PRODUCER GAS AS FUEL FOR THE DIESEL ENGINE

By

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These studies show that coconut shell charcoal was a suitable fuel for a small downraft gas producer to supply gaseous fuel to a 5-brake horsepower diesel engine. At the rated capacity, the engine ran on 90 percent of the energy from producer gas with a net calorific value of 130 Btu/n ft^3, and 10 percent from the liquid fuel, which was either diesel oil or crude coconut oil. The engine ran on 100 percent producer gas when conditions favored compression ignition of the air gas mixture. Such conditions included a lower load than the rated capacity and a richer gas heating value.

Introduction

The work described in this report is the result of a project supported by the Philippine Coconut Authority (PCA) entitled "Studies on the Performance of the Diesel Engine Utilizing Producer Gas from Coconut Shell Charcoal". The details of the project were contained in a letter of agreement between PCS and the U.P. Industrial Research Center (UPIRC) dated 21 October 1976.

The objectives of the project were (1) to design and fabricate an appropriate gas producer for supplying gaseous fuel to a single cylinder 5-brakehorsepower diesel engine; (2) to study the performance of the diesel engine when using producer gas as the main fuel and diesel oil or crude coconut oil as igniter or pilot fuel.

The experimental work was done over a period of six months, with Dr. Ibarra E. Cruz of the U.P. College of Engineering as Project Director and Engineer Deogracias Domingo of PCA as project coordinator.

Experimental Work

Fig. 1 shows the experimental lay-out for studying the performance of a single-cylinder diesel engine when using producer gas as its main fuel. The engine had a bore of 4.5 inches and a stroke of 4.25

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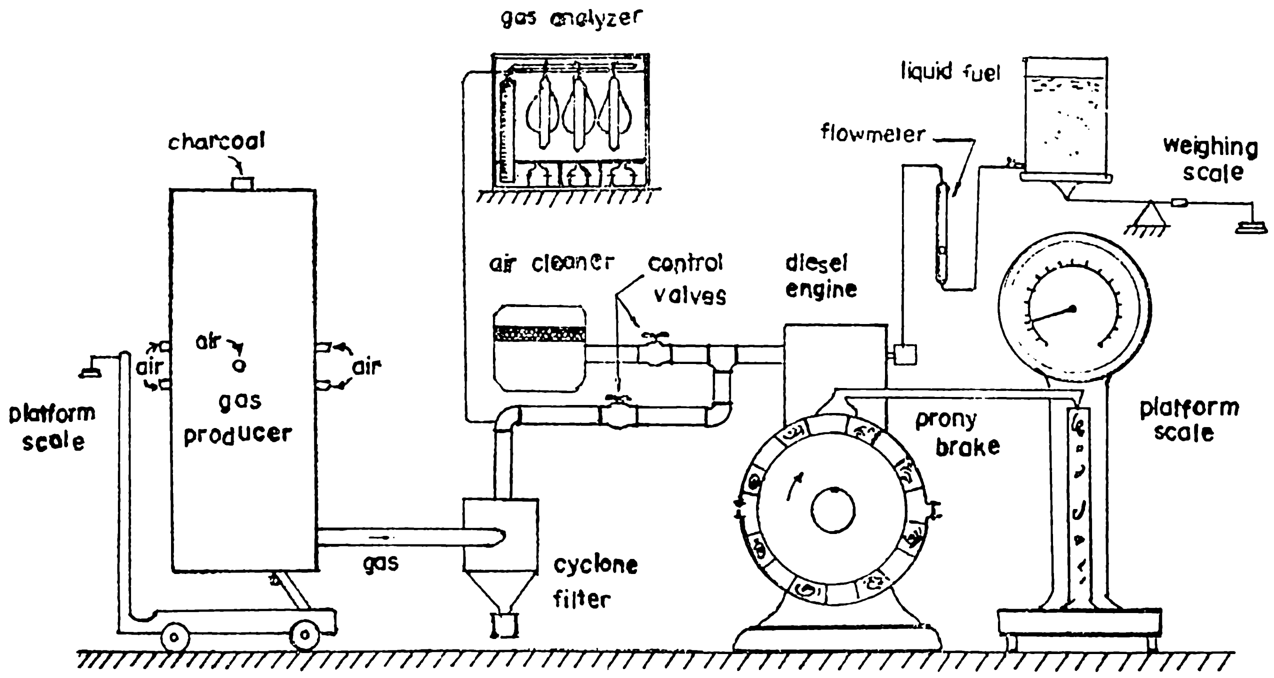


Fig.1 EXPERIMENTAL LAY-OUT FOR STUDYING DIESEL ENGINE PERFORMANCE WHEN USING PRODUCER GAS

inches. The compression ratio was approximately 14. The engine was manufactured by lister.

In Fig. 1, it is seen that the producer gas is aspirated into the engine together with the air. Therefore, the only modification necessary to allow the diesel engine to use producer gas is a gas pipeline connection to the air intake pipe to the engine with appropriate control valves for proportioning the air-gas mixtures.

The gas producer was mounted on a platform scale to allow weight measurements of the charcoal consumed during a test run. From derived relations, the weight rate of producer gas utilized in the engine was calculated. The liquid fuel tank, containing either diesel or crude coconut oil, was likewise mounted on a weighing scale. Hence, the rate of liquid fuel intake of the engine was known in each run. A rotameter or flowmeter was also installed in the liquid fuel line to serve as a check on the rate of fuel consumption.

The engine torque output was measured by a prony brake mounted on another platform scale, and the engine RPM by a hand tachometer. The brake horsepower output in each run was thus determined. The engine was started in the normal way by hand cranking, with the air intake value fully open and the producer gas valve fully closed. Thus the engine was run on diesel fuel (or crude coconut oil) alone at the start.

The gas producer was a suction--type downraft reactor with 12 air holes around the mid-section of the cylindrical body, and a single gas outlet at the bottom (see Fig. 2). Connected to the gas outlet was a cyclone separator to filter out from the gas entrained dust and fines before it went to the engine.

At the start-up, the gas producer was initially filled with charcoal crushed to about 1-inch size, up to the level of the air-holes. Feeding of fuel was done by opening the top of the reactor. A burning zone was started on the top of the charcoal bed by igniting small pieces of wood and when the charcoal was burning evenly at all levels of the air-holes (this took about 10 minutes to occur from the time the fire was lighted), the producer was charged with more charcoal until it was full. The top of the reactor was then closed and the gas intake valve to the engine slowly opened. The engine would now suck gas from the reactor and speed up. Since the engine was controlled by a speed governor adjusted to about 1000 RPM, the liquid fuel intake would be automatically reduced as more producer gas was sucked into the engine, until a minimum use of liquid fuel, that which was required only for ignition, was reached.

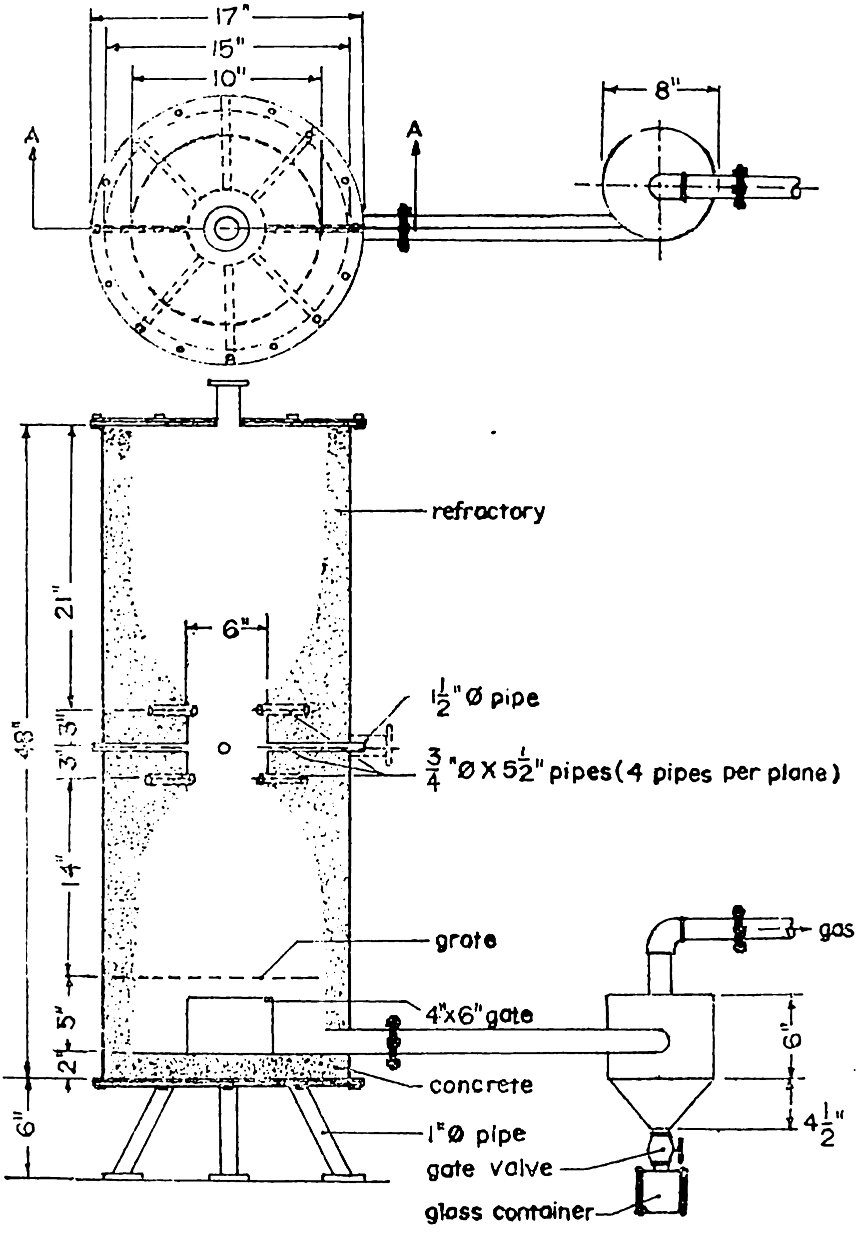


Fig. 2 GAS PRODUCER DESIGN

The initial design of the gas producer was such that it could be converted readily to operate either as a downdraft or as an updraft reactor. Also, the air intake could either be by suction from the engine or by forced draft from a compressed air tank. There was only a single air inlet at the side of the producer when it was operated as a down-draft reactor, (see dotted lines for the 1.5-in. air inlet in Fig. 2), so that the combustion zone was concentrated in the vicinity of this single air inlet. When operated as an updraft reactor, the air entered from below the reactor and passed up uniformly through a bar grate. The quality of the gas thus produced was better in the updraft producer. However, downdraft operation produced a cleaner gas, particularly when fuel with high voltile matter was used. Therefore after about half of the experimental runs were finished, the gas producer was redesigned to operate permanently as a suction, downdraft reactor.

Redesigning the reactor involved providing for additional air inlets so that air distribution to the combustion zone could be more uniform. Also to increase the depth of the combustion zone, the air holes were distributed around three circumferential planes (4 holes to a plane) spaced 3 inches apart thus extending the burning zone to a depth of at least 6 inches. Furthermore, the cross-sectional area of this combustion zone was reduced to 6 inches diameter (28 sq. in.) from the original 10 inches (79 sq. in.) and the longitudinal cross section of the reactor now exhibited a constriction or a "throat" at the combustion zone as seen in Fig. 2. The purpose of this throat was to make combustion more intense at this zone, i.e. combustion rate per unit of cross-sectional area would now be higher and hence temperatures and a deeper combustion zone would lend to more efficient cracking of the volatile and tarry material in the fuel and to the production of more combustible gases.

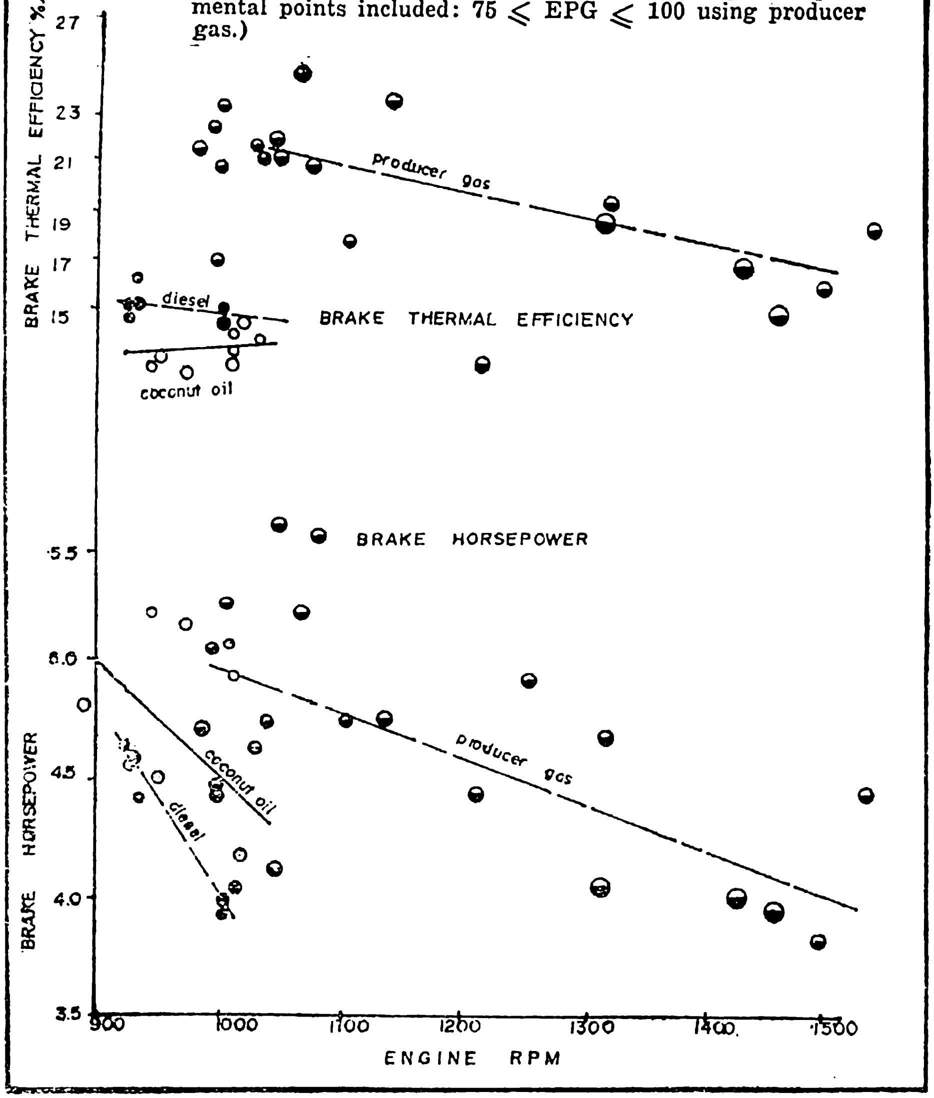
As discussed in the following section, results of subsequent test runs showed that the quality of the gas produced in downdraft operation, after the re-design of the producer, was improved significantly.

Experimental Results

An experimental run consisted for funning the engine for 15 minutes at approximately constant conditions of power output and engine speed. If the run was longer (in multiples of 15 minutes) with engine speed and power output approximately constant, then the run was correspondingly weighted. For instance, a run for 60 minutes was considered as comprising 4 runs each of 15 minutes duration.

Table 1 shows the results of 144 experimental runs. Sample computations for BHP output, brake thermal efficiency of the engine,

FIG. 3. Comparison of Diesel Engine Performance When Using Producer Gas ⊖, Coconut Oil ○, Diesel ●, range of experimental points included: 75 ⩽ EPG ⩽ 100 using producer gas.)



percentage energy from producer gas (EPG) utilized in the engine, and cold gas thermal efficiency of the gas producer are given in Annex 1. The analysis of produced gas were obtained by means of the Burrel gas analyzer.

In Runs 2 to 71, the gas producer was operated as a downdraft reactor with a single 1.5-inch air inlet, and a uniform inside diameter of 10 inches. In Runs 73 to 142 the re-designed producer was used in which 12 air inlet holes of 0.5 inch diameter were distributed around three circumferential planes of the reactor, and a contriction or "throat" at the combustion zone was made.

The improvement in the gas quality due to changes in the design was found to be statistically significant. Before the re-designing, the mean net calorific value ($C\ V_n$) of the producer gas in 57 runs was 116 Btu/n ft^3. After re-designing, the mean $C\ V_n$ in 65 runs was 11 t Btu/n ft^3. A t-test of the means resulted in a t value of 2.09. For 120 degrees of freedom, in order for the hypothesis to be true that the means are equal, the t value in 99 out of 100 cases must fall within $\pm$ 2.62. Since 2.89 is outside + 2.62, the hypothesis that the means are equal is rejected.

The plot of experimental points (for runs when EPG was equal or greater that 75 percent) on coordinates of brake horsepower against engine RPM and brake thermal efficiency against RPM, is shown in Fig. 3. The varying size of the circles representing experimental points correspond to varying numbers of runs coinciding at each single point.

It is interesting to note that the engine was run on liquid fuel (diesel oil or crude coconut oil) alone, the experimental points indicate that the maximum engine RPM attained was only slightly over 1000 RPM. This was due to the fact that the engine speed governor controlled fuel injection to maintain engine speed at approximately 1000 RPM. However, when the engine was running on producer gas alone (100 percent EPG), the maximum speed attained was greater than 1500 RPM an increase of more than 50 percent. This was understandable by the governor any longer (no liquid fuel was being used) but by the amount of producer gas allowed into the engine by the gas control valve.

That the engine could run on 100 percent producer gas without the need of even a small amount of liquid fuel for ignition purposes, was one of the pleasant surprises of these experiments. It was discovered that the right combination of air-producer gas mixture, gas calorific value, and engine load could lead to a condition whereby the combustible charge of air and gas could be ignited by piston compression alone. This condition usually occurred about an hour after start-up of the gas producer, when the gas quality had improved compared to that at start-up. A precaution that had to be observed, however, was to adjust the air-gas mixture to make it less rich as the gas calorific value continued to improve. This was to prevent severe detonation as pre-ignition was enhanced more and more.

An examination of Fig. 3 shows that the brake horsepower and the brake thermal efficiency decrease with increasing engine speed. This is consistent with the fact friction losses increase with engine RPM. In spite of the higher engine speeds and higher friction losses of the engine when producer gas was used, the mean thermal efficiency (18.4 percent) was significantly higher than the mean thermal efficiency when the engine was run on diesel oil alone (15.2 percent) or when run on crude coconut oil alone (13.4 percent thermal efficiency). The results of a statistical t-test comparing the means of the thermal efficiencies are shown in Annex 2. It was also observed that the engine emitted a clear smokeless exhaust when on producer gas even at full load. When on diesel alone, the engine exhaust was smoky, particularly at higher loads.

Discussion of Results

The scatter of points in Fig. 3 is quite noticeable and one factor accounting for this is the variable quality of the producer gas from run to run. It was observed that not only engine speed, but also the quality of the gas produced had an effect on both the load that the engine could carry and the brake thermal efficiency. The experimental points plotted in Fig. 1 (for runs when producer gas was used) were therefore correlated by linear regression using the least squares method. The results are equations for brake horsepower (BHP) and brake thermal efficiencies (e_b, %) in terms of both engine speed and net calorific value of the producer gas:

$$\mathrm{BHP} = 7.25 - 1.90 \times 10^{-3}\ (\mathrm{RPM}) - 3.22 \times 10^{-3}\ (\mathrm{CV_n}) \qquad (1)$$

Coefficient of correlation $R^2 = 0.48$

$$e_b = 25.63 - 1.34 \times 10^{-2}\ (\mathrm{RPM}) + 8.60 \times 10^{-2}\ (\mathrm{CV_n}) \qquad (2)$$

Coefficient of correlation $R^2 = 0.41$

Equations (1) and (2) were plotted for different values of RPM and CV_n in Fig. 4. It is evident in Fig. 4 that:

(1) Both engine load and brake thermal efficiency decrease with engine speed.

(2) Brake thermal efficiency increases with gas calorific value.

(3) Brake horsepower decreases with gas calorific value.

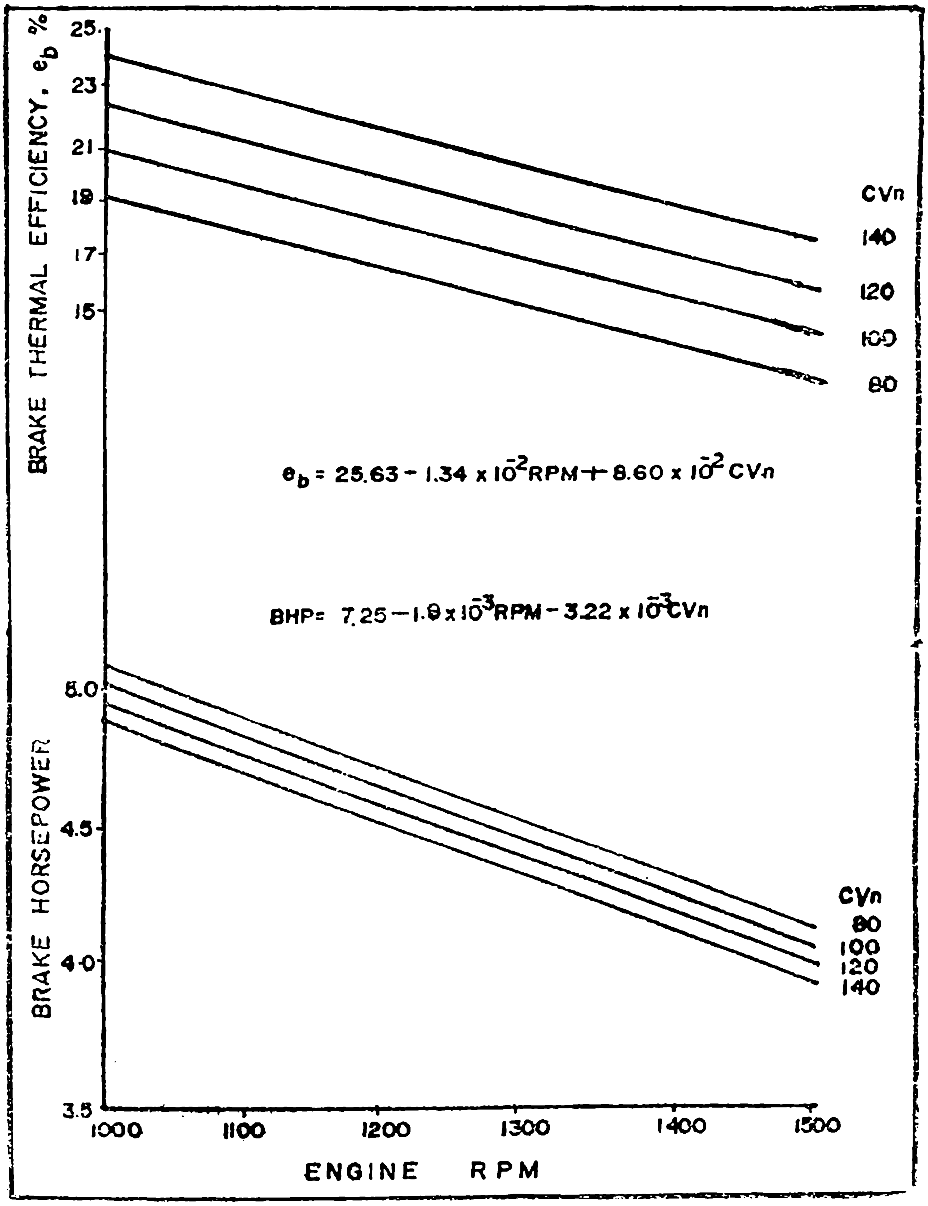


FIG. 4—*Brakehhorsepower and Thermal Efficiency of Diesel Engine (When using Producer Gas) as a Function of Gas Calorific Value and Engine Speed.*

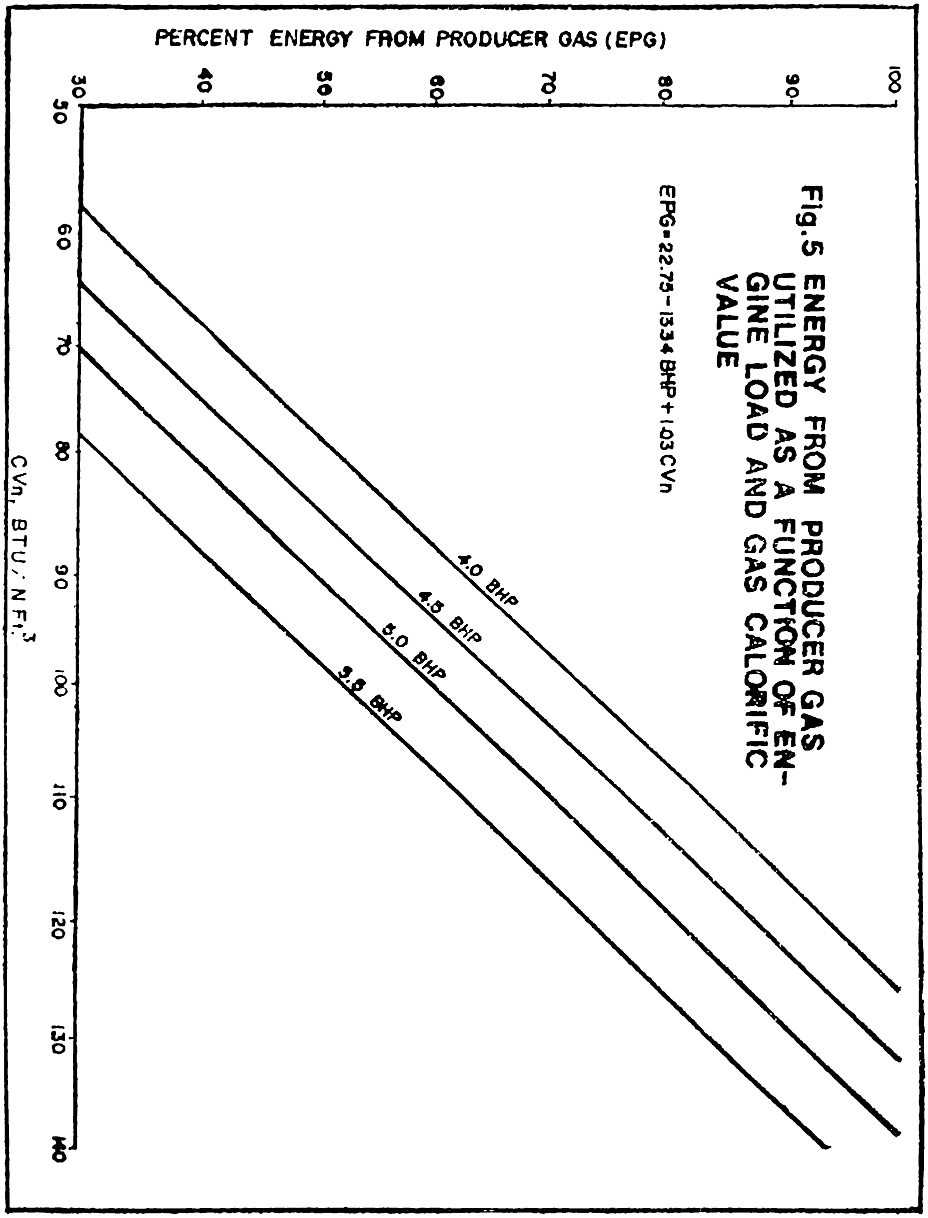


Fig.5 ENERGY FROM PRODUCER GAS UTILIZED AS A FUNCTION OF ENGINE LOAD AND GAS CALORIFIC VALUE

The reason for the deceases of BRP and e_b with RPM (Observation 1) has been mention before as being due to the increase of friction horsepower with increasing speed.

Observations (2) and (3), can be explained in terms of the percentage of energy from producer gas (EPG) utilized by the engine. The higher the calorific value of the gas, the higher was the percentage EPG. (See Fig. 5).

That a diesel engine using producer gas as the main fuel would have a a higher thermal efficiency that when it was using diesel fuel alone could be explained thermodynamically by the fact that for the same compression ratio, the Otto cycle (constant volume combustion of fuel) is more efficient than the Diesel cycle (constant pressure combustion). Constant volume combustion is closely approximated in a diesel engine when the combustible mixture of gas and air aspirated into the engine cylinder is ignited either by the normal injection of a small amount of diesel fuel, or pre-ignited by the high pressure compression of the mixture.

That the power output of an internal combustion engine (e.g. diesel engine) is decreased with increasing percentage use of producer gas, is explained by the fact that for the same engine displacement volume, less energy is contained in the volume of producer gas aspirated by the engine than in the volume of liquid fuel carburated or injected into the engine (See Annex 3).

The values of percentage EPG in Table 1 were correlated in terms of BHP and CV_a to give the equation:

$$EPG = 22.75 - 13.34\ (BHP) + 1.03\ (CV_n) \qquad (3)$$

Coefficient of correlation, $R^2 = 0.53$

The above equation is plotted in Fig. 5 which shows that percentage EPG increases with gas calorific value and decrease with engine load.

Fig. 4 and Fig. 5 are useful in approximating the conditions when the diesel engine is run on producer gas. For instance at a load of 4BHP, the CV_n must be 126 Btu/n ft^3 for 100 percent EPG (Fig. 5), the engine speed would be slighly less than 1500 RPM and e_b about 16.5 percent (Fig. 4). If the load is increased to 4.5 BHP without increasing CV_n, EPG becomes 96 percent, engine speed is 1225 RPM, and e_b 20 percent. If EPG is desired to be 100 percent at a load of 4.5 BHP, then VC_n must be 133 Btu/n ft^3, engine speed would be about 1220, and e_b about 21 percent.

Further Studies

All the experimental runs reported here involved the use of a prepared fuel, namely, coconut shell charcoal. Preliminary runs were made, with raw coconut shell, coir dust and rice husks as fuel for the gas producer. Problems however were encountered in cleaning the gas before using in the engine. The key to success in using these raw fuels which contain a lot of volatile matter and tars is a more suitably designed gas producer and an appropriate gas cleaning system. Future studies should give due consideration to those two aspects.

Conclusions and Recommendations

These studies show that:

(1) Coconut shell charcoal is a suitable fuel for a gas producer to supply fuel gas to a small diesel engine (5 Bhp) without the use of a complicated gas cleaning system.

(2) The more suitable design of a gas producer for use with the diesel engine is the down-draft reactor with multiple air inlet holes, so distributed around the cylindrical body of the reactor as to maintain a more uniform and a deeper combustion zone in the fuel bed. A constriction or "throat" at the combustion zone helps engine performance.

(3) The percentage of the total energy supplied to the engine and coming from producer gas increases with gas calorific value but decreases with engine load. For instance, with a gas heating value of 130 Btu/n $ft.^3$ and an engine output of 4 BHP, the engine can run on producer gas alone. At the same heating value but with a load of 5 BHP, the engine can run on 90 percent energy from producer gas and 10 percent from the liquid fuel.

(4) When conditions favor ignition by compression of the producer gas-air mixture, the engine can run on 100 percent producer gas. Such conditions include a richer fuel-air mixture and a higher gas heating value.

(5) The thermal efficiency of the engine when using producer gas is significantly higher than the efficiency when using liquid fuel. The exhaust gas when burning producer gas was clear and smokeless.

(6) It is recommended that future studies include the use of agricultural residues like coconut shells, wood waste, coir dust, rice hulls and the like as fuel for the gas producer. Also, project must

be initiated on the pilot plant applications of producer gas as fuel for larger, multiple cylinder diesel engines doing actual practical work.

TABLE 1. *Experimental Results of Dual Fuel Operation of a Five-Bhp Diesel Engine*

1. Run No.	1	2 to 5	6 to 9	10	11	12 to 15	16 to 19
2. Time, Min.	9	60	60	15	21	60	60
3. Engine RPV	910	964	982	932	924	953	979
4. BHP Output	4.41	4.48	4.72	4.43	4.63	4.75	4.83
5. Brake Therm. Eff., %	13.5	19.7	21.6	17.5	14.6	14.8	17.2
6. Fuel Used[a]	DO	PG/DO	PG/DO	DO	DO	PG/DO	PG/DO
7. Liquid Fuel lb/h	4.56	1.94	0.70	3.54	4.14	2.35	1.24
8. Exhaust Gas Temp. °F[b]		825	590		>1100	840	665
9. Producer Gas Temp. °F		215	238			205	250
10. Charcoal lb/h	0	2.42	4.62	0	0	4.61	5.72
11. Energy from Producer Gas (EPG), %	0	35	76	0	0	44	66
12. Cold Gas Therm. Eff., %		60	66			56	60
13. Producer Gas							
% CO_2		7.1	3.9			9.1	7.4
% O_2		0.5	0.2			0.4	0.2
% H_2		9.4	10.2			9.0	9.5
% CO		22.0	27.8			18.9	22.0
% CH_4		1	1			1	1
14. Net Calorific Value, CV_n, Btu/n ft^3		104	125			93	105

15. NOTES:

a) DO: Diesel Oil Only; PG/DO; Producer Gas-Diesel Oil Dual-Fuel Operation; CO: Crude Coconut Oil Only.

b) Engine exhaust gas temperatures when operating on liquid fuel only were beyond the range of the temperature gauge, and hence were not tabulated.

TABLE 1. *(Continued) Experimental Results of Dual Fuel Operation of a Five-Bhp Diesel Engine.*

1. Run No.	20	21	22 & 23	24 & 25	26 & 27	28 & 29	30 & 31
2. Time, Min.	15	15	30	30	30	30	30
3. Engine RPM	1002	924	931	995	1105	1026	992
4. BHP Output	4.27	4.58	4.59	4.43	4.76	4.64	5.06

5. Brake Therm. Eff., %	22.1	16.2	15.1	17.2	17.7	21.9	22.6
6. Fuel Used	PG/DO	DO	DO	PG/DO	PG/DO	PG/DO	PG/DO
7. Liquid Fuel lb/h	0.18	3.94	3.96	0.74	0.05	0.13	0.57
8. Exhaust Gas Temp. °F	605			694	663	679	716
9. Producer Gas Temp. °F	258			130	130	155	159
10. Charcoal lb/h	5.28	0	0	6.60	7.92	5.72	5.28
11. Energy from Producer Gas (EPG), %	93	0	0	78	99	95	80
12. Cold Gas Therm. Eff., %	63			57	63	66	64
13. Producer Gas							
% CO_2	5.8			8.8	5.8	4.0	5.3
% O_2	0.5			0.2	0.2	0.2	0.2
% H_2	9.6			9.2	9.8	10.2	9.9
% CO	24.2			19.7	24.6	27.6	25.5
% CH_4(c)	1			1	1	1	1
14. Net Calorific Value, CV_n, Btu/n ft^3	112			96	114	124	117
15. NOTES:							

c) The value of CH_4 equals 1% is an estimate for Runs 2 to 69.

TABLE 1. (*Continued*) *Experimental Results of Dual Fuel Operation of a Five-Bhp Diesel Engine.*

1. Run No.	32 & 33	34 & 35	36 & 37	38	39 & 40	41	42 & 45
2. Time, Min.	30	30	30	15	30	15	60
3. Engine RPM	1000	1034	995	962	892	962	994
4. BHP Output	5.27	4.75	4.48	4.40	4.32	4.06	4.76
5. Brake Therm. Eff., %	23.4	21.3	21.0	17.2	13.6	11.3	18.4
6. Fuel Used	PG/DO	PG/DO	PG/DO	PG/DO	CO	PG/CO	PG/CO
7. Liquid Fuel lb/h	0.51	0.13	0.48	1.61	4.82	3.39	1.23
8. Exhaust Gas Temp. °F	711	700	728	765		878	786
9. Producer Gas Temp. °F	173	187	206	201		100	115
10. Charcoal lb/h	5.27	6.16	5.28	4.36	0	4.40	5.49
11. Energy from Producer Gas (EPG), %	83	96	83	52	0	38	69

12. Cold Gas Therm. Eff., %	66	65	62	56		58	61
13. Producer Gas							
% CO_2	4.0	5.0	6.3	9.0		8.1	7.0
% O_2	0.2	0.2	0.2	0.2			
% H_2	10.2	10.0	9.7	9.1		9.4	9.7
% CO	27.6	26.0	23.8	19.4		21.2	23.6
% CH_4	1	1	1	1		1	1
14. Net Calorific Value, CV_n, Btu/n ft^3	124	119	111	95		102	108
15. NOTES:							

TABLE 1. (*Continued*) *Experimental Results of Dual Fuel Operation of a Five-Bhp Diesel Engine.*

1. Run No.	46 & 47	48 & 49	50 & 51	52 to 55	56 to 59	60 to 63	64 to 67
2. Time, Min.	30	30	30	65	60	60	60
3. Engine RPM	950	967	1047	1076	1097	1105	1078
4. BHP Output	4.52	5.17	5.41	5.15	5.12	5.47	5.55
5. Brake Therm. Eff., %	13.9	12.4	11.5	13.0	13.4	18.9	20.9
6. Fuel Used	CO	CO	PG/CO	PG/CO	PG/CO	PG/CO	PG/CO
7. Liquid Fuel lb/h	4.95	6.36	4.61	3.75	2.82	1.52	0.94
8. Exhaust Gas Temp.°F							
9. Producer Gas Temp. °F			100	100	110	110	120
10. Charcoal lb/h	0	0	5.28	4.47	6.16	5.72	6.38
11. Energy from Producer Gas (EPG), %			36	37	51	65	77
12. Cold Gas Therm. Eff., %	0	0	60	63	60	62	60
13. Producer Gas							
% CO_2			7.5	6.1	7.4	6.2	7.5
% O_2							
% H_2			9.6	9.9	9.6	9.8	9.6
% CO			22.2	24.5	22.3	24.3	22.2
% CH_4			1	1	1	1	1
14. Net Calorific Value, CV_n, Btu/n ft^3			105	114	106	113	105
15. NOTES:							

TABLE 1. (*Continued*) *Experimental Results of Dual Fuel Operation of a Five-Bhp Diesel Engine.*

1. Run No.[d]	68 & 69	70 & 71	72	73 to 76	77 to 80	81 to 88	89 to 96
2. Time, Min.	30	30	24	65	60	120	120
4. BHP Output	5.60	4.04	3.99	4.16	4.13	4.06	4.02
3. Engine RPM	1043	1008	1000	1043	1043	1311	1426
5. Brake Therm. Eff., %	21.3	12.6	14.6	16.7	22.1	18.7	16.7
6. Fuel Used	PG/CO	CO	DO	PG/DO	PG/DO	PG/DO	PG/DO
7. Liquid Fuel lb/h	0.98	4.83	3.56	1.00	0.61	0.12	0.05
8. Exhaust Gas Temp. °F				812	745	768	780
9. Producer Gas Temp. °F	140			98	114	127	131
10. Charcoal lb/h	5.71	0	0	5.28	3.96	6.16	6.16
11. Energy from Producer Gas (EPG), %	76	0	0	69	75	96	98
12. Cold Gas Therm. Eff., %	65			62	67	64	72
13. Producer Gas							
% CO_2	4.5			5.5	5.1	5.4	3.7
% O_2				0.6	0.5	0.3	0.4
% H_2	10.2			7.0	9.4	8.7	10.6
% CO	27.1			23.4	23.1	26.2	26.2
% CH_4	1			1.9	1.8	1.6	2.5
14. Net Calorific Value, CV_n, Btu/n ft^3	123			110	115	121	134
15. NOTES:							

d) Starting with Run 73, the gas producer used was the re-designed reactor with 12 inlet air holes and a combustion zone "throat".

TABLE 1. (*Continued*) *Experimental Results of Dual Fuel Operation of a Five-Bhp Diesel Engine.*

1. Run No.[e]	97	98	99 to 102	103 to 106	107 to 115	116 & 117	118
2. Time, Min.	21	15	60	60	140	30	15
3. Engine RPM	1000	1030	1211	1492	1458	1016	1007
4. BHP Output	3.92	4.14	4.44	3.84	3.95	4.17	4.95
5. Brake Therm. Eff., %	15.0	13.7	12.8	15.8	14.8	14.5	13.1
6. Fuel Used	DO	CO	PG/CO	PG/CO	PG/CO	CO	CO
7. Liquid Fuel lb/h	3.42	4.57	0.12	0	0.02	4.36	5.74
8. Exhaust Gas Temp. °F			754	776	767		
9. Producer Gas Temp. °F			100	124	135		
10. Charcoal lb/h	0	0	10.12	6.60	7.54	0	0

11. Energy from Producer Gas (EPG), %	0	0	98	100	100	0	0
12. Cold Gas Therm. Eff., %			63	69	66		
13. Producer Gas							
% CO_2			6.7	4.2	4.6		
% O_2			0.5	0.8	0.6		
% H_2			9.4	9.3	9.4		
% CO			22.3	22.4	26.0		
% CH_4			1.7	1.9	1.6		
14. Net Calorific Value, CV_n, Btu/n ft^3			111	113	123		
15. NOTES:							

e) Starting with run 73, the values given for percentage CH_4 are obtained from actual gas analysis (Burrell).

TABLE 1. *(Continued) Experimental Results of Dual Fuel Operation of a Five-Diesel Engine*

1. Run No.	119 to 122	123 to 126	127 to 130	131 to 134	135 to 138	139 to 142	143	144
2. Time, Min.	60	60	60	60	60	60	15	15
3. Engine RPM	1067	1252	1531	1317	1135	1062	1008	942
4. BHP Output	4.37	4.93	4.45	4.68	4.77	5.22	5.07	5.21
5. Brake Therm. Eff., %	17.2	27.3	18.2	19.4	23.8	24.8	14.0	12.6
6. Fuel Used	PG/CO	PG/CO	PG/CO	PG/CO	PG/CO	PG/CO	CO	CO
7. Liquid Fuel lb/h	2.14	0.00	0.01	0.01	0.03	0.76	5.50	6.28
8. Exhaust Gas Temp. °F	817	766	798	759	721	768		
9. Producer Gas Temp. °F	93	124	133	136	130	126		
10. Charcoal lb/h	4.84	5.28	6.60	6.82	5.50	4.62	0	0
11. Energy from Producer Gas (EPG), %	45	100	100	100	99	76	0	0
12. Cold Gas Therm. Eff., %	44	65	70	66	68	66		
13. Producer Gas								
% CO_2	11.3	4.5	2.8	4.5	4.0	3.8		
% O_2	1.7	0.5	0.4	0.5	0.2	0.4		
% H_2	5.0	9.5	10.7	8.5	10.8	8.8		
% CO	13.8	26.3	27.2	23.9	26.0	25.4		
% CH_4	1.0	1.0	0.9	1.8	1.0	1.1		
14. Net Calorific Value, CV_n, Btu/n ft^3	66	118	123	115	121	114		
15. NOTES:								

Annex 1. Equations for Evaluating Engine Performance; Sample Calculations

1. Brake horsepower:

$$BHP = \frac{(P)\ (RPM)}{1500.6} \qquad (1)$$

where P is the force in pounds exerted by the arm of the prony brake on the scale. It is equal to the scale reading minus the tare weight in pounds. The prony brake constant is 1500.6 in the denominator of equation (1).

2. Higher Heating Values:

a) For Producer Gas (Btu/n ft^3):

$$HV_1 = 3.41\ (\%\ CC) + 3.43\ (\%\ H_2) + 10.67\ (\%\ CH_4)$$

(higher heating value of dry gas at 0°C)

b) For Diesel Fuel (Btu/lb):

$HV_2 = 19{,}494$

c) For Crude Coconut Oil (Btu/lb):

$HV_2 = 16{,}775$

d) For Charcoal (Btu/lb)

$HV_3 = 12{,}971$

3. Brake Thermal Efficiency:

$$e_b = \frac{2545\ (BHP)}{13.76\ F_1\ (HV_1) + F_2(HV_2)} \times 100\% \qquad (2)$$

$$F_1 = \frac{(69.3)\ (359)\ W_1}{12(\%\ CO_2 + \%CO + \%CH_4)\ 13.76} = \frac{150.67\ W_1}{\%CO_2 + \%CO + \%CH_4} \qquad (3)$$

where

W_1 = Charcoal consumed in lb/h

F_1 = Producer gas consumed, in lb/h

F_2 = Liquid fuel consumed, in lb/h

13.76 = Specific volume of producer gas in ft^3/lb at 0°C

69.3 = % C in the ultimate analysis of charcoal (5.5% H, 22.2% O, 3% ash)

Substituting equation (3) and the corresponding HV in equation (2), the thermal efficiency when the engine is run on producer gas and diesel oil becomes

$$e_b = \frac{100\ BHP}{E_1 + E_2} \qquad (4)$$

where

$$E_{\bar{1}} = (W_1) \frac{2.778\ (\%CO) + 2.794\ (\%H_2) + 8.694\ (\%CH_4)}{\%CO_2 + \%CO + \%CH_4} \qquad (5)$$

and

$$E_2 = 7.660\ F_2 \quad \text{for diesel fuel} \qquad (6)$$

or

$$E_2 = 6.591\ F_2 \quad \text{for crude coconut oil} \qquad (7)$$

4. Percentage Energy from Producer Gas (EPG):

In equation (4), the denominator represents the total energy supplied to the engine, divided as follows:

E_1 = Energy from producer gas

E_2 = Energy from the liquid fuel

Therefore:

$$EPG = \frac{100\ E_{\bar{1}}}{E_1 + E_2} \qquad (8)$$

5. Cold Gas Thermal Efficiency:

$$\text{Cold Gas Efficiency} = \frac{\text{Net Calorific Value of Gas, } CV_n}{\text{Heating in Solid Fuel Used}} \text{ x } 100$$

$CV_n = 3.18\ (\%CO) + 2.70\ (\%H_2) + 8.94\ (\%CH_4)$ Btu/n ft^3 (at 60°F, saturated)

$$\text{Heat in Solid Fuel Used} = \frac{(HV_n)\ 12(\%\ CO_2 + \%\ CO + \%\ CH_4)}{379\ (\%\ C)}$$

For charcoal, the net heating value HV_n = 12,510 Btu/lb and and % C = 69.3

Substituting in the gas efficiency equation:

Cold Gas Efficiency =

$$\frac{55.64\ (\%\ CO) + 47.24\ (\%\ H_2 + 156.59\ (\%\ CH_4)}{\%\ CO_2 + \%\ CO + \%\ CH_4} \qquad (9)$$

6. Sample Calculation:

Find the engine performance parameters for Run 38 when the following data are known:

BHP = 4.40, Diesel fuel consumption = 1.61 lb/h.

Charcoal consumption = 4.36 lb/h.

Producer gas analysis: 9.0% CO_2, 0.2% O_2, 9.1% H_2, 19.4% CO 1.0% CH_4

Solution:

From eq (5)

$$E_{\bar{1}} = 4.36 \frac{2.778(19.4) + 2.794(9.1) + 8.694(1.0)}{9.0 + 19.4 + 1.0} = 13.052$$

From eq (6)

$$E_2 = 7.660\ (1.61) = 12.333$$

From eq (4)

$$e_b = \frac{100\ (4.40)}{13.052 + 12.333} = 17.33\%$$

From eq (9)

$$EPG = \frac{100(13.052)}{13.052 + 12.333} = 51.4\%$$

From eq (9)

$$\text{Cold Gas Efficiency} = \frac{55.64(19.4 + 47.24(9.1) + 156.59(1.0)}{9.0 + 19.4 + 1.0}$$

$$= 56.66\%$$

$CV_n = 3.18(19.4) + 2.7(9.1) + 8.95(1) = 95.21$ Btu/n ft^3

Annex 2. Statistical t-Test for the Means of the Brake Thermal Efficiencies

We want to test the null hypothesis:

$$\mu_1 = \mu_2$$

where μ_1 and μ_2 are the means of two normal populations from which independent random samples $(x_1, x_2, \ldots, x_{n_1})$ and $(y_1, y_2, \ldots y_{n_2})$ are obtained.

Define

$$\bar{x} = \frac{1}{n_1} \sum_{i=1}^{n_1} x_i$$

$$\bar{y} = \frac{1}{n_2} \sum_{i=1}^{n_2} y_i$$

$$t = \frac{\bar{x} - \bar{y}}{\sqrt{\frac{1}{n_1} + \frac{1}{n_2}}\sqrt{\frac{\sum x_i^2 - n_1\bar{x}^2 + \sum y_i^2 - n_2\bar{y}^2}{n_1 + n_2 - 2}}}$$

with $n_1 + n_2 - 2$ degrees of freedom:

(Reference: Hewlett-Packard HP-25 Application Programs, p. 124)

The above statistic was used to test the null hypothesis that the means of the brake thermal efficiencies of the engine (found in Table 1) when using dual fuel and when using liquid fuel alone, are equal. The results are as follows:

a) Comparing the means of the brake efficiencies when using producer gas (18.41%) with the mean when using diesel oil alone (15.20%) the computed t-value was 2.44 with 126 degrees of freedom. For the hypothesis to be true (that the means are equal), the t value in 98 out of 100 cases must fall within $\pm$ 2.33. The hypothesis is therefore rejected since the computed $t = 2.44$ is outside $\pm$ 2.33.

b) Similarly, the hypothesis that the means are equal when comparing engine operation using producer gas (18.41%) with that when using crude coconut oil alone, (13.39%) is rejected. The computed t is 5.05, and t must fall within $\pm$ 2.58 in 99 out of 100 cases, for 132 degrees of freedom for the hypothesis to be true.

c) Comparing engine operation when using diesel oil (15.20%) alone with that when using crude coconut oil alone (13.39%) results in a computed t value of 4.39. The t value must fall within $\pm$ 2.845 in 99 out of 100 cases, for 20 degrees of freedom, in order for the hypothesis to be true.

Annex 3. Theoretical Reduction in Power Output When Producer Gas in Used in the Diesel Engine

Let A_1 — theoretical air required for the combustion of producer gas, n ft^3/n ft^3

A_2 — theoretical air required for the combustion of diesel fuel, nft^3/lb

C_{n1} — net calorific value of producer gas, Btu/n ft^3

C_{n2} — net calorific value of diesel fuel, Btu/lb

F_1 — weight of producer gas consumed by the engine, lb/h

F_2 — weight of pilot fuel (diesel) consumed by the engine for ignition purposes, lb/h

F_3 — weight of diesel fuel consumed by the engine when running on diesel alone, lb/h

E_2 — energy input to engine when running on dual fuel (producer gas and diesel), Btu/h

E_3 — energy input to engine when running on diesel alone, Btu/h

EPG — percentage or fraction of total energy input to engine supplied by producer gas.

If producer gas is aspirated together with air into the engine, the sum of the volumes of (a) producer gas, (b) air for combustion of producer gas, and (c) air for combustion of the pilot fuel, is theoretically equal to the volume of air aspirated by the engine when running on diesel fuel alone (assuming that no excess air is used and that the volume of liquid injected into the engine is negligible).

Therefore

$$13.76\ A_1\ F_{\bar{1}} + 13.76\ F_1 + A_2\ F_2 = A_2\ F_3 \qquad (1)$$

or

$$\frac{F_3}{F_2} = 1 + 13.76\ \frac{F_1}{F_2}\ \left(\frac{1 + A_{\bar{1}}}{A_2}\right) \qquad (2)$$

where 13.76 is the specific volume of producer gas in ft³/lb.

The energy input when engine is on dual fuel is:

$$E_2 = 13.76\ F_{\bar{1}}\ C_{n1} + F_2\ C_{n2} \qquad (3)$$

When on diesel fuel only, the input is:

$$E_3 = F_3\ C_{n2} \qquad (4)$$

From equations (2), (3) and (4)

$$\frac{E_2}{E_3} = \frac{1 + (13.76\ F_1/F_2)\ C_{n\bar{1}}/C_{n2}}{1 + (13.76\ F_1/F_2)\ (1 + A_{\bar{1}})/A_2} \qquad (5)$$

Equation (5) gives the ratio in energy input into the engine when the engine runs on dual fuel compared to when it runs on diesel fuel alone. If the thermal efficiency remains constant, then equation (5) is also the ratio of the power outputs.

From Table 76 in Technical Data on Fuel (Sixth Edition, 1961) by H. M. Spiers:

$$A_1 = 0.00775\ C_{n1} \qquad \text{nft}^3/\text{nf}^3 \qquad (6)$$

$$A_2 = 0.00756\ C_{n2} + 32\ \text{nft}^3/\text{lb} \qquad (7)$$

Substituting equations (6) and (7) in (5):

$$\frac{E_2}{E_3} = \frac{1 + (13.76\ F_1/F_2)(C_{n\bar{1}}/C_{n2})}{1 + (13.76\ F_1/F_2)(1 + 0.00775\ C_{n\bar{1}})/(32 + 0.00758\ C_{n2})} \quad (8)$$

From the definition of EPG (the ratio of energy from producer gas to total energy input):

$$EPG = \frac{13.76\ F_1\ C_{n\bar{1}}}{13.76\ F_1\ C_{n\bar{1}} + F_2\ C_{n2}} \quad (9)$$

or

$$13.76\ (F_1/F_2) = \frac{(EPG)\ (C_{n2})}{(1 - EPG)\ (C_{n1})} \quad (10)$$

Substituting equation (10) in (8):

$$\frac{E_2}{E_3} = \frac{1 + \dfrac{EPG}{1 - EPG}}{1 + \dfrac{(EPG)(C_{n2})}{(1\text{-}EPG)(C_{n\bar{1}})}\ \dfrac{1 + 0.00775\ C_{n\bar{1}}}{32 + 0.00756\ C_{n2}}} \quad (11)$$

Equation (11) was solved for varying values of EPG (percent) and $C_{n\bar{1}}$ (Btu/n ft^3) with a fixed value of C_{n2} (19,000 Btu/lb). The results are shown in the Table A3. Subtracting from 100 percent the values in Table A3 will give the theoretical percentage reduction in power.

$C_{n\bar{1}}$ % EPG	100	120	140	160
0	100	100	100	100
25	81	84	87	89
50	68	73	77	80
75	59	64	69	72
100	52	57	62	66