat loss. The maximum brake power and thermal efficiency of Greaves and Cummins engine were 165 kW, 55 kW and 28.3%, 27.4% respectively. The optimum ignition timing of Greaves and

Cummins engines were in a range of 12-14° BTDC and 22-24° BTDC. Maximum net engine outputs of different engine are shown in Table. 1.1.

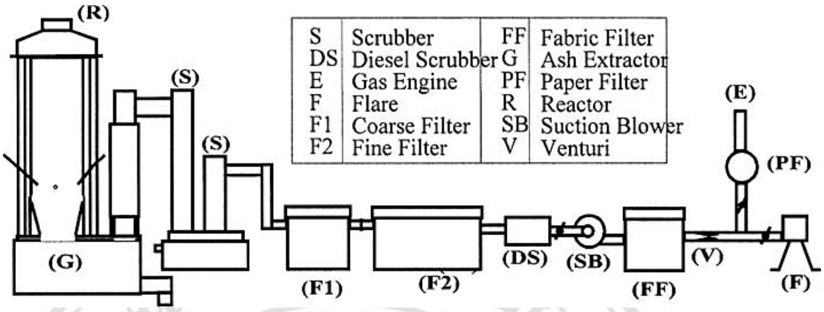

Figure 1.21 Schematic diagram of experimental setup by Shidhar et al. (2005)

Table 1.1 Maximum power output of different engine (Shidhar et al., 2005)

Engine	CR	Spark timing (Degree)	A/F Ratio	Net Elec. Power (kW)	Net Brake. Power (kw)	BTE (%)
E1	17	6	1.1	17.5	20	30.7
	14.5	10	1.1	16.4	18.8	29
	13.5	14	1.06	16.2	18.6	29.3
	11.5	15,17	1.07	15.3	17.6	27.5
E2	12	12,14	0.94	165	182	28.3
E3	10	22,24	1.01	55	60	27.4

Raman et al. (2013) reported test results of producer gas on an SI engine, compared to natural gas and diesel operation at a CR of 12:1. The engine was 100 kW in natural gas, 6 cylinders, four strokes and 85 kW_e of rated power output. The gas generator was a downdraft gasifier, shown in Figure 1.22. The engine speed was fixed at 1500 rpm and load was varied between 5-100%. The maximum overall efficiency was 21% at 85% of full load, while, maximum power output was reduced by 21.4%. The biomass consumption was 1.2 kg/kWh. The volumetric efficiency, energy density, CR and adiabatic flame temperature affected power efficiency.

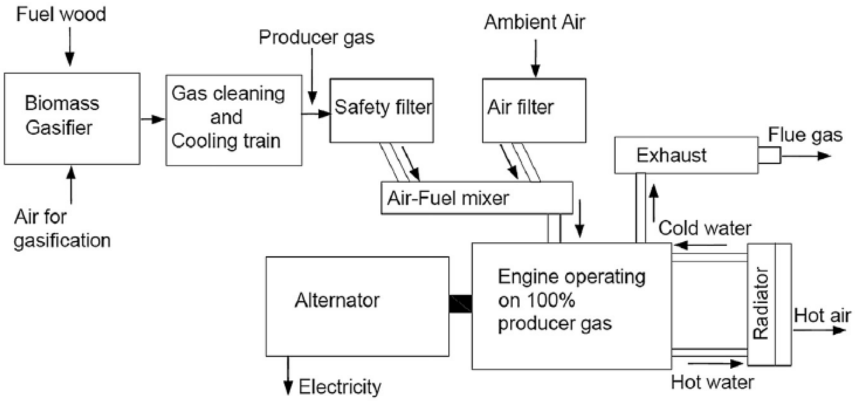

Figure 1.22 Schematic diagram of producer gas engine by Raman et al. (2013)

From previous research works, it was clear that. The use of producer gas on internal combustion engine was three of types, consisting of SI engines at low CR, diesel engine in dual mode, and SI engines at high CR or diesel converted to SI engine. For the producer gas engine at low CR, this tested engine was in a range of 3.5-240 kW, single to twelve of cylinder and 7-12:1 of CR. The power output was reduced 30-50%, compared to gasoline or natural gas. For diesel engine in dual mode or diesel/producer gas engine, this tested engine was in a range of 5.2-68.4 kW, single to six of cylinder. CR was original diesel engine. The power output of engine was lower than diesel mode by 0.3-0.9%. The thermal efficiency was in a range of 17-22%. For SI engines at high CR, most engines modified from diesel engine were in a range of 16-26.5 kW, two to three of cylinder and 10.1-17:1 of CR. The power output of engine was reduced by 12-20%, compared with diesel. The thermal efficiency was 19-30%. Comparison advantages, disadvantages and feasibility of three type engines for use in the work. The SI engines at low CR was low possibility producer gas as fueled, due to lower power output and thermal efficiency. The use of dual fuel in diesel engine was possibility, but high fuel cost. The use of SI engines at high CR was highest possible, because, the power output and thermal efficiency was high than engine at low CR and lower fuel cost. Previously, there are no studies in small agricultural engines and parameters that affect engine performance such as CR, type of combustion chamber, spark timing, engine speed and load Therefore, the SI engines at high CR are interesting in terms of engine performance and fuel cost.

1.3 Objectives of the Study

1.3.1 To develop producer gas engines adapted from existing small agricultural engines.

1.3.2 To carry out parametric study of CR, spark timing, and type of combustion chamber that affect performance of small producer gas engines.

1.3.3 To deploy mathematical model of an IC engine in predicting producer gas engine performance.

1.4 Expected Benefits

1.4.1 Agricultural engines fueled with producer gas are demonstrated. Knowledge obtained can be applied to other engines that can be used in farm and generator.

1.4.2 Explanation on effect of CR, combustion chamber, ignition timing, load and engine speed on small producer gas engines performance.

1.4.3 Optimization of engine parametric to application in modified and designed producer gas engine in the future

1.5 Scope of Research

1.5.1 Engine size studied was 10 kW or smaller.

1.5.2 Engine speed tested was in the range of 1,000-2,000 rpm.

1.5.3 Fuel used for gasification was charcoal or wood.

1.5.4 CR was between 10:1-21:1.

1.5.5 Combustion chambers used were chosen from bowl in piston and bath tub.

1.5.6 Gasifier used was downdraft type, batch feeding, and 5-15 kg/h.

1.6 Thesis outlines

This thesis contains seven chapters and ten appendices. The first chapter begins with a background and significance of the problem. Then, literature reviews of relevance to the thesis, the objective, benefits and scope of this study. Chapter 2 presents is a theories including biomass gasification, characteristics of producer gas and gas engine fundamentals. Chapter 3 explains modification to a small producer gas engine, a diesel engine converted to a spark ignition engine. This detail chapter shows original engine configuration, components modification of the engine and specification of producer gas engine after modified. Chapter 4 describes the thermodynamic model of producer gas engine performance which is zero dimensional model. Chapter 5 gives the experimental methodology, providing detail on the experimental apparatus, instrument, and experimental procedure. Chapter 6 reports the results and discusses performance of producer gas engine, a comparison with diesel engine, emission of producer gas and diesel engine and prediction of performance of small engine with producer gas as fuel. Finally, Chapter 7 presents conclusions and suggestion for future developments.