APPENDIX

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APPENDIX A

Calculation Method and Testing Result

A.1 Calculation calorific value of producer gas

Gas	Heating value of gas (kJ/kg mol)
CO	282,990
H_2	285,840
CH_4	890,360

Find volume of producer gas from ideal gas at 1 atm and 25 ˚C

$$PV = RT$$

when $R = 8.314\ kJ/kg\ mole\ K$

$T = 25\ ℃\ (298\ K)$

$P = 1.013\ kN/m^2$

Therefore $V = (8.314\ \times 298)/1.013 \times 10^3$

$= 24.46\ m^3/kg\ mole$

The combustible of producer gas consist of CO 30.86%, H_2 8.5%, CH_4 0.34%

Calorific value of producer gas

$$HV_{pg} = \frac{Mole\ fraction \times\ Heating\ value\ of\ gas}{V}$$

$$HV_{pg} = \frac{(0.3086 \times 282990) + (0.085 \times 285840) + (0.0034 \times 890360)}{24.46}$$

$$HV_{pg} = 4646.71\ kJ/Nm^3$$

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A.2 Composition and calorific value of producer gas from charcoal

	Example	CO (%)	H_2 (%)	CH_4 (%)	O_2 (%)	CO_2 (%)	N_2 (%)	Calorific value (kJ/Nm3)
Charcoal	No.1	28.6	9.6	0.61	6.39	5.28	49.52	4,653.78
	No.2	28.5	9.7	0.62	6.7	4.9	49.58	4,656.96
	No.3	28.7	9.7	0.61	6.22	4.95	49.82	4,678.54
	No.4	28.5	9.8	0.6	6.65	4.99	49.46	4,661.36
	No.5	28.4	9.66	0.58	6.82	5.15	49.39	4,626.15
	No.6	28.4	9.58	0.57	6.92	5.24	49.29	4,613.16
	No.7	28.6	9.5	0.6	6.45	4.92	49.93	4,637.87
	No.8	28.5	9.6	0.62	6.35	4.98	49.95	4,645.27
Average		28.53	9.64	0.60	6.56	5.05	49.62	4,646.64

A.3 Calculation performance of producer gas and diesel engine

Testing conditions and calculated

Ambient pressure	0.92 kPa
Air density	1.1 kg/m^3
Air temperature	32 ± 5˚C
Cylinder	Single cylinder
Compression ratio	14:1
Swept volume	5.98×10^{-3} m^3
Engine speed	1700 rpm
Load	Full load
Spark ignition timing	45 degree
Fuel/ Calorific value	Producer gas/4646.71 kJ/Nm^3
Producer gas flow rate	0.0289 m^3/s, 10.41 m^3/hr
Force from load cell	8 kg, 11.74 kg
Radius of torque	0.23 m
Producer gas/charcoal ratio (σ)	0.2271 kg/m^3
Diesel heating diesel	45560.48 kJ/m^3

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1. Calculation performance of producer gas engine

(Cavity combustion chamber)

1.1 Torque $(T_b), (Nm)$

$$T_b = F \times r$$

$$T_b = 8 \times 9.81 \times 0.23$$

$$T_b = 18.05\ Nm$$

1.2 Brake power $(P_b), (W)$

$$P_b = 2\pi N T_b$$

$$P_b = 2 \times \pi \times 18.05 \times (1700/60)$$

$$P_b = 3214\ W$$

1.3 Brake thermal efficiency $(BTE), (\%)$

$$BTE = (P_b/\dot{V}_{pg}\, HV_{pg}) \times 100$$

$$BTE = (3.214 \times 100)/(0.0289 \times 4646.71)$$

$$BTE = 23.90\ \%$$

1.4. Brake mean effective pressure $(bmep), (kPa)$

$$bmep = 120\ P_b/V_d\ N$$

$$bmep = (120\ \times 3214)/(598 \times 10^{-6} \times 1700 \times 1000)$$

$$bmep = 379.13\ kPa$$

1.5 Brake specific fuel consumption $(BSFC), (kg/kWh)$

$$BSFC = \dot{m}_b/P_b$$

$$\dot{m}_b = \dot{m} \times \sigma$$

$$\dot{m}_b = 10.41 \times 0.2271$$

$$\dot{m}_b = 2.36\ kg/h$$

$$BSFC = 2.36/3.214$$

$$BSFC = 0.74\ kg/kWh$$

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1.6 Brake specific energy consumption $(BSEC), (kJ/kWh)$

$$BSEC = \dot{V_{pg}} HV_{pg}/P_b$$

$$BSEC = (10.41 \times 4646.81)/(3.214 \times 1000)$$

$$BSEC = 15.07\ MJ/kWh$$

2. Calculation performance of diesel engine

2.1 Torque $(T_b), (Nm)$

$$T_b = F \times r$$

$$T_b = 11.74 \times 9.81 \times 0.23$$

$$T_b = 26.49\ Nm$$

2.2 Brake power $(P_b), (W)$

$$P_b = 2\pi N T_b$$

$$P_b = 2 \times \pi \times 26.49 \times (1700/60)$$

$$P_b = 4717\ W$$

2.3 Brake thermal efficiency $(BTE), (\%)$

$$BTE = (P_b/\dot{m_f}\, LHV_{Di}) \times 100$$

$$BTE = (4.717 \times 100)/(0.000384 \times 45560.48)$$

$$BTE = 26.95\ \%$$

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2.4. Brake mean effective pressure $(bmep), (kPa)$

$$bmep = 120\ P_b/V_d\ N$$

$$bmep = (120 \times 4717)/(598 \times 10^{-6} \times 1700 \times 1000)$$

$$bmep = 556.45\ kPa$$

2.5 Brake specific energy consumption $(BSEC), (MJ/kWh)$

$$BSEC = \dot{m_f} LHV_{Di}/P_b$$

$$BSEC = (1.38 \times 45560.48)/(4.717 \times 1000)$$

$$BSEC = 13.36\ MJ/kWh$$

A.4 Experimental data of small producer gas engine performance (bath tub combustion chamber at 14:1 of CR)

Engine speed (rpm)	Load (%)	Spark timing (Degree)	$\dot{V}_{pg}$ (m^3/h)	$\dot{m}_b$ (Kg/h)	T_b (Nm)	P_b (W)	BTE (%)	$BSFC$ (Kg/kWh)	$BSEC$ (MJ/kWh)	$bmep$ (kPa)
1100	20	25	4.87	1.10	4.42	0.50	8.09	2.18	44.50	92.88
1100	40		5.19	1.18	5.14	0.59	8.83	1.99	40.76	108.05
1100	60		5.27	1.19	5.66	0.65	9.58	1.84	37.59	118.95
1100	80		5.35	1.21	6.38	0.73	10.64	1.65	33.85	134.11
1100	100		5.43	1.23	6.94	0.80	11.40	1.54	31.57	145.96
1300	20	30	7.03	1.59	5.97	0.81	8.96	1.96	40.17	125.58
1300	40		7.43	1.69	7.10	0.96	10.08	1.75	35.72	149.28
1300	60		7.83	1.78	8.39	1.14	11.30	1.56	31.87	176.29
1300	80		8.31	1.89	9.72	1.32	12.33	1.43	29.19	204.25
1300	100		8.63	1.96	11.05	1.50	13.50	1.30	26.66	232.21
1500	20	35	8.23	1.87	6.70	1.05	9.90	1.78	36.36	140.75
1500	40		8.47	1.92	8.12	1.27	11.66	1.51	30.87	170.60
1500	60		9.03	2.05	9.86	1.54	13.28	1.33	27.11	207.10
1500	80		9.59	2.18	12.29	1.93	15.59	1.13	23.09	258.28
1500	100		10.07	2.28	15.38	2.41	18.58	0.95	19.37	323.20
1700	20	40	11.11	2.52	8.23	1.46	10.22	1.72	35.23	196.04
1700	40		11.91	2.70	10.40	1.85	12.04	1.46	29.90	247.60
1700	60		12.63	2.87	12.56	2.23	13.72	1.28	26.24	299.16
1700	80		12.87	2.92	15.02	2.67	16.10	1.09	22.36	357.71
1700	100		13.67	3.10	18.61	3.31	18.77	0.94	19.18	443.10
1900	20	40	12.79	2.90	9.29	1.85	11.20	1.57	32.15	195.25
1900	40		13.27	3.01	11.71	2.33	13.60	1.29	26.48	245.96
1900	60		13.91	3.16	14.66	2.91	16.25	1.08	22.16	308.04
1900	80		14.95	3.39	18.36	3.65	18.93	0.93	19.02	385.76
1900	100		Engine knock							

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A.4 Experimental data of small producer gas engine performance (cavity combustion chamber at 14:1 of CR)

Engine speed (rpm)	Load (%)	Spark timing (Degree)	$\dot{V}_{pg}$ (m^3/h)	$\dot{m}_b$ (Kg/h)	T_b (Nm)	P_b (W)	BTE (%)	$BSFC$ (Kg/kWh)	$BSEC$ (MJ/kWh)	$bmep$ (kPa)
1100	20		3.50	0.79	3.80	0.43	9.70	1.82	37.13	79.85
1100	40		3.58	0.81	4.31	0.49	10.74	1.64	33.51	90.51
1100	60	35	3.66	0.83	4.77	0.55	11.64	1.51	30.93	100.23
1100	80		3.82	0.86	5.50	0.63	12.86	1.37	27.99	115.63
1100	100		3.74	0.85	6.16	0.71	14.70	1.20	24.49	129.37
1300	20		5.07	1.15	4.94	0.67	10.27	1.71	35.07	103.78
1300	40		5.31	1.20	5.91	0.80	11.73	1.50	30.70	124.16
1300	60	40	5.74	1.30	7.17	0.97	13.17	1.34	27.33	150.70
1300	80		6.06	1.37	8.89	1.21	15.46	1.14	23.28	186.72
1300	100		6.38	1.45	10.65	1.45	17.59	1.00	20.46	223.68
1500	20		6.36	1.44	5.62	0.88	10.74	1.64	33.51	118.00
1500	40		7.32	1.66	7.62	1.19	12.65	1.39	28.45	159.94
1500	60	45	8.04	1.82	10.04	1.57	15.19	1.16	23.70	210.89
1500	80		8.84	2.00	13.02	2.04	17.91	0.98	20.10	273.44
1500	100		9.32	2.11	15.34	2.41	20.03	0.88	17.98	322.26
1700	20		7.94	1.80	7.22	1.28	12.54	1.40	28.70	151.65
1700	40		8.66	1.96	9.47	1.68	15.09	1.17	23.85	199.04
1700	60	45	9.30	2.11	11.95	2.12	17.74	0.99	20.30	251.17
1700	80		9.94	2.25	14.77	2.63	20.51	0.86	17.55	310.41
1700	100		10.42	2.36	18.05	3.21	23.90	0.74	15.07	379.13
1900	20		8.61	1.95	7.81	1.55	13.98	1.26	25.74	183.53
1900	40		9.33	2.12	10.32	2.05	17.04	1.03	21.13	242.32
1900	60	50	9.65	2.19	12.77	2.54	20.38	0.86	17.66	299.79
1900	80		10.13	2.30	14.32	2.85	21.89	0.81	16.52	336.33
1900	100		Engine knock							

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A.4 Experimental data of diesel engine performance

Engine speed (rpm)	Load (%)	$\dot{m_f}$ (Kg/h)	T_b (Nm)	P_b (W)	BTE (%)	$BSFC$ (Kg/kWh)	$BSEC$ (MJ/kWh)	$bmep$ (kPa)
1100	20	0.26	4.38	0.50	14.86	0.53	24.22	92.10
1100	40	0.27	5.01	0.58	16.52	0.48	21.79	105.29
1100	60	0.27	5.35	0.62	17.55	0.45	20.51	112.48
1100	80	0.28	5.81	0.67	18.78	0.42	19.16	122.03
1100	100	0.28	6.43	0.74	20.39	0.39	17.66	135.22
1300	20	0.32	4.61	0.63	15.58	0.51	23.10	96.76
1300	40	0.35	5.53	0.75	17.14	0.46	21.00	116.11
1300	60	0.41	6.70	0.91	18.97	0.45	18.98	140.75
1300	80	0.42	8.01	1.09	20.72	0.38	17.37	168.24
1300	100	0.44	9.35	1.27	22.79	0.35	15.79	196.44
1500	20	0.44	5.78	0.91	16.50	0.48	21.82	121.48
1500	40	0.56	8.30	1.30	18.36	0.43	19.61	174.40
1500	60	0.62	10.23	1.61	20.47	0.39	17.59	214.84
1500	80	0.67	12.12	1.90	22.60	0.35	15.93	254.57
1500	100	0.69	13.72	2.16	24.76	0.32	14.54	288.14
1700	20	1.14	14.33	2.55	17.65	0.45	20.40	300.93
1700	40	1.20	16.81	2.99	19.68	0.40	18.30	352.99
1700	60	1.24	19.48	3.47	22.04	0.36	16.34	409.07
1700	80	1.34	23.32	4.15	24.44	0.32	14.73	489.79
1700	100	1.38	26.49	4.72	26.95	0.29	13.36	556.45
1900	20	1.98	22.21	4.42	18.87	0.45	19.08	466.41
1900	40	2.04	25.88	5.15	21.40	0.40	16.82	543.50
1900	60	2.08	29.65	5.90	24.07	0.35	14.96	622.80
1900	80	2.12	34.46	6.86	27.36	0.31	13.16	723.90
1900	100	2.13	39.20	7.80	31.02	0.27	11.60	823.42

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APPENDIX B

Specific Heat of Producer Gas Composition

B.1 Specific heat of carbon monoxide gas (CO)

Carbon monoxide gas (CO)	
Temperature (T) (K)	Specific Heat (C_p) (kJ/kg-K)
175	1.039
200	1.039
225	1.039
250	1.039
275	1.04
300	1.04
325	1.041
350	1.043
375	1.045
400	1.048
450	1.054
500	1.064
550	1.075
600	1.087
650	1.1
700	1.113
750	1.126
800	1.139
850	1.151
900	1.163
950	1.174
1000	1.185
1050	1.194
1100	1.203
1150	1.212
1200	1.22
1250	1.227
1300	1.234
1350	1.24

Carbon monoxide gas (CO)	
Temperature (T) (K)	Specific Heat (C_p) (kJ/kg-K)
1400	1.246
1500	1.257
1600	1.267
1700	1.275
1800	1.282
1900	1.288
2000	1.294
2100	1.299
2200	1.304
2300	1.308
2400	1.311
2500	1.315
2600	1.318
2700	1.321
2800	1.324
2900	1.326
3000	1.329
3500	1.339
4000	1.346
4500	1.353
5000	1.359
5500	1.365
6000	1.37

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B.2 Specific heat of hydrogen (H_2)

Hydrogen gas (H_2)	
Temperature (T) (K)	Specific Heat (C_p) (kJ/kg-K)
175	13.12
200	13.53
225	13.83
250	14.05
275	14.2
300	14.31
325	14.38
350	14.43
375	14.46
400	14.48
450	14.5
500	14.51
550	14.53
600	14.55
650	14.57
700	14.6
750	14.65
800	14.71
850	14.77
900	14.83
950	14.9
1000	14.98
1050	15.06
1100	15.15
1150	15.25
1200	15.34
1250	15.44
1300	15.54
1350	15.65
1400	15.77
1500	16.02
1600	16.23
1700	16.44
1800	16.64
1900	16.83

Hydrogen gas (H_2)	
Temperature (T) (K)	Specific Heat (C_p) (kJ/kg-K)
2000	17.01
2100	17.18
2200	17.35
2300	17.5
2400	17.65
2500	17.8
2600	17.93
2700	18.06
2800	18.17
2900	18.28
3000	18.39
3500	18.91
4000	19.39
4500	19.83
5000	20.23
5500	20.61
6000	20.96

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B.3 Specific heat of methane (CH_4)

Methane gas (CH_4)	
Temperature (T) (K)	Specific Heat (C_p) (kJ/kg-K)
200	2.087
225	2.121
250	2.156
275	2.191
300	2.226
325	2.293
350	2.365
375	2.442
400	2.525
450	2.703
500	2.889
550	3.074
600	3.256
650	3.432
700	3.602
750	3.766
800	3.923
850	4.072
900	4.214
950	4.348
1000	4.475
1050	4.595
1100	4.708

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B.4 Specific heat of oxygen (O_2)

Oxygen gas (O_2)	
Temperature (T) (K)	Specific Heat (C_p) (kJ/kg-K)
175	0.910
200	0.910
225	0.911
250	0.913
275	0.915
300	0.918
325	0.923
350	0.928
375	0.934
400	0.941
450	0.956
500	0.972
550	0.988
600	1.003
650	1.017
700	1.031
750	1.043
800	1.054
850	1.065
900	1.074
950	1.082
1000	1.090
1050	1.097
1100	1.103
1150	1.109
1200	1.115
1250	1.120
1300	1.125
1350	1.130
1400	1.134
1500	1.143
1600	1.151
1700	1.158
1800	1.166
1900	1.173

Oxygen gas (O_2)	
Temperature (T) (K)	Specific Heat (C_p) (kJ/kg-K)
2000	1.181
2100	1.188
2200	1.195
2300	1.202
2400	1.209
2500	1.216
2600	1.223
2700	1.230
2800	1.236
2900	1.243
3000	1.249
3500	1.276
4000	1.299
4500	1.316
5000	1.328
5500	1.337
6000	1.344

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B.5 Specific heat of carbon dioxide (CO_2)

Carbon dioxide (CO_2)	
Temperature (T) (K)	Specific Heat (C_p) (kJ/kg-K)
175	0.709
200	0.735
225	0.763
250	0.791
275	0.819
300	0.846
325	0.871
350	0.895
375	0.918
400	0.939
450	0.978
500	1.014
550	1.046
600	1.075
650	1.102
700	1.126
750	1.148
800	1.168
850	1.187
900	1.204
950	1.220
1000	1.234
1050	1.247
1100	1.259
1150	1.270
1200	1.280
1250	1.290
1300	1.298
1350	1.306
1400	1.313
1500	1.326
1600	1.338
1700	1.348
1800	1.356
1900	1.364

Carbon dioxide (CO_2)	
Temperature (T) (K)	Specific Heat (C_p) (kJ/kg-K)
2000	1.371
2100	1.377
2200	1.383
2300	1.388
2400	1.393
2500	1.397
2600	1.401
2700	1.404
2800	1.408
2900	1.411
3000	1.414
3500	1.427
4000	1.437
4500	1.446
5000	1.455
5500	1.465
6000	1.476

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B.6 Specific heat of nitrogen (N_2)

Nitrogen (N_2)	
Temperature (T) (K)	Specific Heat (C_p) (kJ/kg-K)
175	1.039
200	1.039
225	1.039
250	1.039
275	1.039
300	1.040
325	1.040
350	1.041
375	1.042
400	1.044
450	1.049
500	1.056
550	1.065
600	1.075
650	1.086
700	1.098
750	1.110
800	1.122
850	1.134
900	1.146
950	1.157
1000	1.167
1050	1.177
1100	1.187
1150	1.196
1200	1.204
1250	1.212
1300	1.219
1350	1.226
1400	1.232
1500	1.244
1600	1.254
1700	1.263
1800	1.271
1900	1.278

Nitrogen (N_2)	
Temperature (T) (K)	Specific Heat (C_p) (kJ/kg-K)
2000	1.284
2100	1.290
2200	1.295
2300	1.300
2400	1.304
2500	1.307
2600	1.311
2700	1.314
2800	1.317
2900	1.320
3000	1.323
3500	1.333
4000	1.342
4500	1.349
5000	1.355
5500	1.362
6000	1.369

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APPENDIX C

Publication and Conferences

C.1 Papers in International Journals and conferences

(1) Homdoung , N., Tippayawong , N., Dussadee , N., "Effect of ignition timing advance on performance of a small producer gas engine", International Journal of Applied Engineering Research, Vol. 9, 2014, 2341-2348.

(2) Homdoung , N., Tippayawong , N., Dussadee , N., "Performance investigation of a modified small engine fueled with producer gas", Maejo International Journal of Science and Technology, Vol. 9, 2015, 10-20.

(3) Homdoung , N., Tippayawong , N., Dussadee , N., "Performance and emissions of a modified small engine operated on producer gas", Energy Conversion and Management, Vol. 96, 2015, 286-292.

(4) Homdoung , N., Tippayawong , N., Dussadee , N., "Prediction of small spark ignited engine performance using producer gas as fuel", Case study in thermal Engineering, Vol. 5, 2015, 98-103.

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(5) Homdoung , N., Tippayawong , N., Dussadee , N., "Performance investigation of a small engine fueled with producer gas and diesel in dual fuel operation", The 3rd TSME International Conference on Mechanical Engineering, 24-27 October, Chiang Rai, Thailand, 2012.

International Journal of Applied Engineering Research
ISSN 0973-4562 Volume 9, Number 13 (2014) pp. 2341-2348
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Effect of Ignition Timing Advance on Performance of a Small Producer Gas Engine

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Abstract

In this work, a small, single cylinder, naturally aspirated, compression ignition engine was modified into a spark ignited (SI) engine where producer gas was used solely as fuel. Experiments were carried out at various engine speeds and loads to study effect of ignition timing adjusted to maximum brake torque (MBT) on overall engine performance. From the tests, it was found that coefficient of variation from representative measurements was in a range of 1. 75-3. 0%. As expected, the performance of the engine was dependent on ignition timing advance. The optimum ignition timing of the small producer gas engine was observed to be between 20° to 25° BTDC at 1100 rpm, and increase with engine speed. Maximum brake mean effective pressure and minimum brake specific fuel consumption rate were 195. 48 kPa, and 0. 93 kg/kWh, respectively, obtained at 1700 rpm on full load. At this condition, brake thermal efficiency of about 19% was achieved.

Keywords : Biomass, Ignition timing, Small engine, Producer gas, Renewable energy

1. Introduction

Limitation of conventional fossil fuel reserves and reduction of environmental impact have intensified the search for alternative fuels in internal combustion engines. Renewable fuel is an obvious solution to this problem. Biomass derived, producer gas is an interesting source that can be the fuel of choice in the future. The producer gas derived from biomass via gasification has average composition consisting of 4-10 % H_2, 28-32 % CO, 0-2 % CH_4, 1-3 % CO_2 and 55-65 % N_2 with mean calorific value of about 4500 – 5600 kJ/Nm^3 [1]. The stoichiometric air to fuel ratio is 1. 25 ± 0. 05 on mass basis. The laminar flame speed is in a range of 10-12 cm/s [2]. However, when use in an engine, the power output and efficiency were reported to decrease, compared to a typical liquid fuel [3]. Adjusting ignition timing may improve the engine performance. With respect to previous works on ignition timing effect on performance of SI engines, Lawankar et al. [4] tested a medium sized, SI engine with gasoline and LPG. They found optimum ignition timing of the engine to be 20° BTDC for gasoline and 30° BTDC for LPG, respectively. Gopal et al. [5] reported appropriate ignition timing for CNG and gasoline engines in which the maximum brake thermal efficiency occurred at 27° BTDC for CNG, and at 32° BTDC for gasoline. For CNG, duration of the burn was needed to increase due to slower flame speed [6]. Kakaee et al. [7] reported similar ranges to Lawankar et al [4] and Gopal et al. [5]. Shidhar et al. [2] worked on varying ignition timing of a range of SI engines with producer gas operation at high compression ratio (CR) mode. Appropriate ignition timings were identified. Works on SI engines on different gases such as methane and landfill gas [8], biogas [9] and hydrogen [10] were also available.

To the authors' knowledge, it is clear that currently there is no report on small engines with producer gas operation. It is therefore the focus of this work to investigate if improvement can be achieved with adjustment of the ignition timing advance for a small producer gas engine.

2. Material & Methods

2. 1 Apparatus

In this study, producer gas was generated from a downdraft gasifier, shown in Figure 1. The reactor can generate producer gas up to 27 Nm^3/h. Charcoal consumption rate was between 5-6 kg/h. The gas cleaning and cooling unit consists of a cyclone, a water scrubber, an organic filter and a fabric filter. Tar and particulate matter before entering to the engine were less than 50 mg/Nm^3. The modified engine was of a single cylinder type, naturally aspirated, four-stroke and water cooling, and usually employed as an agricultural powertrain. Modification of the engine was conducted on the ignition system, cylinder head, and air-producer gas mixer. The optimum CR was achieved at 14:1. Ignition system was installed in place of a fuel injection system. The ignition timing can be varied in a range of 0° to 60° BTDC. The gas mixer design was based on air-gas carburetor and operating between 1000-2000 rpm.

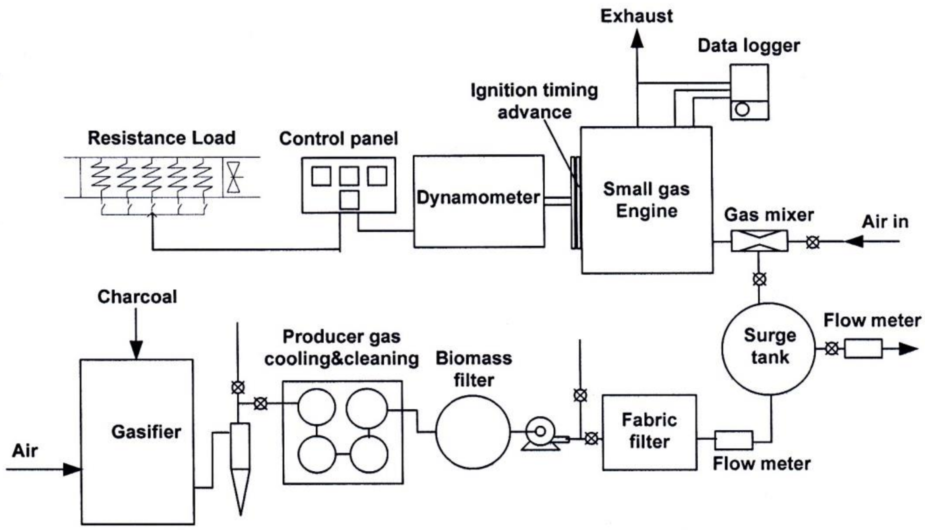

Figure 1: Schematic diagram of gasifier system

2. 2 Data analysis

A Shimadzu GC-8A gas chromatography machine was used to measure mole fractions of CO, H_2, CH_4, CO_2 and N_2 in the producer gas. Average chemical compositions were found to be CO = 30. 5±2%, H_2 = 8. 5 ± 2%, CH_4 = 0. 35%, CO_2 = 4. 8±1%, and O_2 = 6. 3±0:5%. Calculated calorific value of the producer gas was 4. 64 MJ/Nm^3. The density of charcoal was about 250–300 kg/m^3 with average moisture content of 7%. The experiment conditions were at ambient pressure of 0. 92 kPa. Average air density was 1. 1 kg/m^3. Ambient temperature during the testing period was 30 ± 3°C.

2. 3 Test procedures

Engine tests were carried out at varying ignition timings between 20°-50° BTDC. The engine speeds were in a range of 1100–1900 rpm on part load and full load mode. All experimental were done at the corresponding MBT. Air and fuel were tuned to achieve the maximum power. The measurements were recorded at an average interval of 10 min, after achieving a stable operation. Charcoal consumption at each load was monitored by weighing the mass of charcoal feeding into the gasifier. The producer gas and airflow rates were measured using Lutron YK-80 flow meters. F609 Chauvin Arnoux watt meter was used. Electrical load consists of ten 100W bulbs with ten 500W heaters. Temperatures of exhaust gas, water and oil lubricant were measured using type K of thermocouples connected to Yokokawa DX 220-1-2 data logger. The coefficient of variation (COV), specific fuel consumption (BSFC), brake mean effective pressure (BMEP), brake thermal efficiency (BTE), optimum ignition timing were evaluated.

3. Results and Discussion

General observation revealed that the exhaust gas temperature of small producer gas engine was in a range of 298-420°C, while water and oil temperatures were between 93-104°C. The exhaust gas, water and oil temperatures increased with increasing engine speed, due to increased fuel input to engine cylinders and sub segment increase of turbulence intensity, heat release rate, and maximum flame temperature [8]. They were stable throughout the tests.

3. 1 Coefficient of variation

A COV is a measure of cyclic variability that occurs during early stage of combustion and around peak pressure. Figure 2 shows variation of COV of BMEP with engine speed at 60% load and full load. For each speed, the ignition timing was adjusted to MBT timing. The COV of BMEP was found to vary between 1. 75 to 3. 0%. Minimum COV occurred at 1300 rpm. At higher engine speeds, the COV of BMEP 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 cylinder, the mixing of air-producer gas and residual gas in cylinder for each cycle [11]. In comparison between operation loads, the use of full load appeared to show higher COV than part load.

3. 2 Brake mean effective pressure

Figure 3 shows effect of ignition timing, engine speed and load on BMEP of the small producer gas engine. The results show that BMEP tended to increase with appropriate advance ignition timing that mostly depend 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 in cylinder will burn as the piston is moving down, leading to decreasing pressure and performance. With advanced ignition timing, the mixer in cylinder will burn while the piston is moving up in compression stroke. The best ignition timing found in this experiment on full load 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 appeared to show knocking. For 60% load, the best ignition timings were similar to the full load. The maximum BMEP (195. 48 kPa) occurred in full load at 1700 rpm, whereas the minimum of BMEP (64. 45 kPa) was obtained at 1100 rpm.

3. 3 Brake specific fuel consumption

Figure 4 shows variation of BSFC with adjusted ignition timing, engine speed and load of the small producer gas engine. The BSFC rate tends to decrease with ignition timing. In comparison of difference load and engine speed, the minimum BSFC rate occurred on full load operation and at 1700 rpm of engine speed. Increasing engine speed tended to decrease BSFC rate. The lowest BSFC rate of 0. 93 kg/kWh in small engine was achieved. Generally, the BSFC rate of producer gas engine was in a range between 1. 2-2 kg/kWh [12].

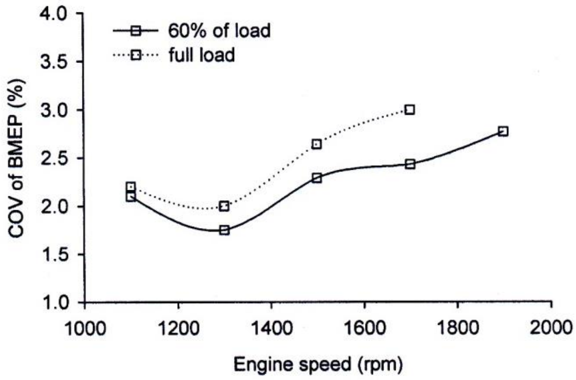

Figure 2: COV of BMEP with engine speed for two different loads

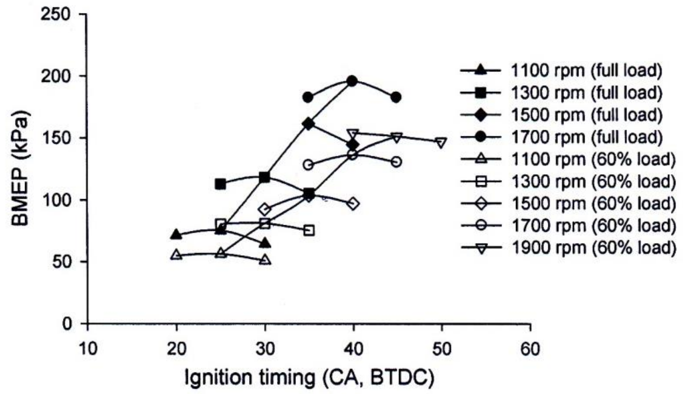

Figure 3: Relation of ignition timing, engine speed and load on brake mean effective pressure

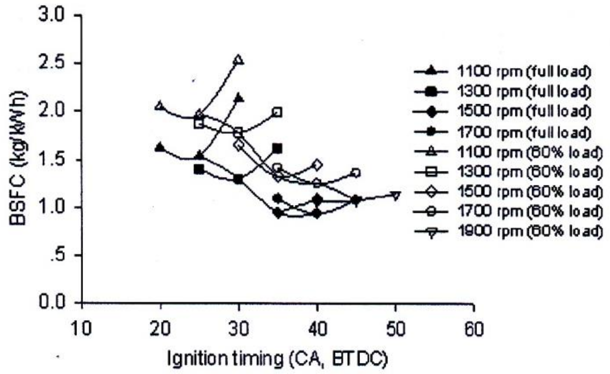

Figure 4: Relation of ignition timing, engine speed and load on brake specific fuel consumption

3. 4 Brake thermal efficiency

Using producer gas in small engines adjusted to suitable ignition timing, high BTE can be obtained. Adjusting ignition timing related to combustion process in cylinder directly affected the power output and fuel consumption. Figure 5 shows BTE as a function of ignition timing, engine speed and load. Maximum BTE of 18. 8% was obtained at highest engine speed on full load. This was in similar magnitude to those from medium and large engines. Typical thermal efficiency of large producer gas engines was in a range of 18-24 % [12, 13, 14].

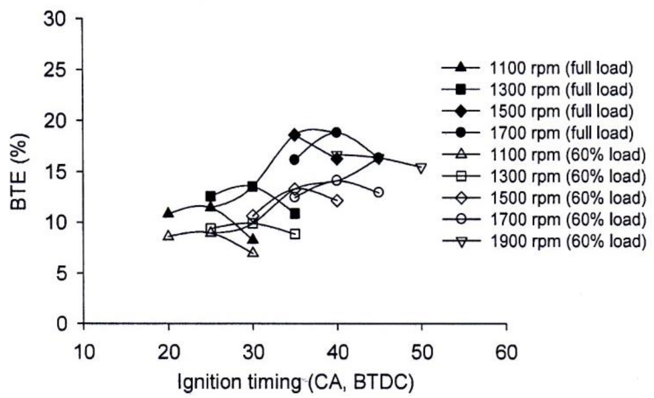

Figure 5: Relation of ignition timing, engine speed and load on brake thermal efficiency

3. 5 Optimum ignition timing

Figure 6 summarizes optimum ignition timing of the small producer gas engine obtained at each engine speed on part load and full load. The ignition timing tended to increase with engine speed because the air-producer gas mixture in cylinder was turbulent due to fast moving of gas. The burning time became shorter at higher engine speeds. So, it was necessary to increase the burn duration. At 1100 rpm, maximum power output occurred during 20° to 25° BTDC. Engine speed of 1500 rpm is interesting because most applications will 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. At 1900 rpm maximum speed, the small producer gas engine was unable to operate at full load due to deceleration and knocking when adjusted to 40° ignition timing advance. Meanwhile, the good acceleration stability was observed at 60 % of load or lower. Therefore, the best power output on mid load was expected to occur during 40° to 45° BTDC of ignition timing advance.

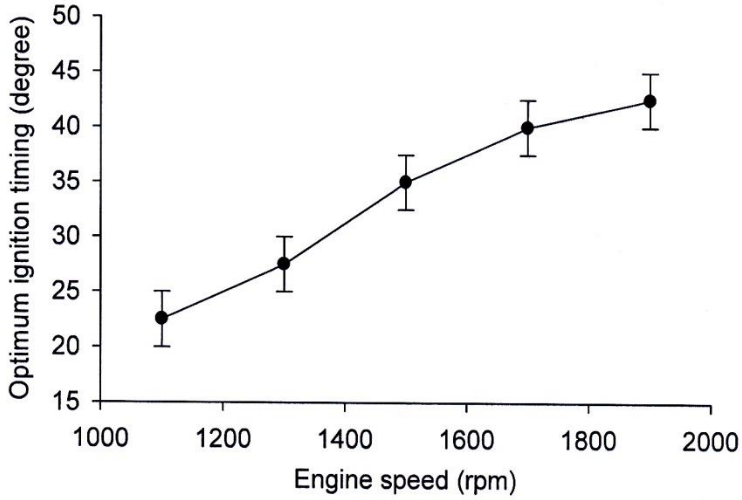

Figure 6: The optimum ignition timing of small producer gas engine with varying engine speed

4. Conclusions

From the investigation, it was found that a small agricultural engine can operate satisfactorily well with producer gas. Adjusting ignition timing can improve performance of the producer gas engine. In this work, the optimum ignition timing of the small producer gas engine were between 20° to 25° BTDC at 1100 rpm, 25° to 30° BTDC at 1300 rpm, 32. 5° to 37. 5° BTDC at 1500 rpm and 40° BTDC of 1700 rpm. Appropriate ignition timing advance enabled BMEP to increase. The maximum BMEP of 195 kPa was achieved at 1700 rpm of full load. At this speed, the lowest BSFC rate of 0. 93 kg/kWh and maximum BTE of the small producer gas engine was achieved.

Acknowledgment

The authors would like to thank the Graduate School and the Department of Mechanical Engineering of Chiang Mai University, as well as the Energy Research Centre of Maejo University for providing test facilities and technical supports. Financial supports from the Energy Policy and Planning Office, and the Commission on Higher Education were highly appreciated.

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Maejo Int. J. Sci. Technol. **2015**, *9*(01), 10-20

Maejo International Journal of Science and Technology

ISSN 1905-7873
Available online at www.mijst.mju.ac.th

Full Paper

Performance investigation of a modified small engine fueled with producer gas

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Received: 18 February 2014 / Accepted: 19 January 2015 / Published: 27 January 2015

Abstract: Producer gas from biomass gasification can be used as a replacement fuel in spark-ignition engines. In this study, a small, single-cylinder, naturally aspirated diesel engine was modified into a spark-ignition engine. A conventional swirl chamber was replaced by a bath tube combustion chamber. Optimum spark ignition time was set for each engine speed to give maximum brake torque. It was fueled with 100% producer gas and coupled to a 5.0-kW dynamometer. A downdraft gasifier was used to generate producer gas from charcoal. Engine performance in terms of engine torque, brake power, brake thermal efficiency and brake specific fuel consumption were evaluated at variable compression ratios between 9.7:1-17:1. Engine speed and load were varied between 1100-1900 rpm and 20-100% respectively. At a certain combination of compression ratio, engine speed and load, deceleration and knocking were detected. Maximum engine torque and brake power were 18.6 Nm and 3.3 kW respectively, at a compression ratio of 14:1, full load and 1700 rpm. The best specific fuel consumption of 0.94 kg/kWh and maximum brake thermal efficiency of about 19% were obtained.

Keywords: small engine, producer gas, compression ratio, spark ignition, renewable energy

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INTRODUCTION

Escalating oil prices and increasingly scarce fossil fuels, coupled with an exploding population, have created an energy crisis, especially in developing countries where machines are used in food production. In Thailand, the agricultural sector commonly uses small, internal combustion engines, with power and speed mostly in the range of 2.2-10.4 kW and 1000-2000 rpm respectively [1]. Farms use them for mechanical work, pumping, power generation and plowing. Using producer gas in engines offers an alternative energy source, reducing dependence on fossil fuels. However, producer gas poses a problem as more combustible carbon monoxide content is needed to produce a similar output to gasoline. This is because the engine operates at a lower thermal efficiency with power de-rated by more than 30% due to the lower energy density of producer gas compared to that of gasoline and diesel fuels [2].

Attempts to develop internal combustion engines, especially for producer gas as fuel, are ongoing, with three primary types: (i) spark ignition (SI) engines using gas, (ii) compression ignition (CI) engines using gas and diesel in dual fuel mode, and (iii) engines converted from CI to SI using 100% gas. Based on previous researches, converting a CI engine into an SI engine operated at medium and high levels of compression ratio (CR) shows promise. A number of studies of SI engines fueled by producer gas have been carried out. Parke and Clark [3] and Martin and Wauters [4] showed that the engine power was 34-50% less than gasoline engines at conventional CR [5]. Munoz et al. [6] reported test results on a small SI engine at a CR of 8.2: 1. A power de-rating of 50% was observed. Ando et al. [7] reported that SI engines using producer gas at a CR of 9.4:1 caused a 45% average power reduction at all engine speeds. Shah et al. [8] found that a small SI engine using producer gas at a low CR had 1.8 times less power than using gasoline. Dasappa et al. [9] studied the use of producer gas with a 100-kW SI engine at a CR of 9.7:1. The maximum thermal efficiency was 18% and at low CR the engine power was reduced.

Ramachandra [10] studied medium and high CRs in a converted SI engine and found that the engine ran smoothly, with power output reduced by 20% compared to the original CI engine [5]. Shasikantra et al. [11] converted a CI engine to operate as an SI engine with producer gas as fuel at a CR of 11:1. They obtained a high thermal efficiency in the range of 20-24%. Aung [12] adapted a producer gas engine converted from a CI engine at a CR of 10:1. The power and torque output were 40% less than that with diesel mode. Raman and Ram [13] reported on an SI engine using producer gas at a CR of 12:1. The maximum thermal efficiency was 21% at 85% of full load. Sridhar et al. [5] modified a CI engine into an SI engine and used producer gas as fuel at a CR of 17:1. The engine brake power was reduced by 20% and the maximum overall efficiency obtained was 21%.

Most of these studies used medium to large engines. There have been very few studies on small engines. The objective of this research is to analyse the performance of a small engine fueled with 100% producer gas and determine the most appropriate CR, load and engine speed.

MATERIALS AND METHODS

Experimental Set-up

A schematic diagram of the gas generator system used in this study is shown in Figure 1. The gas generator design is based on a downdraft gasifier [14], and configured to operate on charcoal or wood. It consists of a gasifier, a gas conditioner and gas filters. The producer gas can be produced with a charcoal consumption rate between 5-6 kg/h. The efficiency of the gasification system is 70-75% and can generate up to 27 Nm^3/h of producer gas. The conditioning system improves the quality of the producer gas to ensure that the engine runs smoothly. The gas conditioning system consists of a heat exchanger, cyclone, Venturi scrubber, tar box, moisture separator, biomass filter, fabric filter and paper filter. The set-up also includes a water treatment plant for closed-loop water re-circulation.

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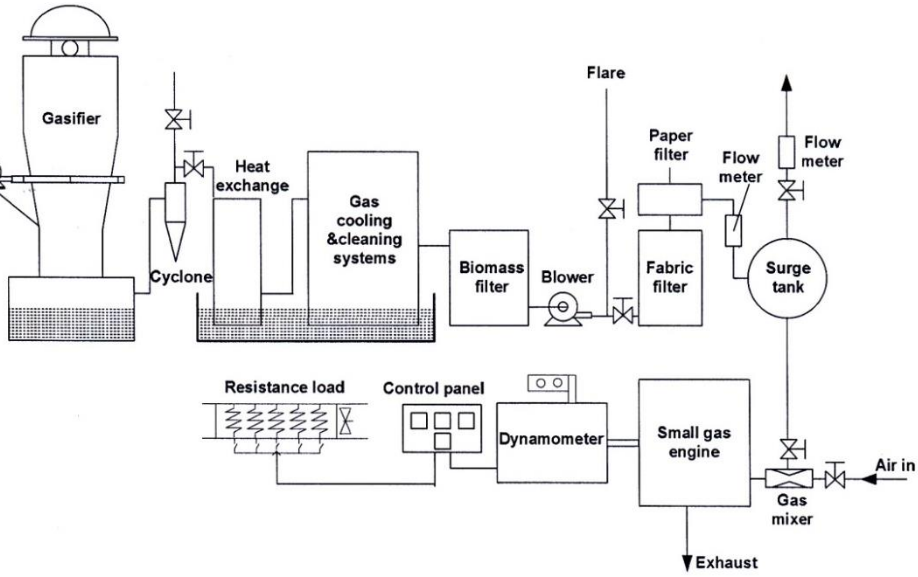

Figure 1. Schematic diagram of the experimental set-up for gas generator system

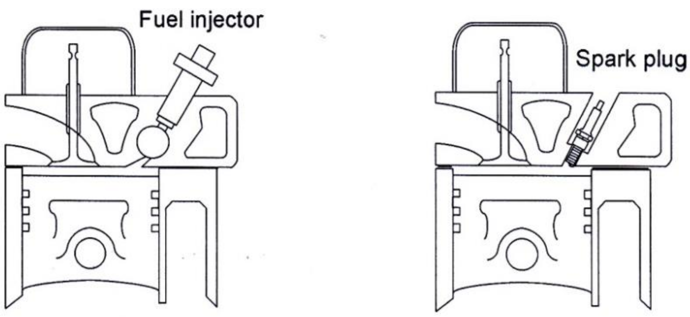

Figure 2. Small producer gas engine before and after modification of cylinder head

Engine Modification

A conventional, small, agricultural, water-cooled diesel engine with a CR of 21:1 was used in this experiment. The four-stroke, single cylinder, indirect injection engine was capable of producing a maximum power output of 8.2 kW. The engine specifications are given in Table 1. For the producer gas feeding system, a gas mixer was designed, manufactured and installed. The original diesel injection system was replaced with a spark plug as shown in Figure 2. The distributor and ignition coil were taken from a Mitsubishi 4G15 engine. The vacuum and centrifugal advances were disabled because the engine ran at a constant speed. The distributor was modified by replacing the magnetic pick-up with a spark timing plate stuck to the flywheel. The spark-ignition timing could be adjusted between 0-60°. The CR was adjusted to a range of 9.7-17: 1. Variable CR was achieved by using a thicker head gasket (between 4.7-8.2 mm). The volumes of the cylinder head and piston head were measured using a hypodermic syringe with low-viscosity oil. The cylinder head bolts and push rods were modified and the stoichiometric ratio of air to producer gas was approximately 1: 1.2. This volume ratio was used in the design of the gas mixer, which was based on Janisch [15] and used to supply the engine operating between 1000-2000 rpm with the appropriate mixture of air and gas. The air mixer was a Venturi with a throat diameter of 25 mm. Producer gas and air could be controlled by adjusting two screws.

Table 1. Specifications of original engine dynamometer set-up

Engine make, model	Kubota, ET11
Engine power	8.2 kW
Bore × Stroke	92×90 mm^2
Number of cylinder	1
Engine arrangement	Horizontal
Type of cooling	Water, thermo siphon system
CR	21:1
Combustion chamber	Pre-chamber
Ignition system	Compression ignition
Alternator efficiency	85%

Experimental Apparatus and Procedure

All experiments involving the engine were performed only after the gasifier system stabilised, normally about 1 hour from start-up. The stability of the gasifier system was achieved when the temperatures of the gasification zone and burner flame stabilised. The gas generator was operated using charcoal (size 25×25×25-50×50×50 mm according FAO [2] and Shaw [16]) which was available locally. Its density and average moisture content were measured based on ASTM C373-88 and ASTM D 2016-74 [17] and were found to be 250-300 kg/m^3 and 7% respectively. The gas composition was determined at random intervals using a Shimadzu GC-8A gas chromatography fitted with a ShinCarbon ST Micropacked column and a thermal conductivity detector. The conditions used were similar to those reported previously [18, 19]. The average chemical composition was 30.5±2% CO, 8.5±2% H_2, 0.35% CH4, 4.8±1% CO_2, 6.3±0.5% O_2 and N_2 (balance). The calculated mean calorific value of the producer gas was 4.64 MJ/Nm^3. The tar and particulate matter in the producer gas was measured according to Hasler et al. [20] and found to be less than 50 mg/Nm^3. Experiments were conducted at CRs of 9.7:1, 14:1 and 17:1. A higher CR engine using producer gas is of interest as it might offer a higher efficiency with better tolerance to knocking. Modifying an engine to have a higher CR is straightforward by simply decreasing the thickness of the cylinder head gasket. Engine tests were carried out by varying engine speeds with rpm and loading range of 1100-1900 and 20-100% respectively. The data were acquired at the corresponding maximum brake torque timing for each 1100, 1300, 1500, 1700 and 1900 rpm of the engine speed test condition. The air and fuel were tuned to achieve maximum power and after a stable operation, several measurements were taken over an average of 10-min. interval. Charcoal consumption at different loads was monitored by weighing the amount fed into the gasifier. The producer gas and airflow rates were measured using a Lutron YK-80 flow meter. The electrical load consisted of ten 100W bulbs with ten 500W heaters; a F609 Chauvin Arnoux watt meter was used

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for monitoring the load. The engine torque was measured using a load cell. The brake power, thermal efficiency and fuel consumption were evaluated using the following equations [21]:

$$P = 2\pi N\tau \qquad (1)$$

where P is the brake power, τ is the engine torque (Nm) and N is the engine speed (s^{-1});

$$BSFC = \frac{m_f^{\bullet}}{P} \qquad (2)$$

where *BSFC* is the brake specific fuel consumption and $m_f^{\bullet}$ is the mass flow rate of biomass (kg/h);

$$BTE = \frac{P}{V_{pg}^{\bullet} LHV_{pg}} \qquad (3)$$

where *BTE* is the brake thermal efficiency, expressed as ratio of the output power to the power supplied by the fuel, $V_{pg}^{\bullet}$ is the producer gas flow rate (m^3/s) and LHV_{pg} is the lower heating value of the producer gas (MJ/Nm^3).

RESULTS AND DISCUSSION

Gas Engine Operation

Table 2 provides a general overview of operation of a small engine with producer gas. It is representative of the results of analysing the engine performance. It can be observed that, at a low CR (9.7:1), the engine was able to be gradually loaded and stabilised up to 1500 rpm. With increasing engine speed, acceleration was good and the engine power increased. The engine decelerated and became unstable when the speed was increased to 1700-1900 rpm. The observed deceleration might be due to a reduced energy density compared to gasoline. The low CR of the engine might cause a lower pressure inside the combustion chamber [22] and affect flammability of the producer gas [7]. The lower volumetric efficiency might be reduced for gaseous fuel operation compared to conventional liquid fuels [23]. At a medium CR (14:1), however, the engine was observed to have good acceleration stability and its power increased with speed, although knocking occurred at full load and 1900 rpm. Finally, at a high CR (17:1), the small engine operated well between 1100-1500 rpm, but severe knocking symptoms occurred at 1700-1900 rpm and 80-100% of full load. Knocking might result from the increasing compression ratio, as well as increasing load and engine speed, leading to an increase in gas density, temperature and ignition lag in the combustion chamber [21].

Engine Brake Torque

Figure 3 shows the variation in engine torque of the small producer gas engine at 1500 rpm with different engine loads and CRs. A maximum torque of 15.38 Nm was obtained at CR = 14:1 and full load. For all CRs, the brake torque was 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 is that, compared to low CR, the work in expansion stroke exceeds that in the compression stroke [13]. At high CR, the engine torque was low due to abnormal combustion, leading to knocking [24]. Comparing engine torque versus speed at full load, the suitable CR for the small producer gas engine was found to be 14:1 at 1700 rpm and 18.61 Nm of maximum torque. At 1900 rpm, the engine was unable to operate due to severe knocking.

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Table 2. Operation of modified small engine fueled with producer gas at different test conditions

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

Note: ✓ = OK; x = Erratic; xx = Knocking

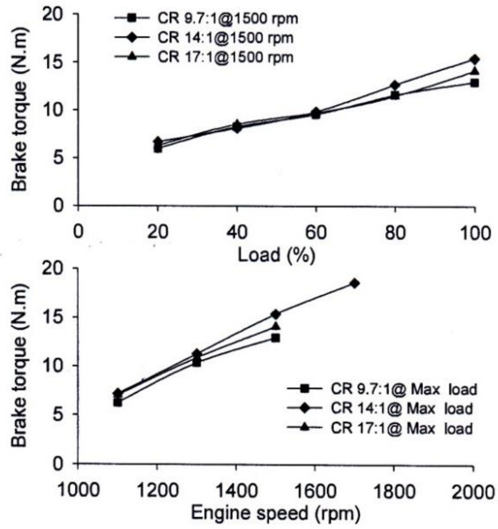

Figure 3. Engine brake torques at different loads and engine speeds

Brake Power

Figure 4 shows the effect of load and engine speed on the brake power for each CR considered. The engine brake power increased as engine load increased at all CRs. At 1500 rpm, an engine brake power of 2.41 kW was achieved at 14:1 of CR. The maximum engine brake power of 3.31 kW was achieved at 1700 rpm and medium CR.

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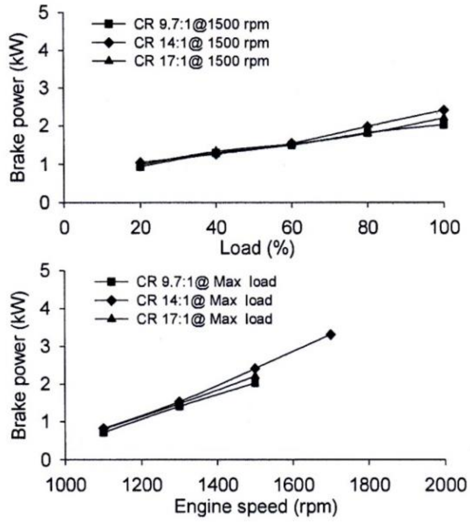

Figure 4. Engine brake power at different loads and engine speeds

Brake Thermal Efficiency

Figure 5 shows the BTE as a function of engine load and speed at different CRs. The efficiency tended to increase with engine load. This might be attributed to a better combustion of the relatively rich gas-air mixture at high loads. The BTE at medium CR was slightly higher than those at low and high CRs; reduction of BTE was due to a higher producer gas flow rate and poor combustion. At medium CR, a maximum BTE of 18.6% was obtained at full load. The small producer gas engine operated successfully at 1100-1500 rpm at both low and high CRs. The engine could operate up to 1700 rpm at medium CR, but at 1500-1700 rpm, the BTE tended to level off.

Brake Specific Fuel Consumption

The gasification rate from charcoal to producer gas was 25 Nm^3/h. The charcoal-to-gas conversion rate was arrived at by measuring the gas flow rate and fuel consumption rate. The specific charcoal consumption rate for the small producer gas engine was 0.94 kg/kWh. When the engine was operated at medium CR at full load (Figure 6), fuel consumption was reduced with increasing engine speed. The low and high CRs consumed more fuel than medium CR. Generally, the BSFC rate of the producer gas engine is in a range of 1.2-2 kg/kWh [9, 12]. At full load, the specific consumption rate decreased as engine speed increased. The lowest BSFC occurred between 1400-1500 rpm.

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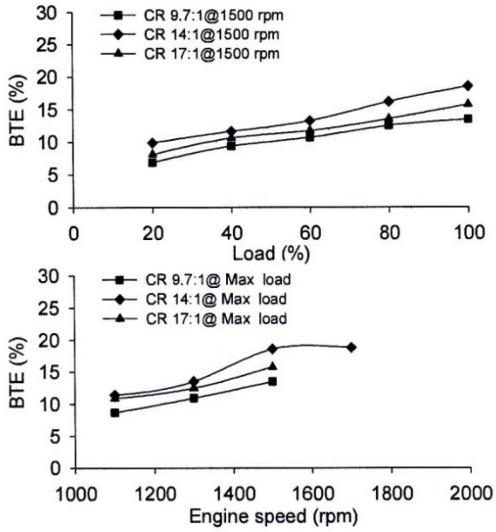

Figure 5. BTE at different loads and engine speeds

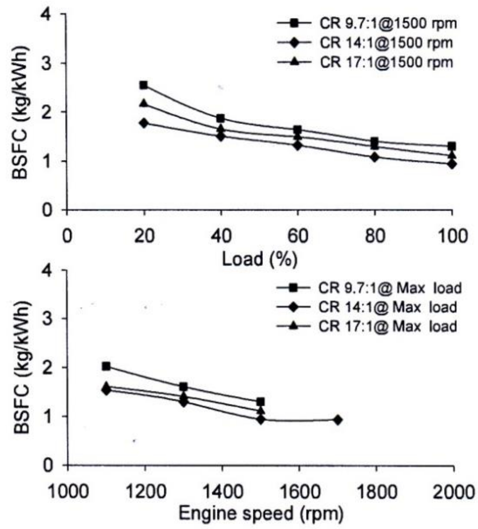

Figure 6. BSFC at different loads and engine speeds

Comparison with Previous Results

The performance of engines converted from CI or SI engines and fueled with producer gas at typical and high CRs, including that in this study, is summarised in Table 3. Most engines tested were large, with 2-6 cylinders and total engine displacement in the range of 1800-14000 cm^3, while that in this study was a small, single-cylinder engine with displacement of less than 600 cm^3. The CRs of the engines used were mostly low due to concerns about possible knocking [11] and the flexibility of using other fuels as primary fuel [12]. No sign of knocking at high CR was reported [9, 25]. Most reports on large engines did not provide information on torque and power. The overall efficiency of these large engines was in a range of 18-21%, which is similar to the efficiency values obtained in this work. The BSFC of our small engine was lower than those reported for the large engines.

Table 3. Performance of modified engines operated on producer gas

Performance specifications	**[5]**	**[9]**	**[12]**	**[13]**	**This study**
Engine power (kW)	28	283.48	26.5	99.2	8.2
Total displacement (cm^3)	3307	14000	1853	12316	598
Bore x Stroke (mm)	110x116	140x152	100x118	132x150	92x90
Number of cylinder	3	6	2	6	1
CR	17:1	8.5	10:1	12:1	14:1
Max torque/engine speed (Nm/rpm)	-	-	64/1400	-	18.6/1700
Max brake power/engine speed (kW/rpm)	-	-	12/1400	-	3.3/1700
BTE (%)	21	18	-	20.7	18.58
BSFC (kg/kWh)	-	1.36	2	1.2	0.94

Note: '-' = not available

CONCLUSIONS

We converted a small diesel engine into an SI gas engine. The modified engine successfully ran with 100% producer gas at high CRs. The most appropriate CR was 14:1 at full load with a maximum engine speed of 1700 rpm. The maximum engine torque and brake power was 18.61 Nm and 3.31 kW respectively.

ACKNOWLEDGEMENTS

We acknowledge support from Chiang Mai University and the Energy Policy and Planning Office, Ministry of Energy. The authors also thank the Energy Research Centre, Maejo University for providing test facilities.

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Energy Conversion and Management 94 (2015) 286–292

Contents lists available at ScienceDirect

Energy Conversion and Management

journal homepage: www.elsevier.com/locate/enconman

Performance and emissions of a modified small engine operated on producer gas

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ARTICLE INFO

Article history:
Received 27 October 2014
Accepted 28 January 2015

Keywords:
Biomass
Gas engine
Spark ignition
Producer gas
Compression ratio
Renewable energy

ABSTRACT

Existing agricultural biomass may be upgraded converted to a gaseous fuel via a downdraft gasifier for spark ignition engines. In this work, a 0.6 L, naturally aspirated single cylinder compression ignition engine was converted into a spark ignition engine and coupled to a 5 kW dynamometer. The conventional swirl combustion chamber was replaced by a cavity chamber. The effect of variable compression ratios between 9.7 and 17:1, and engine speeds between 1000 and 2000 rpm and loads between 20% and 100% of engine performance were investigated in terms of engine torque, power output, thermal efficiency, specific fuel consumption and emissions. It was found that the modified engine was able to operate well with producer gas at higher compression ratios than with gasoline. The brake thermal efficiency was lower than the original diesel engine at 11.3%. Maximum brake power was observed to be 3.17 kW, and the best BSFC of 0.74 kg/kWh was achieved. Maximum brake thermal efficiency of 23.9% was obtained. The smoke density of the engine was lower than the diesel engine, however, CO emission was higher with similar HC emission.

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1. Introduction

Energy is important in driving economic growth. Depletion of conventional energy sources and escalating fuel prices are causing an energy crisis. A possible solution may be found with renewable energies such as biomass, solar, hydropower and wind energy. Biomass is especially abundant, environmentally friendly and is an attractive substitute to fossil fuels. Biomass can be converted to producer gas by gasification, and utilized for generation of power and heat [1,2]. It has the potential to be used to drive internal combustion engines, compared with other forms of energy. Producer gas engines were first introduced around 1914–18, but was used widely during the World War II. More than one million of vehicles used producer gas in Europe, North America and Australia [3]. The use of producer gas in internal combustion engines was seen again during the oil crisis of 1973. However, the use of producer gas to run internal combustion engines, so for, has not been very successful because the power is usually de-rated during the operation. A major cause of lower performance with producer gas is due to its low energy density, compared to gasoline, diesel or natural gas [3,4]. The engine performance may be improved by two methods. One may improve quality of the fuel by focusing on increasing the content of hydrogen and carbon monoxide. This may be achieved by improving gasifier design, combustion processes, characteristics of biomass and quality control systems [5]. Alternatively, engine modifications that improve the use of producer gas may be undertaken. Most previous works on producer gas engines were conducted at compression ratio (CR) of about 10, either adapted directly from spark ignited (SI) engines or modified compression ignited (CI) engines. Munoz et al. [6] carried out tests of a small SI engine with producer gas, at the originally low CR. The power was found to be reduced by 50%, compared to gasoline usage. Similar findings were reported by Ando et al. [7] and Shah et al. [8]. Dasappa et al. [9] experimented on a 100 kW SI engine with producer gas for over 1000 continuous hour at CR of 8.5. The power output was found to reduce by 45%, while the maximum overall efficiency was 18%. Low volumetric efficiency and low energy density of the combustible mixture may be the main causes. Tsiakmakis et al. [10] studied a small SI engine with CR of 10 fueled with producer gas mixed with propane. A loss in power output by about 10% was reported for 55:45 mixture of producer gas and propane. At CR of 10.5, Shivapuji and Dasappa [11] who investigated combustion characteristics of internal combustion engines operated on producer gas reported a de-rating of only about 19% for a 76 kW turbocharged SI engine. Raman and Ram [12] reported test results of producer gas on an SI engine,

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http://dx.doi.org/10.1016/j.enconman.2015.01.078
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compared to natural gas operation at a CR of 12:1. The maximum overall efficiency was 21% at 85% of full load, while maximum power output was reduced by 12.4%.

For a 100% producer gas fueled SI engine, important modifications affecting engine performance would include changes to CR, spark ignition timing, air/fuel ratio and combustion chamber configuration [3,13–15]. Increasing CR was thought to give a lesser extent of power de-rating. A producer gas engine can operate at higher CR than a gasoline engine. The power output and thermal efficiency has been shown to rise by increasing the CR to those comparable to CI engine operation. However, limitation of knock still exists with producer gas operation [16]. Sridhar et al. [14,15] converted CI engines to operate as SI engines at CR of 11.5–17:1 with producer gas as fuel. For the large engine with CR = 12, power de-rating of 22–30% was reported. For the 24 kW engine with CR = 17, the overall efficiency achieved was reported to be 21%, with power output reduced by 17–19%. Homdoung et al. [17] modified a small agricultural CI engine into an SI engine with CR of 14. It was operated solely with producer gas, achieving a maximum brake thermal efficiency of about 19%.

Recent progress has been reported on producer gas utilization in SI engines with relatively high CR. However, there appeared to be a lack in research works regarding small engine development for producer gas. Therefore, the work was thought necessary to determine if a high CR small SI engine can operate well with producer gas. Thus, this work was interested in modifying a CI engine into an SI engine for producer gas with different CRs, comparable to diesel engine. Effect on its performance in terms of torque, power output, thermal efficiency, fuel consumption and emissions under varying loads and speeds was evaluated.

2. Methodology

2.1. Engine modification

In this experiment, a small agricultural CI engine was converted into SI engine and operated 100% on producer gas. The conventional engine was a small agricultural, diesel engine. It was an 8.2 kW, single cylinder, four strokes, indirect injection engine, 598 cc and CR of 21. (The detailed specifications of small producer gas engine and conventional diesel engine used in the experiment are shown in Table 1.) The modifications to the engine include changes to the combustion chamber, reduction of CR, mounting of ignition system in place of injector nozzle, and mounting of air–gas mixer.

The combustion chamber used for the producer gas engine had a cavity piston, adapted from the swirl chamber engine of the original diesel. The combustion chamber had a bowl in the piston and a flat cylinder head. This chamber was suitable for high CR and expected to provide high thermal efficiency. The symmetrical geometry of that chamber enabled minimum and near equal flame travel. Agitation was started by swirling the charge and completed by compression turbulence. CR was modulated to be in the range of 9.7–17. Variable CR was achieved by increasing the number of gaskets and extension in the range to 40–50 mm of a hollow in piston bowl. The cylinder head bolts and push rods were modified. Volumes of the cylinder head and piston head were measured using a hypodermic syringe with low viscosity oil.

Additional components of the spark ignition system consisted of a distributor, an ignition coil and spark plug. The spark ignition system selected was an electric ignition system, taken from a Mitsubishi 4G15 engine. The vacuum and centrifugal advances were disabled because the engine was run at a constant speed. Modification of the distributor was done by a magnet attached to the flywheel of the engine and a pick-up installed on the casing. When the magnet on flywheel rotated closed to the pick-up, a spark was initiated by a transistor and the ignition coil. Every revolutions of the engine provided a spark in combustion chamber. The spark ignition timing can be adjusted in a range of 0–60° TDC. For mounting of spark plug, the injector nozzle was removed. Auxiliary combustion chamber operated smoothly with new cylinder head. The gas mixer of the engine was of the venturi type. Air and producer gas was mixed before entering combustion chamber. The gas mixer was used to supply the suitable mixture of air and gas required for the engine, operating between 1000 and 2000 rpm and 25 mm of a throat diameter.

2.2. Experiment apparatus and setup

Charcoal from longan tree was used. It is found in Northern Thailand and has a high calorific value, compared to another charcoals [18]. The average density of charcoal was about 250–300 kg/m^3 with 7% moisture content. The heating value was 28,000 kJ/kg. The producer gas used in this study was from a fixed bed downdraft gasifier run at atmospheric pressure. The gasification system consists of a gasifier, a gas cooler and gas cleaner, shown in Fig. 1. The capacity of the gasifier in term of charcoal consumption was between 5 and 6 kg/h and could generate producer gas in a range of 25–30 Nm^3/h. The gas cooler was a heat exchanger installed in a 100 L water tank. Cooling was conducted between cold water and hot producer gas. The gas cleaner included a cyclone, a water scrubber kit, a moisture separator, a biomass filter, a fabric filter and a paper filter. The water scrubber kit was a venturi scrubber and a pack bed scrubber installed over the tar box remover. The closed-loop water treatment plant used a 335 W water pump. The producer gas composition was determined using Shimadzu GC-8A gas chromatography. The composition of the gas feed on the test engine was of CO 30.5 ± 2%, H_2 8.5 ± 2%, CH_4, 0.35%, CO_2 4.8 ± 1%, and O_2, 6.3 ± 0:5%, and balance nitrogen. The mean calorific value of the producer gas was 4.64 MJ/Nm^3. The tar and particulate matter measurements were carried out at the entrance of the engine. They were found to be lower than 50 mg/Nm^3. Charcoal consumption was measured by an electronic weighing balance. During experiments, the gasifier was filled with charcoal every 2.5 h. The measurement of producer gas flow rates was conducted using Lutron YK-80 flow meters before entering the engine.

The engine torque was measured by a dynamometer set and monitored by a display panel. The electrical loads were from ten 100 W bulbs with ten 500 W heaters. F609 Chauvin Arnoux watt meter was used. Emissions from the SI engine were tested using Koeng KEG 200 gas analyzer with Heshbon HBN 1500B to measured CO, HC and smoke density and as a comparison, with the original diesel engine, before modification. The diesel consumption was measured using JZA electronic-weighing scale gravimetric fuel

Table 1
Specifications of the small SI engine operated on producer gas and diesel engine.

	Modified engine	Original engine
Fuel	Producer gas	Diesel
Type	4 Stroke/naturally aspirated	4 Stroke/naturally aspirated
Bore × Stroke	92 × 90 mm	92 × 90 mm
Number of cylinder	Single cylinder	Single cylinder
Rated output	3.2 kW/1700 rpm	8.2 kW/1800 rpm
Rated speed	1900 rpm	2400 rpm
CR	9.7:1–17:1	21:1
Combustion chamber	Piston cavity	Swirl
Ignition system	Spark ignited	Compression ignited
Type of cooling	Water	Water
Loading device	Electrical generator	Electrical generator

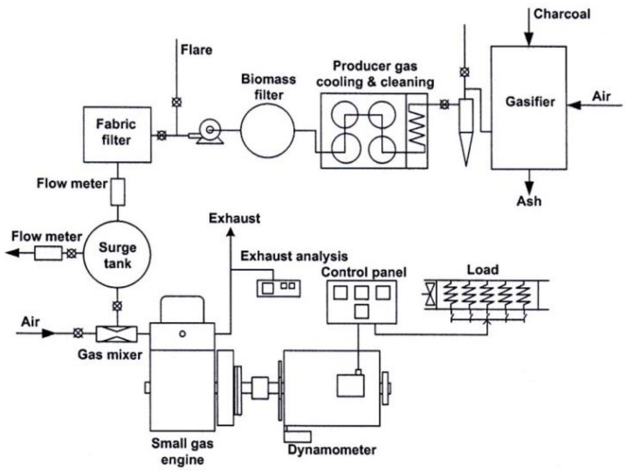

Fig. 1. Schematic diagram of experimental setup used in this study.

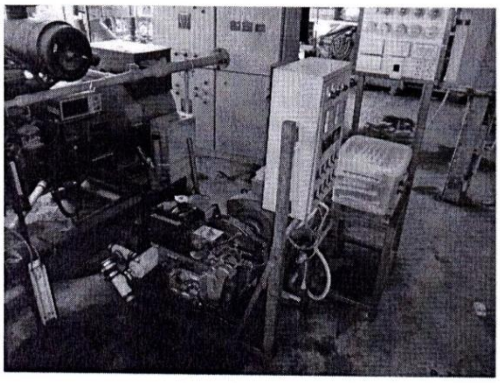

Fig. 2. Photograph of the small producer gas engine setup.

flow measurement. The photograph of the experimental setup is shown in Fig. 2.

2.3. Test procedure

Experimental investigations of the producer gas engine and diesel engine were carried out at different loads in a range of 20–100%. Three repeated experiments were conducted at each load. The engine speed was varied for 1100, 1300, 1500, 1700, 1900 rpm. During experiment, the ambient pressure was average of 0.92 kPa, average air density was 1.1 kg/m^3 and air temperature was 32 ± 5 °C. The CR of the producer gas engine was set between of 9.7:1, 14:1, 17:1 and 19:1. Appropriate spark ignition timing was dependent on engine speed. In this work, optimum spark timing was chosen from the value that gave maximum brake torque (MBT), which was investigated and reported in our previous work [17]. From the previous findings, it was necessary to retard the spark ignition time, compared to a typical gasoline engine. In this work, the spark timings used were 35° BTDC for 1100 rpm, 40° BTDC for 1300 rpm, 40° BTDC for 1500 rpm, and 45° BTDC for 1700 rpm, respectively. Air and fuel were controlled and measured by means of flow meters and regulators. The measurement was conducted over an interval of 10 min, after achieving a stable operation. The air and fuel were finely tuned in such a way that maximum brake torque was achieved. Variation of air–fuel ratio was not carried out in this study. The mixture value was fluctuated narrowly around an equivalence ratio of unity under normal producer gas operation, similar to those reported by Sridhar et al. [15]. Measurement emission was carried out measured CO, HC and smoke density with choose at CR of 14:1 due to best engine performance. Data analysis for performance evaluation of the small producer gas and diesel engines was as follows:

Brake power:

$$P = 2\pi N\tau \tag{1}$$

where τ is the engine torque (Nm) and N is the engine speed of engine (s^{-1}).

Brake specific fuel consumption (*BSFC*):

$$BSFC = \frac{\dot{m_f}}{P} \tag{2}$$

where $\dot{m_f}$ is the mass flow rate of biomass in small producer gas engine (kg/h) and diesel (kg/h).

Brake thermal efficiency:

$$BTE = \frac{P}{\dot{V}_{pg} LHV_{pg}} \tag{3}$$

$$BTE = \frac{P}{\dot{m_f} LHV_{Di}} \tag{4}$$

Brake specific energy consumption (*BSEC*):

$$BSEC = BSFC \times LHV_{pg} \tag{5}$$

$$BSEC = BSFC \times LHV_{Di} \tag{6}$$

where $\dot{V}_{pg}$ is the producer gas flow rate (m^3/s), LHV_{pg} and LHV_{Di} are the lower heating values of producer gas (MJ/Nm^3) and diesel (kJ/kg), and $\dot{m_f}$ is the diesel fuel mass flow rate (kg/s).

3. Results and discussion

3.1. Engine torque and power

In general, the small producer gas engine was able to work continuously and operated smoothly on producer gas with appropriate tuning. Preliminary engine reliability test was conducted and evaluated in terms of variation in engine power output. The coefficient of variation (COV) is defined as the ratio between the standard deviation and the mean value. The COV of the engine power was found to vary narrowly between 1.75% and 3.0%, which were in similar magnitude or better than those reported in previous work [17].

Engine torque of the producer gas engine at varying CR compared to the original diesel engine is shown in Fig. 3. The engine torque of producer gas and diesel engines was found to increase as load and engine speed increased. Maximum engine torque, for producer gas engine, of 18.61 was achieved at 1500 rpm on full load, while the diesel engine obtained (20 Nm) at 1900 rpm on full load. For the producer gas engine, it was expected that higher CR engine would develop higher torque than lower CR. Reduced torque at low CR was anticipated because total work of all the engine cycles was less than the total work at higher CR. However, in this work, the engine torque was not found to vary markedly with change in CR. Use of high CR would result in higher flame speed as temperature of cylinder gases would be expected to increase, which in turn resulted in retarded MBT ignition timing. If the CR was too high, the engine would knock. In general, the CR of an SI engine without knock occurring was between of 6 and 10, while the CR of a gas engine can be as high as 17:1 before the onset of knock [15]. The engine torque of the diesel engine was always higher than that of the producer gas engine. Reduction of engine torque can be attributed to low energy density of producer gas which is a limitation of gaseous fuels, compared with liquid fuels [19]. Moreover, reduction of CR caused the engine torque to decrease. Finally, the volumetric efficiency of the engine was low; hence, engine torque was decreased. Gaseous fuel restricted air entering the combustion chamber. Comparing engine torque of the producer gas engine, maximum engine torque occurred at 14:1 of CR on maximum load. The engine torque of diesel engine was lower than that of the producer gas engine, between 1100 and 1500 rpm of engine speed. However, when engine speed was increased to more than 1600 rpm, the diesel engine showed higher torque than the producer gas engine. Increase in engine torque in the diesel engine was due to plentiful oxygen available, leading to more complete combustion.

Fig. 4 shows effect of loads and speeds on the brake power. Both engine brake powers were found to increase with engine load and speed. The brake power of the producer gas engine was always less than that of the diesel engine. Power de-rating was caused by low energy density of the combustible mixture as well as low volumetric efficiency. Increase in CR would expect to reduce the power de-rating. However, like the brake torque, the brake power of the producer gas engine was not found to vary with CR. Maximum brake power for the producer gas engine of 3.5 kW was obtained at 14:1 of CR and was unable to increase over 1700 rpm for all CRs considered. At 9.7:1 and 14:1 of CR, the engine showed deceleration due to low flammability and energy density of the producer gas, compared to gasoline or natural gas [20]. At CR of 17:1, the engine knock was occurred due possibly to the excessive CR. Using high CR in the engine caused an increase in gas density, temperature, ignition lag in combustion chamber leading to knocking [21].

3.2. Brake thermal efficiency

The brake thermal efficiency is shown in Fig. 5. For the producer gas engine with CR = 14:1, the BTE was always higher than that for CR of 9.7:1 and 17:1. Reduction of brake thermal efficiency was due to higher producer gas flow rates and poorer combustion. The brake thermal efficiency of the producer gas engine at each CR was lower than that of the diesel engine. Hence, the reduction in efficiency occurred due to characteristics of the fuel, lower compression ratio and volumetric efficiency [12]. The brake thermal efficiency tended to increase with engine load and speed. Better combustion was related to slightly rich gas-air mixture at higher loads and speed. The maximum brake thermal efficiencies of the producer gas and diesel engines of 23.5% and 26.9% were achieved at 1700 rpm. However, the brake thermal efficiency of the diesel engine can be increased further with an increase in engine speed

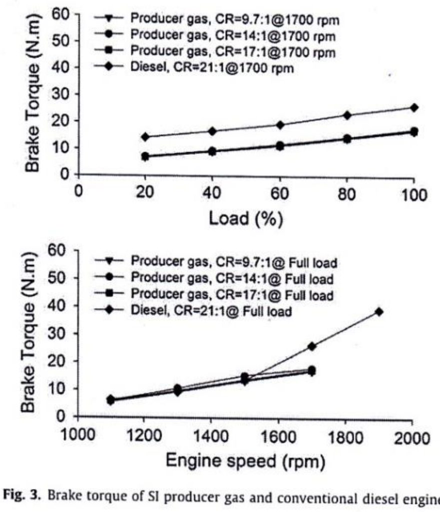

Fig. 3. Brake torque of SI producer gas and conventional diesel engine.

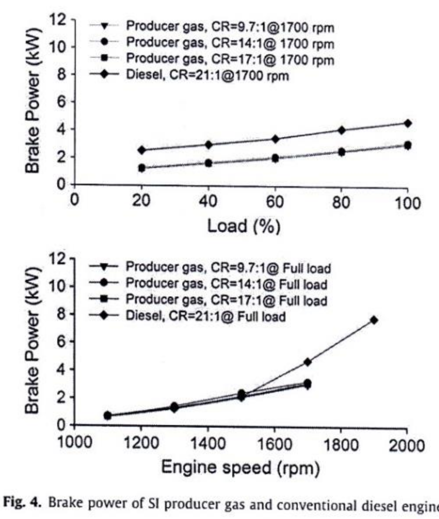

Fig. 4. Brake power of SI producer gas and conventional diesel engine.

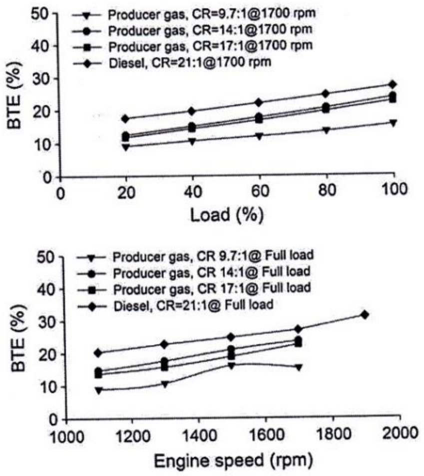

Fig. 5. Brake thermal efficiency of SI producer gas and conventional diesel engine.

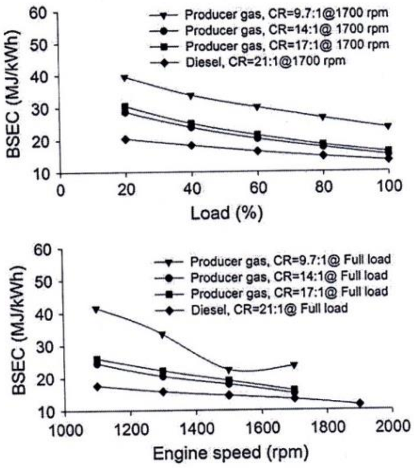

Fig. 7. Specific energy consumption of SI producer gas and conventional diesel engine.

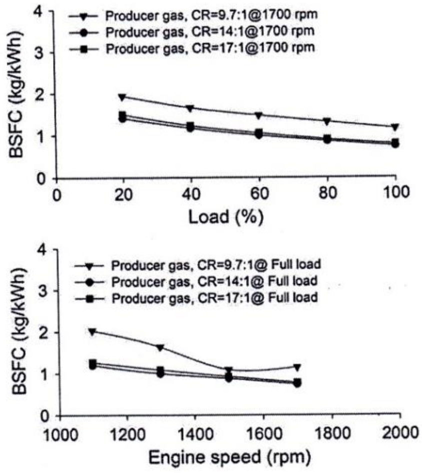

Fig. 6. Brake specific fuel consumption of producer gas engine.

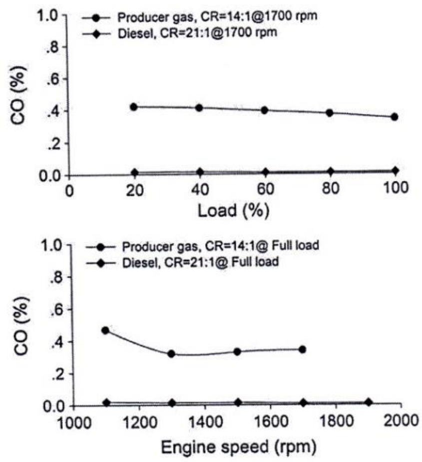

Fig. 8. Comparison CO emission of producer gas engine and diesel engine.

to about 30–35%. Comparing with previous works, it was found that the efficiency values from this work were in similar magnitude to those reported in [12,14]. The engine efficiency of about 21% was achieved at the CRs of 12:1 and 17:1. However, the value as high as 25–30% were reported for a producer gas engine [3].

3.3. Brake specific fuel and energy consumptions

Fuel consumption at CR of 14 was always lower than that found at CR 9.7:1 and 17:1, shown in Fig. 6. The specific fuel consumption was decreased with increasing engine load and speed. It was commonly known that low loads and speeds caused poor combustion in engine cylinder. However, as engine load and speed increased, the value tended to improve due to more complete combustion [21]. Minimum specific charcoal consumption rate of the producer gas engine of 0.74 kg/kWh was achieved. Generally, most works reported specific fuel consumption rate to be between 1.2 and 2.0 kg/kWh of wood as fuel. The overall efficiency was in a range of 11.5–21% while CR of that engine was relatively low [12].

Specific energy consumption of the producer gas engine and diesel engine may be calculated from the fuel consumption of charcoal and diesel with calorific value of both fuels. The specific energy consumption of both engines at variable loads and speeds are shown in Fig. 7. The specific energy consumption of the producer gas engine for each CR was always higher than the diesel engine at every loads and speeds. Diesel engines tended to decrease steadily with increasing load or speed. Comparing between different CRs of the producer gas engine, minimum specific energy consumption was obtained at CR of 14:1. Minimum

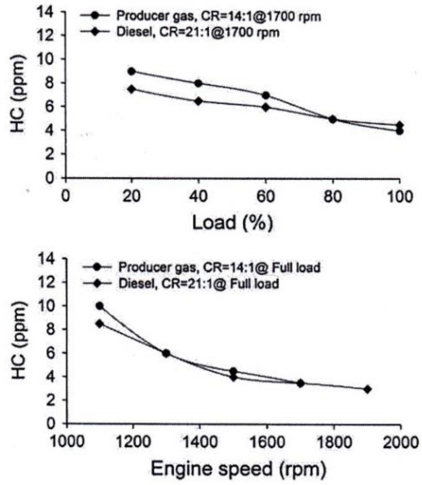

Fig. 9. Comparison HC emission of producer gas engine and diesel engine.

specific energy consumption of producer gas engine of 15.07 MJ/kWh was achieved. This was higher than diesel engine by 11.3% for operation at 1700 rpm and full load. The specific energy consumption of producer gas engine was increased due to factors like energy content of fuel mixture and volumetric efficiency [12].

3.4. Exhaust emissions

CO emission of the producer gas engine was found to be higher than the diesel engine for all loads and speeds, shown in Fig. 8. Higher CO in the exhaust was due to insufficient oxygen for combustion. CO emission from the producer gas engine was slightly reduced with increasing load and speed, while CO from diesel engine was stable when loads were in a range of 60–100%. Reduction of CO emission at high load was due to more complete combustion. Minimum CO emission of 0.34% and 0.01% were achieved at full load. However, in comparison with gasoline operation, CO emission of the producer gas engine was significantly less than that from gasoline operation in a range of 2–6% [22]. Hydrocarbon emissions of both engines were obtained in the range of 3.5–10 ppm and 3–8.5 ppm respectively, shown in Fig. 9. The average HC of the producer gas engine was marginally higher than diesel engine. The HC emissions of both engine was decreased with increased engine loads and speeds. This may be due to efficiency loss at low loads and incomplete combustion in the engine. However, comparison against gasoline engine, the HC emissions of the producer gas engine was lower, which HC emissions from gasoline engine of about 330 ppm was reported [22]. Smoke density of the producer gas engine was observed to be lower than the diesel engine, shown in Fig. 9. Smoke density of the producer gas engine started at 80–100% of load. The smoke density of the producer gas engine was achieved 0–2%, while 1.5–12% for diesel engine was recorded (see Fig. 10).

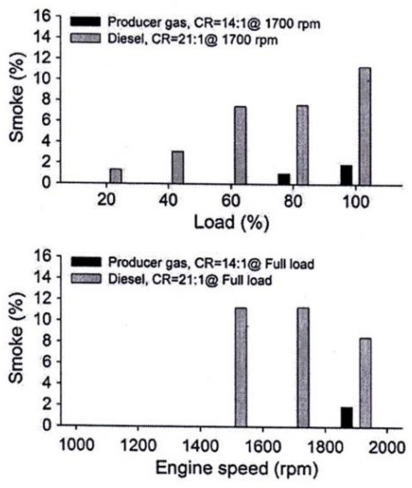

Fig. 10. Comparison smoke density of producer gas engine and diesel engine.

4. Conclusions

In this work, important findings on performance of a small diesel engine converted into a SI engine running on 100% producer gas with high CR were highlighted. The modified SI engine was able to operate with producer gas successfully. It was shown as high as CR = 17:1 may be operated for a small engine fueled with producer gas, without the risk from knock tendency. Reduction in torque and power de-rating were observed for the producer gas engine due mainly to low energy density of the air/fuel mixture and low volumetric efficiency of the engine. However, they were not varied significantly with CR considered. Increasing CR was shown to improve the brake thermal efficiency and the specific energy consumption slightly.

Acknowledgments

The authors would like to acknowledge supports from Chiang Mai University, the Energy Policy and Planning Office and the Commission on Higher Education. The Energy Research Centre of Maejo University was thanked for providing test facilities.

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Performance Investigation of a Small Engine Fueled with Producer Gas and Diesel in Dual Fuel Operation

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Abstract

Producer gas from biomass gasification can be used as a substitute fuel in diesel engines. In this work, performance of a small diesel engine operated on producer gas/ diesel dual fuel mode was investigated. Experimental tests were carried out on an 8.2 kW, single cylinder, naturally aspirated, diesel engine coupled to a 5.0 kW dynamometer. A downdraft gasifier was used to generate producer gas from charcoal as feedstock. Engine speed and load were varied between 1200 – 2000 rpm, and 1.0 – 3.5 kW, respectively. Engine torque, power, specific fuel consumption, diesel replacement rate, and thermal efficiency were evaluated. The dual fuel operation was compared against that with only diesel. It was found that the maximum diesel replacement rate of more than 75 % could be realized at 1400 rpm. Brake specific fuel consumption was in a range between 190 – 222 g/kWh. Efficiency of about 22 % was obtained, compared to 27 % from diesel operation.

Keywords: Biomass gasification, Compression ignition, Engine testing, Renewable energy, Small engines

1. Introduction

Escalating oil price and scarcity of fossil fuels coupled with exploding population have resulted in serious energy crisis. Sustainable technology that utilizes renewable energy sources should be developed to replace fossil fuels. Thailand is an advancing agro-industrial country. There are many biomass resources, especially agricultural residues such as wood chips, charcoal, rice husks, rice straws, corn cobs, sugar canes, etc available. But, at present, they are not largely utilized.

Biomass converted to producer gas via gasification is of great interest because the fuel gas can be used directly in engines. Gasification is an irreversible thermo-chemical process, by which feedstock is thermally decomposed. The end products are principally 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 3.0 – 8.0 MJ/Nm^3. The main advantages of gases as fuel over liquid or solid fuels are that (i) gases burn with higher efficiency than the solid or liquid fuels, (ii) they have a higher rate of heat release (iii) the rate of energy output is easily controlled and adjusted, and (iv) gaseous fuels with good energy utilization can be used for power sources.

Earlier studies reported that producer gas has been tried in two types of existing four stroke engines. Spark ignition, (SI) gasoline engines were operated directly as gas engines and compression ignition, diesel engines were operated on gas and diesel as dual-fuel engines. The first type was generally with lower compression ratio (CR), hence, low efficiency and power output. Munoz et al. [1] reported test results on an SI engine fueled with producer gas at a CR of 8.2:1. Power de-rating of 50% was observed, caused by unsuitability of a gas dosage equipment and low heating value of producer gas used. Sridhar et al. [2] used producer gas on an SI engine converted from diesel engine. Its CR was adjusted to 17:1. They found that increasing CR resulted in decreasing tendency of ignition timing. Maximum thermal efficiency was 21 %. Mustafi et al. [3] reported work using synthetic gas from aqua-fuel on an SI engine at CR between 8:1 and 11:1. They found that syngas affected de-rating of 23 %, compared to natural gas. Higher torque was obtained with increasing CR. Papagiannakis et al [4] reported work using producer gas on an SI engine at a CR of 11:1. They found that the engine ran well. The engine output was similar to natural gas engine. But, the specific fuel consumption was more than natural gas engine by 47 %. Dasappa et al [5] studied the use of producer gas on 100 kW, SI engine coupled to a generator at a CR of 9.7. They found that maximum thermal efficiency was 18 %.

As far as duel fuel operation was concerned, earlier studies on this topic was found to be favorable. Uma et al. [6] used producer gas in a diesel engine on dual fuel mode. They achieved the maximum diesel replacement in a range of 67-86 %. Low emissions of sulphur dioxide (SO_2), hydrocarbons and oxides of nitrogen were reported, compared to diesel mode. Singh et al. [7] tested performance of a diesel engine on dual fuel mode. The maximum diesel replacement of 63% was observed. Brake powers were found to decrease marginally. Ramadhasc et al. [8] presented results from a producer gas fed to a 5.5 kW diesel engine. Specific energy consumption reported for both wood chips and coir pith as fuels were 18 MJ/kWh, compared with about 15 MJ/kWh from diesel. They reported a maximum of 72 % diesel replacement at 50% load. Dasappa et al. [9] used producer gas and diesel on a 68 kW diesel engine, reporting an average diesel replacement of about 75 % with an overall efficiency of 22 %. Lekpradit et al. [10] investigated effect of advanced injection timing on dual fuel operation. They found that increasing advance of the injection timing led to lower diesel consumption, but increase in overall efficiency and diesel replacement. Dussadee et al. [11] reported test results on dual fuel in a 32 kW diesel engine. They achieved a maximum diesel replacement of 60 % with an overall efficiency of 20 %.

The objective of this study was to investigate performance of a small engine fueled with producer gas and diesel in dual fuel mode without modifying the engine. This is to reduce diesel fuel requirement.

2. Experimental

2.1 Apparatus

The engine setup is schematically shown in Fig. 1, consisting of a gasification system and a diesel engine adapted to operate in the diesel and dual fuel modes. The gasification system was configured to operate on different biomass materials as fuels. It consisted of a gas generator, a gas cooler and a gas filter. The other elements of the package were a water treatment plant for closed-loop water recirculation system. The specification of gasifier used is given in Table 1. The engine used in this work was a naturally aspirated, 8.2 kW, small diesel engine. It was a four-stroke, single cylinder, compression ignition engine with bore and stroke of 92 and 90 mm, respectively. Compression ratio used was 21:1. The engine was coupled to a 5 kW dynamometer. The Y-shaped carburetor was used in dual fuel operation with producer gas to enable mixing of gas with intake air. Specifications of the engine are given in Table 2.

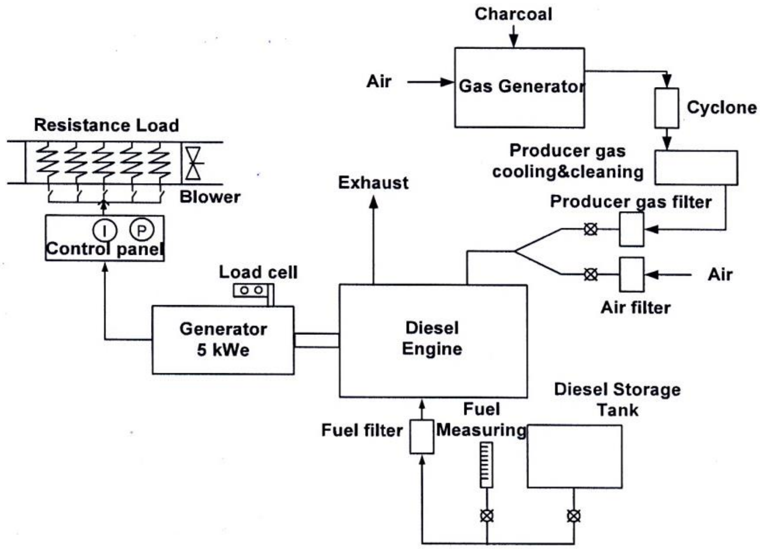

Fig. 1 Schematic diagram of producer gas engine test rig used in this study

Table. 1 Specification of the gasifier

Type of gasifier	Downdraft, batch feeding
Feeding	Manual
Fuel consumption	5 kg/h
Hopper capacity	30 kg
Gas cooling	Water
Biomass size	10 mm (minimum) 50 mm (maximum)

Table. 2 Specifications of the small diesel engine

Parameters	Specification
Type	indirectly injected, 4S, single cylinder, engine
Engine rating (kW)	8.2
Bore (mm)	92
Stroke (mm)	90
Displacement (l)	0.598
Compression ratio	21
Alternator rating (kW)	5
Rated output (kW)	5.0 @ 1500 rpm
Rated speed (rpm)	2400
Loading device	Electrical generator

2.2 Test procedures

The test conditions were at ambient pressure of 0.92 kPa; air density of 1.1 kg/m^3. Ambient temperature during the testing period was 32 ± 3 °C. A load bank was connected to test the engine generator set. Measurements on current, voltage, frequency and fuel consumption were carried out. The static pressures were monitored using water tube manometers. The feedstock used for gasification was charcoal with moisture content between 12 to 15 % dry basis. It was fed to the gasifier through the top opening. Air entered at the combustion zone and producer gas generated left near the bottom of gasifier at the temperature of about 500 - 600 °C. Hot producer gas was allowed to pass through the cooler where its temperature was reduced to ambient level. The cooled gas was then passed through the filter to remove tar and other particulate matter.

At the start, the engine was operated in diesel mode until stable, usually after 30 min. It was then switched to duel fuel mode where producer gas was fed and mixed with intake air. Amount of producer gas was adjusted by means of a control valve.

2.3 Data analysis

Agilant 6890 gas chromatography was used to measure mole fractions of CO, H_2, CH_4, CO_2 and N_2 in the producer gas. They were found to be CO at 18 ± 2%, H_2 at 14 ± 2%, CH_4 at 1 ± 0.5%, CO_2 at 12 ± 2%, and balancing N_2. Tests were carried out at varying engine speeds and loads between 1200 – 2000 rpm, and 1.0 – 3.5 kW, respectively. The producer gas and airflow rates were measured using gas meters. The engine torque, brake power, specific energy consumption, diesel consumption, diesel replacement rate, and thermal efficiency were evaluated. The dual fuel operation was then compared with only diesel.

3. Results and Discussion

3.1 Dual fuel operation

The gasifier was able to work continuously. The small engine was operated smoothly on dual fuel mode. The temperature of input producer gas was in a range of 35 - 40 °C. The flow rate of producer gas was in a range of 5 to 40 m^3/h.

3.2 Engine torque

Fig. 2 shows engine torques of dual fuel mode of operation, compared with diesel fuel operation at various engine outputs. The engine torques of dual fuel mode was found to be slightly lower than diesel mode, by about 2 % on average. Reduction of engine torque was observed due to lower volumetric efficiency during intake, hence insufficient air to complete combustion. Generally, the volumetric efficiency of diesel fuel mode was about 85 – 90 %, but the dual fuel mode had actual volumetric efficiency of lower than 70 %.

3.3 Brake power

Fig. 3 shows engine brake powers of dual fuel mode of operation, compared with diesel fuel operation at various engine speeds. The engine brake powers of dual fuel mode were observed to be similar to diesel mode, in a speed range of 1200–1600 rpm. Between 1800–2000 rpm, dual fuel operation showed lower brake powers than diesel mode. Decrease of brake power at high engine speeds may be due to insufficient oxygen available to complete the combustion [11]. The brake powers of dual fuel mode and diesel mode were between 0.68 – 6.33 kW, and 0.69 – 6.37 kW, respectively.

3.4 Specific energy consumption

Fig. 4 shows variation of specific energy consumption with engine speeds. The specific energy consumption from dual fuel mode of operation was found to be higher than that from diesel mode at all engine speeds. At higher producer gas flow, specific energy consumption was higher. Patterns of specific energy consumption for both modes in a range of 1200 - 1400 rpm were rather constant, but increased between 1600-2000 rpm. At 1500 rpm, specific energy consumption was at minimum. The specific energy consumption in dual fuel and diesel modes at 1500 rpm were 17.7 and 18.7 MJ/kWh, respectively.

3.5 Diesel consumption

Diesel consumption at various engine speeds is shown in Fig. 5. The diesel consumption in dual fuel mode was observed to be lower than diesel mode for all engine speeds. Minimum diesel consumption in dual fuel mode was about 100 g/kWh at 1500 rpm, while for diesel mode operation, it was 360 g/kWh at engine speed of 1800 rpm.

3.6 Diesel replacement rate

Diesel replacement rate under various engine speeds was calculated from diesel consumption in diesel mode and dual fuel mode. The results are shown in Fig. 6. Use of producer gas in dual fuel mode of operation was found to reduce the consumption of diesel at all engine speeds, as expected. The maximum diesel replacement rate was 75 % at engine speed of 1400 rpm. The diesel replacement rate was found to decrease with increasing engine speed. The lowest replacement rate was 58 % at engine speed of 2000 rpm.

3.7 Thermal efficiency

Thermal efficiencies of both diesel and dual fuel mode of operation are shown in Fig. 7. Thermal efficiency of dual fueled engine was found to be lower than those of diesel engine for all engine speeds. Reduction in thermal efficiency was due to higher producer gas flow rates and lower calorific value of producer gas. Higher percentage of producer gas in the gas–air mixture may reduce the amount of fresh air entering the engine combustion chamber. Maximum thermal efficiencies of dual fueled and diesel engine were calculated to be 22 and 27 %, respectively. Both were achieved at engine speed of 1600 rpm.

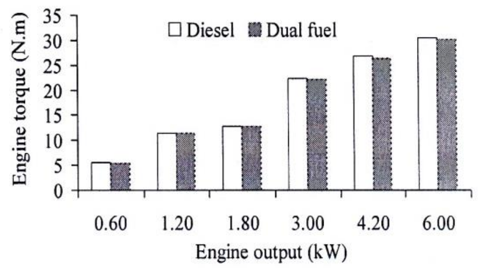

Fig. 2 Comparison of engine torque

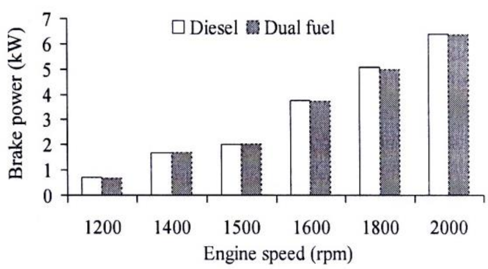

Fig. 3 Comparison of engine brake powers

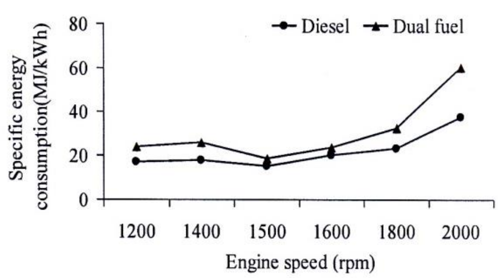

Fig. 4 Comparison of specific energy consumption

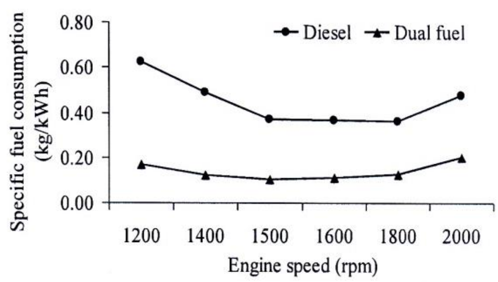

Fig. 5 Comparison of diesel consumption

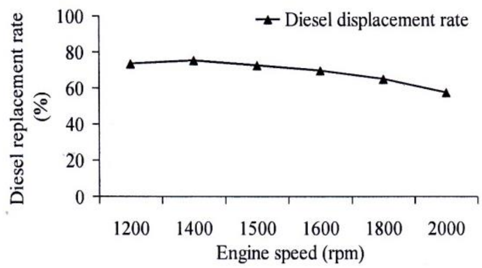

Fig. 6 Diesel replacement rate in dual fuel mode

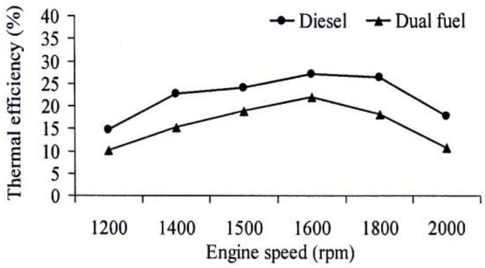

Fig. 7 Comparison of engine thermal efficiency

4. Conclusions

It was shown that unmodified diesel engine was capable of successful running in dual fuel mode of operation with biomass derived producer gas. Important findings on the performance of a small diesel engine in dual fuel mode of operation using producer gas were highlighted in the present paper.

The engine torque and brake power in dual fuel mode operation were slightly lower than those in diesel mode at all engine speeds.

The specific energy consumption in dual fuel mode of operation was higher than that of diesel mode at all engine speeds. But, the diesel consumption in dual fuel mode was much lower than diesel mode at all engine speeds. Maximum diesel replacement rate and thermal efficiency of dual fuel operation were 75 and 22 %, respectively.

5. Acknowledgment

The authors would like to thank the Graduate School, Department of Mechanical Engineering, Chiang Mai University, and the Energy Research Centre, Maejo University for providing test facilities. The Office of the Commission on Higher Education was acknowledged for financial support to NH via "Faculty development scholarship".

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Case Studies in Thermal Engineering

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Prediction of small spark ignited engine performance using producer gas as fuel

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ARTICLE INFO

Article history:
Received 27 October 2014
Received in revised form
19 January 2015
Accepted 15 February 2015
Available online 16 February 2015

Keywords:
Mathematical modeling
Engine simulation
Small engine
Producer gas
Engine performance

ABSTRACT

Producer gas from biomass gasification is expected to contribute to greater energy mix in the future. Therefore, effect of producer gas on engine performance is of great interest. Evaluation of engine performances can be hard and costly. Ideally, they may be predicted mathematically. This work was to apply mathematical models in evaluating performance of a small producer gas engine. The engine was a spark ignition, single cylinder unit with a CR of 14:1. Simulation was carried out on full load and varying engine speeds. From simulated results, it was found that the simple mathematical model can predict the performance of the gas engine and gave good agreement with experimental results. The differences were within $\pm 7\%$.

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1. Introduction

Producer gas was derived from biomass via gasification with average calorific value of about 5 MJ/Nm^3 [1]. Presently, the use of 100% producer gas in spark ignition (SI) engine was not successful, because producer gas has low energy density, hence, low power output and efficiency [2]. Recently, increasing performance of producer gas engine can be done by increasing compression ratio (CR), changing combustion chamber, mounting gas carburetor and modifying the ignition system [3,4]. Experimental evaluation of a producer gas engine can be costly, complicated and time consuming. Ideally, the engine performance may be predicted using mathematical equations [5]. Establishing mathematical models is of interest. In this work, a single zone cylinder model was used. It can provide quick calculation of optimum conditions. Examination of various engine performance parameters may be achieved [6,7]. The basic assumption of the single zone cylinder model was based on mass balance analysis, regardless of chemical reaction, homogeneous charges, and mixing of gases inside the cylinder [8]. Therefore, the objective of this work was to study the use of mathematical model in small producer gas engine comparing with experimental in term torque, brake power, thermal efficiency and specific fuel consumption.

2. Mathematical modeling

The model was combined with physical based equations for describing phenomena and performance of the small producer gas engine. The details of the mathematical models are as follows:

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http://dx.doi.org/10.1016/j.csite.2015.02.003
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2.1. Cylinder pressure

The pressure in cylinder of SI engine can be derived from the first law analysis. The cylinder pressure versus crank angle is shown in Eq. (1) [9].

$$\frac{dP}{d\theta} = \frac{k-1}{V}\frac{dQ}{d\theta} - k\frac{P}{V}\frac{dV}{d\theta} \tag{1}$$

where, P is the pressure inside cylinder, θ is crank angle, k is specific heat ratio, Q is heat releases, V is the cylinder volume and as a function of crank angle, given as

$$V(\theta) = \frac{V_d}{r_c - 1} + \frac{V_d}{2}\left[\frac{l}{a} + 1 - \cos\theta - \left(\left(\frac{l}{a}\right)^2 - \sin^2\theta\right)^{0.5}\right] \tag{2}$$

where, V_d is displacement volume, r_c is compression ratio, l is connecting rod length, a is crank radius.

2.2. Heat input

The total amount of heat input to cylinder versus changes in the crank angle is shown in Eq. 3 [10].

$$\frac{\partial Q}{d\theta} = HV\int_{IVO}^{IVC} m^{\bullet}\, d\theta\frac{df}{d\theta} \tag{3}$$

where, HV is heating value, $m^{\bullet}$ is producer gas flow rate, IVO and IVC are inlet valve open and close positions before and after TDC, $f(\theta)$ is the Wiebe function. Producer gas flow rate through an intake valve was derived empirically from the engine test run between 1100–1900 rpm of engine speed. It is given as

$$m^{\bullet} = 0.00378V_d(0.105N^2 - 0.7922N - 0.0015N^3) \tag{4}$$

where, N is engine speeds and the Wiebe function is used to determine the combustion rate of the fuel, expressed as [11]:

$$f(\theta) = 1 - \exp\left[-5\left(\frac{\theta-\theta_0}{\Delta\theta}\right)^3\right] \tag{5}$$

where, θ is crank angle, θ_0 is start of heat release angle, $\Delta\theta$ is duration of heat release and can be determined from this equation.

$$\Delta\theta = -1.618\left(\frac{N}{1000}\right)^2 + 19.866\left(\frac{N}{1000}\right) + 39.395 \tag{6}$$

2.3. Heat transfer

The heat transfer is necessary for the internal combustion engine to maintain cylinder walls, pistons and piston rings. Normally, the heat transfer in the combustion engine includes conduction, convection and radiation [12]. However, for an SI engine, the primary heat transfer mechanism from the cylinder gases to the wall is convection, with only 5% from radiation [13]. The heat loss to the wall can be determined from the Newtonian convection equation [14] which is given as

$$Q_{loss} = hA(T_g - T_W) \tag{7}$$

where, h is heat transfer coefficient, A is surface area of combustion chamber T_g is gas temperature in cylinder, T_W is cylinder wall temperature. The heat transfer coefficient is instantaneous area average heat transfer coefficient derived from Woschni [15], shown in Eq. (8).

$$h = 0.82b^{-0.2}(P10^{-3}c)0.8T_g^{-0.53} \tag{8}$$

where, b is bore cylinder, c is equal to 6.18. The gas temperature is calculated using following equation from Sitthiracha [16] while, engine speed is in a range of 1000–6000 rpm.

$$T_g = 3.395\left(\frac{N}{1000}\right)^3 - 51.9\left(\frac{N}{1000}\right)^2 + 279.49\left(\frac{N}{1000}\right) + 676.21 \tag{9}$$

Calculation of surface area in cylinder is from the following equation [9] which includes cylinder head, cylinder bore and piston crown. Surface area at any crank angle is given as:

$$A(\theta) = \frac{\pi}{2}b^2 + \pi b\frac{s}{2}\left[\frac{l}{a} + 1 - \cos\theta - \left(\left(\frac{l}{a}\right)^2 - \sin^2\theta\right)^{0.5}\right] \tag{10}$$

2.4. Indicated and brake mean effective pressure

The sums of pressure in cylinder are indicated mean effective pressure (*imep*). The equation is given as [10]:

$$imep = \frac{\oint P\,dV}{V\,d} \tag{11}$$

Therefore, brake mean effective pressure (*bmep*) can be calculated from

$$bmep = imep - \sum fmep \tag{12}$$

2.5. Friction

The friction loss in an internal combustion engine can be analyzed by three components, including the mechanic friction, the pumping work and accessory work. Calculation of engine friction uses an empirical equation [17]. Major frictions include bearing friction, piston and ring friction, wall tension ring friction, valve gear friction, pumping loss, combustion chamber and wall pumping loss. The equations of friction loss are shown in Eqs. (13–18).

$$\text{Bearing friction} \quad fmep_1 = 0.0564\left(\frac{b}{s}\right)\left(\frac{N}{1000}\right) \tag{13}$$

$$\text{Piston and ring friction} \quad fmep_2 = 12.85\left(\frac{P_s}{bs}\right)\left(\frac{100S_l}{1000}\right) \tag{14}$$

$$\text{Wall tension ring friction} \quad fmep_3 = 10\left(\frac{0.377sn_p}{b^2}\right) \tag{15}$$

$$\text{Valve gear friction} \quad fmep_4 = 0.226\left(30 - \frac{4N}{1000}\right)\left(\frac{GD_{iv}}{b^2 s}\right) \tag{16}$$

$$\text{Pumping loss} \quad fmep_5 = 0.0275\left(\frac{N}{1000}\right)^{1.5} \tag{17}$$

Combustion chamber and wall pumping loss

$$fmep_6 = 0.0915\sqrt{\frac{imep}{11.45}}\left(\frac{N}{1000}\right)^{1.7} \tag{18}$$

where, P_s is piston skirt length, S_l is mean piston speed, n_p is number of piston ring, G is number of intake valve per cylinder, D_{iv} is Intake valve diameter, P_{mi} is the sum of pressure in cylinder.

2.6. Torque and brake power

The brake power and torque can be determined by following equations:

$$P_b = 0.5bmepNV_d \tag{19}$$

$$T_b = \frac{P_b}{2\pi N} \tag{20}$$

2.7. Brake thermal efficiency and brake specific fuel consumption

The brake thermal efficiency and brake specific fuel consumption (BSFC) when biomass is used as fuel can be modified from gasoline and diesel engine Eqs. [18,19], as

$$\eta_{th} = \frac{P_b}{m^{\bullet} HV} \tag{21}$$

$$BSFC = \frac{m_b^{\bullet}}{P} \tag{22}$$

where, $m_b^{\bullet}$ is biomass (charcoal) consumption

2.8. Initial temperature and pressure of compression process

From the Otto cycle, the first process is isentropic compression. Calculation of initial temperature and pressure can be as follows [17]:

$$\frac{T_2}{T_1} = r_c^{k-1} \tag{23}$$

$$\frac{P_2}{P_1} = r_c^{k} \tag{24}$$

where, T_1 and P_1 are ambient temperature and pressure while T_2, P_2 are cylinder temperature and pressure in the compression process.

3. Experimental setup and measurements

Model validation was carried out through experimentation. A small SI engine converted from a CI engine was used to operate 100% on producer gas. The engine was of single cylinder, four strokes, 598 cc and bathtub combustion chamber [4]. The detailed specifications of small producer gas engine are shown in Table 1. The power output was measured by a dynamometer set and monitored by a display panel. The best experimental conditions were used to develop mathematical models. They were on full load and 14: 1 of CR, the engine speed between 1000–2000 rpm. Producer gas was derived from charcoal. The composition of the gas was of CO 30.5 ± 2%, H_2 8.5 ± 2%, CH_4, 0.35%, CO_2 4.8 ± 1%, and O_2, 6.3 ± 0:5%, and the balance Nitrogen. The mean calorific value of the producer gas was 4.64 MJ/Nm^3. Parametric study was based on numerical solution to find performance of the engine.

4. Results and discussions

In this study, the 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 engine experiment. They are shown in Figs. 1–3. At low engine speeds, the predicted values were almost equal to the experimental results. At high speeds, there were small differences at engine speeds between 1500–1900 rpm.

This may be attributed to difference in producer gas flow rate entering the cylinder. The producer gas flow rate was derived empirically from the fuel consumption and volumetric efficiency. However, the deviations were likely due to other factors such as pressure and temperature in cylinder in combustion process. The use of a simple model did not consider

Table 1
Engine and operational specifications in simulation.

Engine type	SI engine, 4 stroke, single cylinder
Fuel	Producer gas
Compression ratio	14:1
Spark ignition timing	30° BTDC
Bore × Stroke (mm)	92 × 90
Connecting rod length (m)	0.143
Crank radius (m)	0.0413
Clearance volume (m^3)	4.60×10^{-5}
Swept volume (m^3)	5.98×10^{-3}
Rated output (kW)	3.2 @ 1700 rpm
Ambient pressure (kPa)	0.92 kPa
Ambien temperature (K)	308
Mean wall temperature (K)	400
Air density (kg/m^3)	1.2
Air/fuel ratio	1.2:1
Equivalent air/fuel ratio	1
Duration of combustion	90°

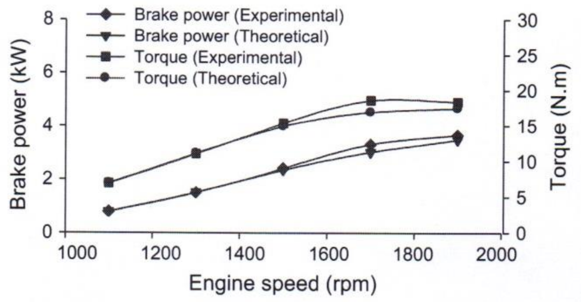

Fig. 1. Comparison between theoretical and experimental brake power and torque.

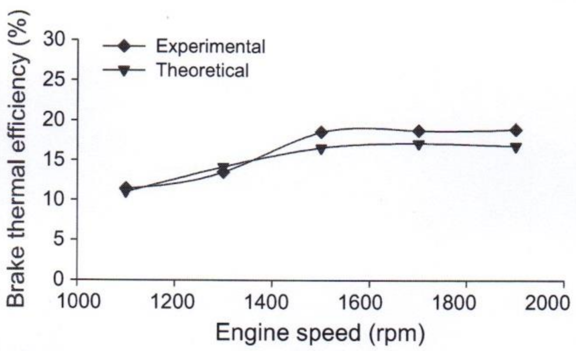

Fig. 2. Comparison between theoretical and experimental brake thermal efficiency.

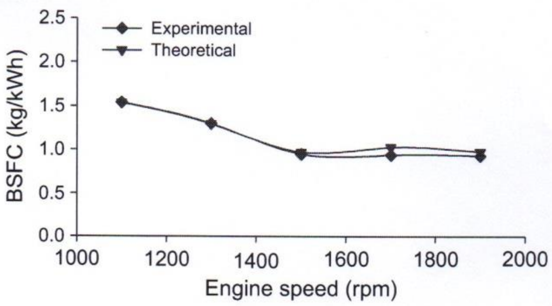

Fig. 3. Comparison between theoretical and experimental brake specific fuel consumption.

micro-analysis of the engine [10]. The average errors of brake power, torque, thermal efficiency and BSFC were −3.30, −3.32, −6.50 and 3.07%, respectively. Therefore, it is concluded that the developed mathematical model gave good agreement and can be applied to the small producer gas engine under the similar conditions.

Table 2
Mean percentage error of thermodynamics model with SI engine.

Engine performance	Mean percentage error (%)			
	This work	[10]	[13]	[20]
Brake power (BP)	−3.30	7.63	−2.74	23.08
Torque	−3.32	–	−3.14	–
Brake thermal efficiency (BTE)	−6.50	0.06	–	21.83
Brake specific fuel consumption (BSFC)	3.07	−0.12	–	–

For comparison, the use of the thermodynamics model to an IC engines is summarized in Table 2. The model validations of 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%. They appeared to be acceptable, compared to the experimental results. The mathematical modeling of this work may be used to predict performance of an SI engine operated on producer gas engine well.

5. Conclusions

The model adopted for this work was found to be acceptable and may be used to predict the performance of producer gas engines. The average percentage errors of brake power, torque, brake thermal efficiency and BSFC were within 6.50%.

Acknowledgment

Supports from Chiang Mai University, the Energy Policy and Planning Office, and the Commission on Higher Education were highly appreciated. The authors would like to thank the Graduate School of Chiang Mai University, as well as the Energy Research Centre of Maejo University for providing test facilities and technical supports.

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Publications

Homdoung , N., Tippayawong , N., Dussadee , N., "Effect of ignition timing advance on performance of a small producer gas engine", International Journal of Applied Engineering Research, Vol. 9, 2014, 2341-2348.

Homdoung , N., Tippayawong , N., Dussadee , N., "Performance investigation of a modified small engine fueled with producer gas", Maejo International Journal of Science and Technology, Vol. 9, 2015, 10-20.

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Homdoung , N., Tippayawong , N., Dussadee , N., "Prediction of small spark ignited engine performance using producer gas as fuel", Case study in thermal Engineering, Vol. 5, 2015, 98-103.

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