STUDIES ON THE PRACTICAL APPLICATION OF PRODUCER GAS FROM AGRICULTURAL RESIDUES AS SUPPLEMENTARY FUEL FOR DIESEL ENGINES

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Introduction

The Philippines is an agricultural country where crop irrigation has become essential to accelerate food production. Many irrigation systems use diesel engines. In rice production, for instance, a government-supported cooperative-type farm based organization has installed since 1975 a total of 334 diesel engine-driven pumps of sizes ranging from 15 to 150 horsepower (1 horsepower is 746 watts) which enabled farmers to harvest two and even three crops in a year. With the continuing increase in price coupled with the scarcity of fuel oils, particularly diesel fuel, it is becoming more and more difficult to continue operating these diesel engine-driven pumps.

The objective of these studies therefore is to assure the continued operation of existing diesel engine, particularly those used in crop irrigation, by converting them to dual-fuel engines with minimum modifications and using producer gas as supplementary fuel. It includes the design of a simple gas producer that can be fabricated inexpensively.

A diesel engine with a little modification, can be operated as a dual-fuel engine, that is, an engine that uses both gaseous fuel and liquid injection fuel. Normally, a diesel engine aspirates air during the intake stroke of the piston and compresses this to a high pressure and temperature. The compression ratio of the diesel engine is high enough so that the temperature of the air inside the engine cylinder after the compression stroke attains a sufficiently high level so as to ignite the diesel fuel that is injected into it. In dual-fuel operation, a mixture of gasseous fuel and air in the proper proportion is aspirated into the engine and compressed during the compression stroke. The gaseous fuel-air mixture is on the lean side so that it does not pre-ignite during the compression stroke. Only the injection of the liquid fuel initiates ignition and final combustion of both gaseous and liquid fuels. Since combustion of the gas aspirated with air provides power, a much less amount of liquid fuel, compared to straight diesel operation, need be injected to produce a given power output. Thus a significant savings in the liquid injection fuel (diesel oil) is affected.

The economics of dual-fuel operation becomes favorable if the gaseous fuel can be obtained from indigenous sources. One such gaseous fuel is producer gas from agricultural residues. Studies on the production of gas from agricultural wastes have earlier been reported (1).

Experimental Work With a Single-Cylinder Engine

Initial experimental work on the use of producer gas from coconut shell charcoal was done with the support of the Philippine Coconut Authority (2). Charcoal as fuel was chosen because the cleaning of the gas from charcoal was simpler due to less tar in the gas.

Figure 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 inches (1 inch is 2.54 centimeters). The engine was manufactured by Lister-Blackstone.

In Figure 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 of 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 diesel oil was likewise mounted on a weighing scale. 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 was started in the normal way by hand cranking, with the air intake valve fully open and the producer gas valve fully closed. Thus the engine was run on diesel fuel alone at the start.

The engine torque output was measured by a prony brake mounted on another platform scale, and the engine RPM by hand tachometer. The brake horsepower output in each run was thus determined.

The gas producer was a suction-type, downdraft reactor with 12 air holes around the mid-section of the cylindrical body, and a single gas outlet at the bottom. Connected to the gas outlet was a cyclone separator (3) to remove entrained dust and charcoal fines out of the gas before it went to the engine. The gas scrubber (4) and filter (8) were later additions when fuels with high tar and volatile matter contents were used.

At 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 aspirate 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 aspirated into the engine, until a minimum use of liquid fuel, that which was required only for ignition, was reached.

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, 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 volatile matter was used. Therefore after about half of the experimental runs were finished, the 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 from the original 10 inches and the longitudinal cross section of the reactor now exhibited a constriction or a "throat" at the combustion zone. The purpose of this throat was to make combustion more intense at this zone. Combustion rate per unit or cross-sectional area would now be higher and hence temperature higher. Thus, the combination of higher 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.

A mass of data was obtained in evaluating the performance of the engine as a dual-fuel engine using producer gas from charcoal. The parameters used for evaluating performance are the brake-horsepower (BHP) output, the brake thermal efficiency (e_b) of the engine, and the percentage energy from producer gas (EPG) utilized in the engine. EPG is defined as the ratio of the heat released by the combustion of producer gas aspirated into the engine to the total heat released by both liquid injection fuel (diesel) and producer gas, multiplied by 100 to express as a percentage. Derivation of the equation for EPG and brake thermal efficiency is found in the Appendix.

To obtain a more convenient form, the mass of data was reduced to multiple linear regression equations to give the following relationships:

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

Coefficient of correlation, $R^2 = 0.48$

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

$R^2 = 0.41$

$$\mathrm{EPG} = 22.75 - 13.34\ (\mathrm{BHP}) + 1.03\ (\mathrm{CVn}) \qquad (3)$$

$R^2 = 0.53$

CVn in the above equationsn is the net calorific value of the gas in Btu/ft^3 at NTP, i.e., at normal temperature and pressure of 273 K and 1 atmosphere. (One Btu/ft^3 is 37.25913 kJ/m^3).

The range of values of the parameters used in obtaining the above equations are as follows:

TABLE 1. *Comparative Performance of a 5-Brakehorsepower (3.7-kW) Diesel Engine Run in (A) Dual-Fuel Mode, and (B) Single-Fuel (Diesel) Mode*

Fuel	*Mode*	*RPM*	*Brake horse-power*	*Pound fuel per Bhp-hour*		*% Diesel Saved*
				Liquid	*Solid*	
Charcoal	A	1043	4.1	0.148	1.0	83
Diesel	B	1000	4.0	0.892	0	—
Coal	A	1288	4.8	0.246	1.3	80
Diesel	B	1246	4.9	1.261	0	—
Coconut Shell	A	1212	4.7	0.208	2.6	72
Diesel	B	1208	4.2	0.730	0	—
Wastewood	A	1221	4.1	0.357	2.8	62
Diesel	B	1237	4.1	0.950	0	—
Rice Hulls	A	1214	3.4	0.323	5.6	59
Diesel	B	1170	3.3	0.795	0	—
Corn Cobs	A	1287	4.5	0.516	1.0	31
Diesel	B	1222	4.2	0.752	0	—

TABLE 2. *Typical Data on Comparative Performance of a Six-Cylinder Compression Ignition Engine Run in (1) Dual-Fuel Mode, and (2) Diesel Fuel Mode.*

Run	*Run Duration Minutes*	*Engine Mode*	*RPM*	*BHP**	*Fuel Consumption kg/h*		*%, e_b*
					Diesel	*Charcoal*	
1	120	Dual	1671	39	5.220	15.6	20.3
2	30	Diesel	1631	41	9.860	0	25.2
3	60	Dual	1619	45	6.880	15.8	20.3
4	60	Dual	1568	28	1.818	14.4	22.5
5	60	Dual	1665	31	1.504	19.8	19.7
7	15	Diesel	1590	31	7.384	0	25.2
8	60	Dual	1750	31	1.374	20.4	19.8
9	60	Dual	1747	37	1.847	20.2	22.0
15	15	Diesel	1800	49	11.640	0	25.6
16	60	Dual	1800	42	4.162	21.9	18.1

* 1 BHP = 746 watts

(1)	Engine speed:	900	RPM	1550
(2)	Brake-horsepower:	4	BHP	6
(3)	Net calorific value in Btu/ft^3	100	CVn	135
(4)	Percent energy producer gas:	75	EPG	100

It is interesting to note that when the engine was run on diesel oil alone, the experimental points indicate that the engine RPM was in the vicinity of 1000 RPM at which point the speed governor was set. However, for short periods of time when conditions were favorable, the engine could run on straight producer gas (EPG = 100%) without any need for the injection of diesel fuel and the maximum speed attained was greater than 1500 RPM, an increase of more than 50 percent. This was understandable since at 100 percent EPG when no liquid fuel was being used, the engine speed was not being controlled by the governor any longer 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 due to the fact that the right combination of producer gas-air mixture, gas calorific value (CVn), and engine load (BHP) led 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 calorific value of the gas had improved to about 125 Btu/ft^3. A precaution that had to be observed, however, was to adjust the gas-air mixture to make it leaner as the gas calorific value continued to improve, to prevent severe knocking of the engine.

Thus from equation (3), for an engine load of 4 BHP, CVn must be 127 Btu/ft^3 for EPG to be 100 percent. From equation (1), the engine speed would be 1496 RPM, and from equation (2), brake thermal efficiency e_b is 16.5 percent.

If the load is to be increased to 5 BHP with CVn the same at 127 Btu/ft^3, from equations (1), (2), and (3), the engine speed becomes 966 RPM, the brake thermal efficiency is 22.4 percent, and the energy supplied by producer gas, (EPG) is 87 percent. This means that only 13 percent of the energy is supplied by diesel fuel.

Other fuels such as coconut shells, woodwaste, coal, and rice hulls were tried for gasification in the gas producer and utilization in the 5-BHP diesel engine. Typical results of these trials are shown in Table 1. However, when using these fuels other than charcoal, the problem of cleaning the gas of tar in the present gas cleaning equipment (gas scrubber with a water spray) had not been satisfactorily solved. The engine had to be dismantled after about 50 hours of operation to clean its insides of tar deposits.

Experimental Work With a Six-Cylinder Engine

In a project supported jointly by the Ministry of Energy, the National Science Development Board, the National Irrigation Administration, and the University of the Philippines, experimental work on the performance of a "Fordson" 6-cylinder, 65 brake horsepower diesel engine was conducted at the U.P. College of Engineering (3). The final objective was to use this engine in dual-fuel operation for irrigation of 40 hectares (400,000m^2) of riceland in Siniloan, Laguna. Charcoal was used for the gas producer.

The experimental procedures were essentially the same as for the single-cylinder engine experiments except that the brake horsepower output was about ten times more in the larger multi-cylinder engine. For measuring power, a hydraulic dynamometer was used. The set-up is shown in Figure 2 except that the weighing scales on which the fuel tank and the gas producer were mounted during the experiments are not drawn. The dimensions of the gas producer are shown in Figure 3.

Table 2 shows typical data obtained when operating the engine both in dual-fuel mode and in diesel fuel mode. Again, the data are reduced to a more convenient form by multiple regression equations:

(1) Brake Horsepower.

(a) Diesel fuel mode:

$$BHP = 4.14 \ (RPM/1000)^{-0.98}(F_2)^{1.22} \qquad (4)$$
$$R^2 = 0.97$$

where F^2 is diesel fuel consumption in kilogram per hour (kg/h).

(b) Dual fuel mode:

$$BHP = 236 \ (RPM/1000)^{2.80}(EPG)^{-0.78} \qquad (5)$$
$$R^2 = 0.88$$

where EPG is percent energy from producer gas. Also, a good correlation with IFC (Injection Fuel Consumption) is obtained:

$$BHP = 10.56 \ (RPM/1000)^{1.90}(IFC)^{0.24} \qquad (6)$$
$$R^2 = 0.84$$

(2) Brake Thermal Efficiency:

(a) Diesel fuel mode:

$$e_b = 17.49 \ (RPM/1000)^{-0.93} (BHP)^{0.23} \tag{7}$$
$$R^2 = 0.65$$

(b) Dual fuel mode: Poor correlations were obtained.

(3) Charcoal Consumption W_1 in kg/h (with 10 to 20% moisture)

(a) Correlation with RPM and EPG:

$$W_1 = 1.87 \ (RPM/1000)^{2.26} (EPG)^{0.25} \tag{8}$$
$$R^2 = 0.70$$

(b) Correlation with RPM and BHP:

$$W_1 = 10.45 \ (RPM/1000)^{3.14} (BHP)^{-0.31} \tag{9}$$
$$R^2 = 0.71$$

(c) Correlation with RPM and IFC:

$$W_1 = 4.91 \ (RPM/1000)^{2.57} (IFC)^{-0.05} \tag{10}$$
$$R^2 = 0.67$$

(4) Diesel Fuel Consumption F_2 in kg/h During Diesel Fuel Mode:

$$F_2 = 0.349 \ (RPM/1000)^{0.918} (BHP)^{0.771} \tag{11}$$
$$R^2 = 0.98$$

The range of values of the parameters in the above equations (4) to (11) are as follows:

(1) Engine speed:	$1550 < RPM < 1850$
(2) Brake horsepower:	$25 < BHP < 50$
(3) Percent energy from producer gas:	$50 < EPG < 90$
(4) Injection fuel consumption in dual fuel mode, kg/h:	$1 < IFC < 7$

EPG or percentage energy from producer gas is obtained indirectly from the following relations (See Appendix):

$$EPG = \frac{100 \ E_1}{E_1 + E_2} \tag{17}$$

where E_1 = Energy from producer gas
E_2 = Energy from the liquid fuel

EPG can be measured directly in the field (by means of a graduated burette connected to the fuel tank) by first measuring the injection fuel consumption IFC during dual fuel operation, then the fuel consumption, F_2 during diesel fuel mode at the same engine RPM and BHP.

The percentage diesel fuel saving is calculated as follows:

$$FS = \frac{100 \ (F_2 - IFC)}{F_2} \tag{13}$$

Table 3 compares the values of EPG and FS for the dual fuel experimental runs in Tables 2 and 4. It is seen that FS and EPG are practically the same.

TABLE 3. *Percentage in Diesel Fuel Saving (FS) and Energy from Producer Gas (EPG)*

Run No.	*RPM*	*BHP*	*Diesel Fuel Consumption kg/h*		*FS, %*	*EPG, %*
			Dual Mode	*Diesel Mode*		
1	1671	39	5.22	9.39	44	55
3	1619	45	6.88	10.21	33	49
4	1568	28	1.82	6.92	74	76
5	1665	31	1.50	7.89	81	84
8	1750	31	1.37	8.22	83	86
9	1747	37	1.85	9.39	80	82
12	1795	46	3.02	11.36	73	70
13	1800	40	1.99	10.33	81	80
16	1800	42	4.16	10.67	61	70
17	1800	38	2.83	9.83	71	79
20	1807	44	3.30	11.19	70	73
21	1803	43	2.68	10.86	76	75

TABLE 4. *Typical Data on Gas Producer Performance During Dual-Fuel Engine Operation of a Six-Cylinder Engine.*

Run No.	1	3	4	5	9	16
Engine	1671	1619	1568	1665	1747	1800
BHP*	39	45	28	31	37	42
Dry Gas Analysis						
% CO_2	3.7	3.3	2.8	4.3	2.9	4.9
% O_2	0.3	0.5	0.3	0.3	0.2	0.6
% CO	27.5	25.7	28.5	28.3	29.5	26.2
% H_2	11.1	11.4	10.0	12.2	11.2	16.7
% CH_4	1.0	0.5	0.7	0.8	0.8	0.8
CV, Btu/nft^3*						
Gross	143	132	139	147	148	155
Net	126	117	124	130	131	136
Gas Temperature, °F*						
T^1, before scrubber	430	385	378	662	673	555
T_2, after scrubber	76	97	72	90	100	88
Cold Gas Efficiency						
N_{th}, %	84	85	83	84	85	91

* Conversion factors
1 BHP = 746 watts
1 Btu/ft^3 = 37.25913 kJ/m^3
°C = 9/5 (°F − 32)

TABLE 5. *Typical Data on Gas Producer Performance During Dual-Fuel Engine Operation of a Single-Cylinder Engine.*

Run No.	12	26	32	77	81	89
Engine RPM	953	1105	1000	1043	1311	1426
BHP	4.8	4.8	5.3	4.1	4.1	4.0
Dry Gas Analysis						
% CO_2	9.1	5.8	4.0	5.1	5.4	3.7
% O_2	0.4	0.2	0.2	0.5	0.3	0.4
% CO	18.9	24.6	27.6	23.1	26.2	26.2
% H_2	9.0	9.8	10.2	9.4	8.7	10.6
% CH_4	1.0	1.0	1.0	1.8	1.6	2.5
Calorific Value, Net, Btu/nft^3	93	114	124	115	121	134
Gas Temperature (without scrubber) °F	205	130	173	114	127	131
13 Cold Gas Efficiency N_{th}, %	62	70	73	74	71	80
Energy from Producer Gas, EPG, %	44	99	83	75	96	98

TABLE 6. *Comparative Performance Data Between the Multi-Cylinder and Single Cylinder Engine-Gas Producer Systems.*

Averages of:	*Single-Clinder*	*Multi-Cylinder*
1. Brake horsepower	4.5	38.6
2. Engine Speed RPM	1048	1735
3. Specific Diesel Fuel Consumption kg/Bhp/h	θ.073	0.079
4. Specific Charcoal Consumption kg/Bhp/h	0.535	0.500
5. Percent, e_b	19.5	21.5
6. Percent EPG	79	73
7. Gas Thermal Eff., %	60	85
8. Gasification Rate (based on grate aera) kg/m^2/h	48	50
9. Gasification Rate, kg/m^2/h (based on throat area)	133	206
10. Net Calorific Value Btu/nft^3	113	128

The above equations were useful in estimating the performance of the engine when it was brought to Siniloan, Laguna to power the irrigation pump. In about 60 hours of test spread over 12 days, the engine in dual-fuel mode when driving the pump at 1200 RPM showed an average injection fuel consumption rate of 2.58 kg/h. This was equivalent to a 49 percent diesel fuel saving since during straight diesel operation at the same engine speed, the fuel consumption was

5.09 kg/h. From equations (4), (5) and (6) the power delivered by the engine was estimated to have been 22 brake horsepower. From equations (8), (9) and (10), the charcoal consumption was calculated at about 8 kg/h.

Maximum diesel fuel saved in dual fuel operation in the laboratory was 80 percent at 1800 RPM and 40 brake horsepower. This condition could have been achieved in the field by throttling the pump discharge to prevent overloading the engine at the higher speed of 1800 RPM. At this condition, the injection fuel consumption rate would be 2.4 kg/h and the charcoal consumption about 22 kg/h. The operation, however, would be inefficient due to throttling the pump discharge and the higher frictional losses at higher speeds. It was decided, therefore, to operate the pump at 1200 RPM.

During the dry months of February to May, 1979, the engine was run continuously for 8 hours a day, two days a week on dual-fuel mode. Four days during the week, the engine was run on straight diesel. For precautionary reasons, the engine was not run in dual-fuel mode all the time, since no experience in maintenance during prolonged operation was yet available. Downtime caused by maintenance problems would have disrupted the rice planting season for the farmers in the area. No problems, however, were encountered during the four months and operations were stopped in June, 1979 only because the rainy season had arrived.

Gas Producer Performance

The performance data for the gas producer supplying gas to the multi-cylinder engine is summarized in Table 4. It is interesting to compare this to the performance of the smaller gas producer used with the single cylinder engine as indicated in Table 5. It is evident that the larger gas producer performed significantly better with an average thermal efficiency of 85 per cent and an average gas net calorific value of 128 Btu/nft^3 compared to 70 percent and 113 Btu/nft^3 for the small gas producer.

The reason for the improved performance of the bigger reactor can be partly explained by the higher specific gasification rate at the throat or combustion area (206 kg/h/m^2) compared to that of the smaller reactor (133 kg/h/m^2). Also, the total gasification rate of the larger reactor (20 kg/h) was 8 times that (2.5 kg/h) of the smaller reactor.

Other data on comparative performances are shown in Table 6. It is also noted that even though the calorific value of gas in the smaller reactor is less, the average percentage of energy input from producer gas (EPG) tends to be higher in the single cylinder engine (79 percent) than in the 6-cylinder engine (73 per cent). The probable reason for this was the better control of proportioning the gas-air mixture in a single cylinder engine than in a multiple cylinder engine.

Abstract

Gasification of various agricultural residues in down-draft, fixed bed gas producers and the utilization of the gas in small diesel engines converted for dual-fuel operation were studied at the College of Engineering, University of the Philippines. Such agricultural residues as coconut shells, wood waste, rice hulls and corn cobs were readily gasified in gas producers of simple design. Cleaning of the gas before its use in diesel engines presented some problems.

Use of charcoal in the gas producers to provide gas to a 5-brake horsepower single cylinder engine and a 65-brake horsepower six cylinder engine proved satisfactory. With charcoal as fuel, the percentage of the total energy from diesel oil replaced by producer gas and utilized in the single cylinder engine was higher (79 percent) compared to that in the six cylinder engine (73 percent). The thermal efficiency of the bigger gas producer, however was significantly better (85 percent) compared to the smaller gas producer (70 percent). The total gasification rate of the bigger reactor (20 kg/h) was 8 times that (2.5 kg/h) of the smaller reactor.

Appendix: Equations for Evaluating Engine and Gas Producer Performance

Brake Thermal Efficiency. The brake thermal efficiency, e_b, is calculated from the following equations:

$$e_b = \frac{(2545\ (BHP)}{(2.2)\,(13.76)\ F_1\ (HV_1) + F_2\ (2.2)\ (HV_2)} \tag{12}$$

$$F_1 = \frac{(69.3)\ (359)\ (2.2)\ W_1}{(12)\,(13.76)\,(\%\ CO_2 + \%\ CO + \%\ CH_4)} \tag{13}$$

Where

W_1 = Dry charcoal consumed in kg/h
F_1 = Producer gas consumed in kg/h
F_2 = Liquid fuel consumed in kg/h
BHP = brake horsepower output of engine
2545 = conversion factor of 1 BHP to British thermal unit (BTU) per hour
359 = Cubic feet of a mole of gas at NTP

2.2 = Conversion factor of 1 kg to pound (lb)

13.76 = Specific volume of producer gas in ft^3/lb at NTP

69.3 = % C in ultimate analysis of dry charcoal (69.3% C, 5.5% H, 22.2% O, 3.0% Ash)

% CO_2, % CO, % CH_4, % H_2 = percentage analysis of dry producer gas (volumetric)

HV_1 = 3.41 (% CO) + 3.43 (% H_2) + 10.67 (% CH_4) (higher heating value of dry producer gas at NTP in Btu/ft^3)

HV_2 = 19,494 Btu/lb (higher heating value of diesel fuel)

Substituting equation (13) and corresponding HV's into equation (12), the thermal efficiency when the engine is run on producer gas and diesel fuel becomes:

$$e_b = \frac{100 \text{ BHP}}{E_1 + E_2} \tag{14}$$

where

$$E_1 = W_1 \frac{6.17 (\% CO) + 6.15 (\% H_2) + 19.13 (\% CH_4)}{\% CO_2 + \% + \% CH_4} \tag{15}$$

and

$$E_2 = 16.85 \ F_2 \tag{16}$$

To correct for moisture content M (percent) of charcoal, multiply equation (15) (1 — M/100).

Percentage Energy from Producer Gas (EPG). In equation (14), the denominator represents the total energy supplied to the engine, allocated as follows: E_1 = energy from producer gas and E_2 = energy from the diesel fuel. Therefore,

$$EPG = \frac{100 \ E_1}{E_1 + E_2} \tag{17}$$

Cold Gas Thermal Efficiency (N_{th}). The cold gas thermal efficiency equation is given as follows:

$$N_{th} = \frac{\text{Net Calorific Value of Gas, } CV_n}{\text{Net Heat in Solid Fuel Used}} \times 100$$

CV_n = 3.18 (% CO) + 2.70 (% H_2) + 8.95 (% CH_4) (in Btu/ft^3 at 60°F and saturated with moisture)

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

For dry charcoal, the net heating value $HV_n = 12{,}510$ Btu/lb and percentage by weight of carbon, % C = 69.3.

Substituting in the cold gas efficiency equation:

$$N_{th} = \frac{55.64\ (\%\ CO) + 47.24\ (\%\ H_2) + 156.59\ (\%\ CH_4)}{\%\ CO_2 + \%CO + \%\ CH_4} \qquad (18)$$

To correct for moisture content. M (percent) of charcoal, divide equation (18) by (1 — M/100).

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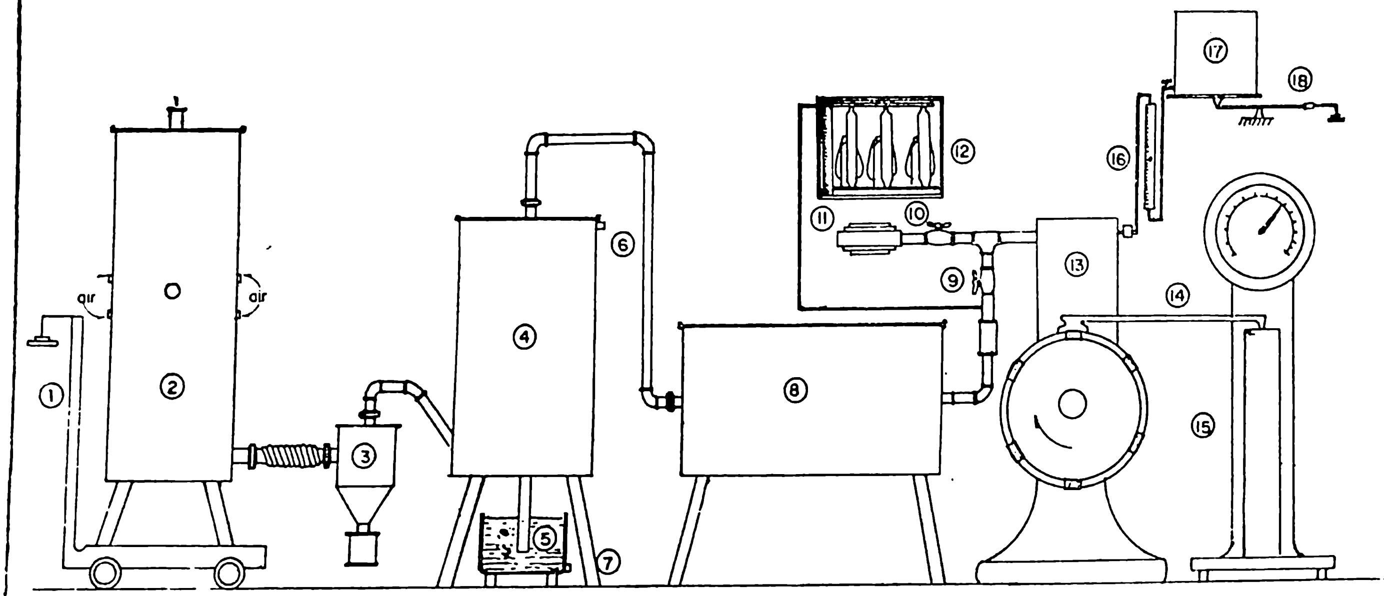

Figure 1: EXPERIMENTAL LAYOUT FOR STUDYING USE OF PRODUCER GAS FROM SOLID WASTE FUELS IN A DIESEL ENGINE

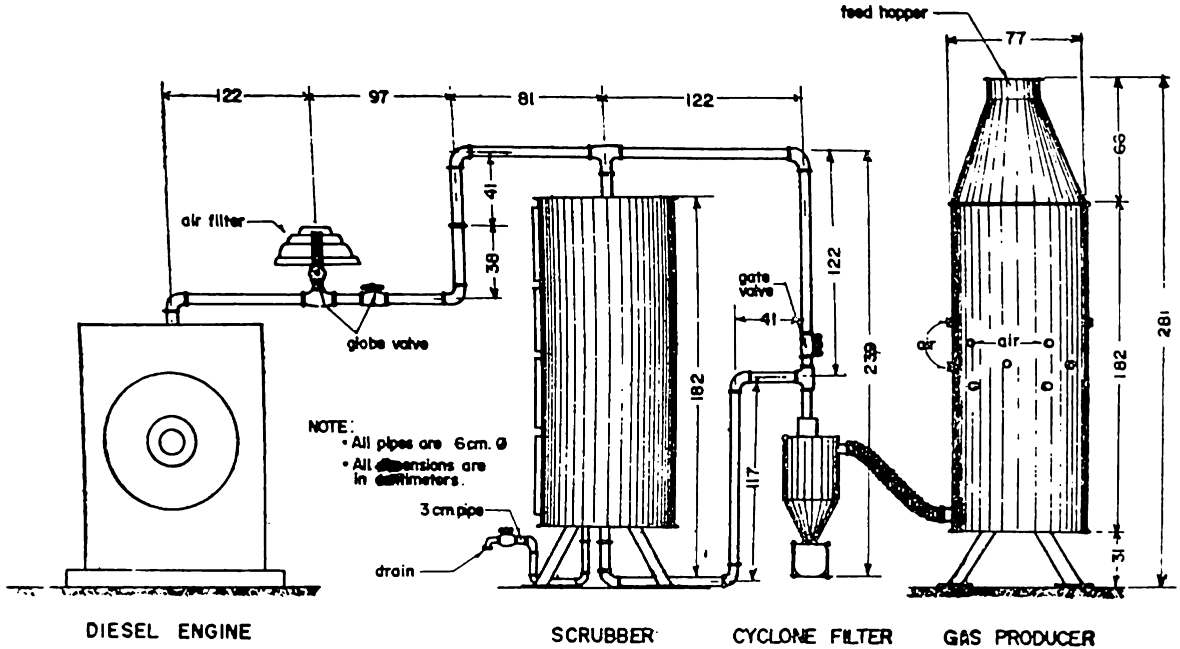

Figure 2: MULTI-CYLINDER DUAL FUEL ENGINE SET-UP

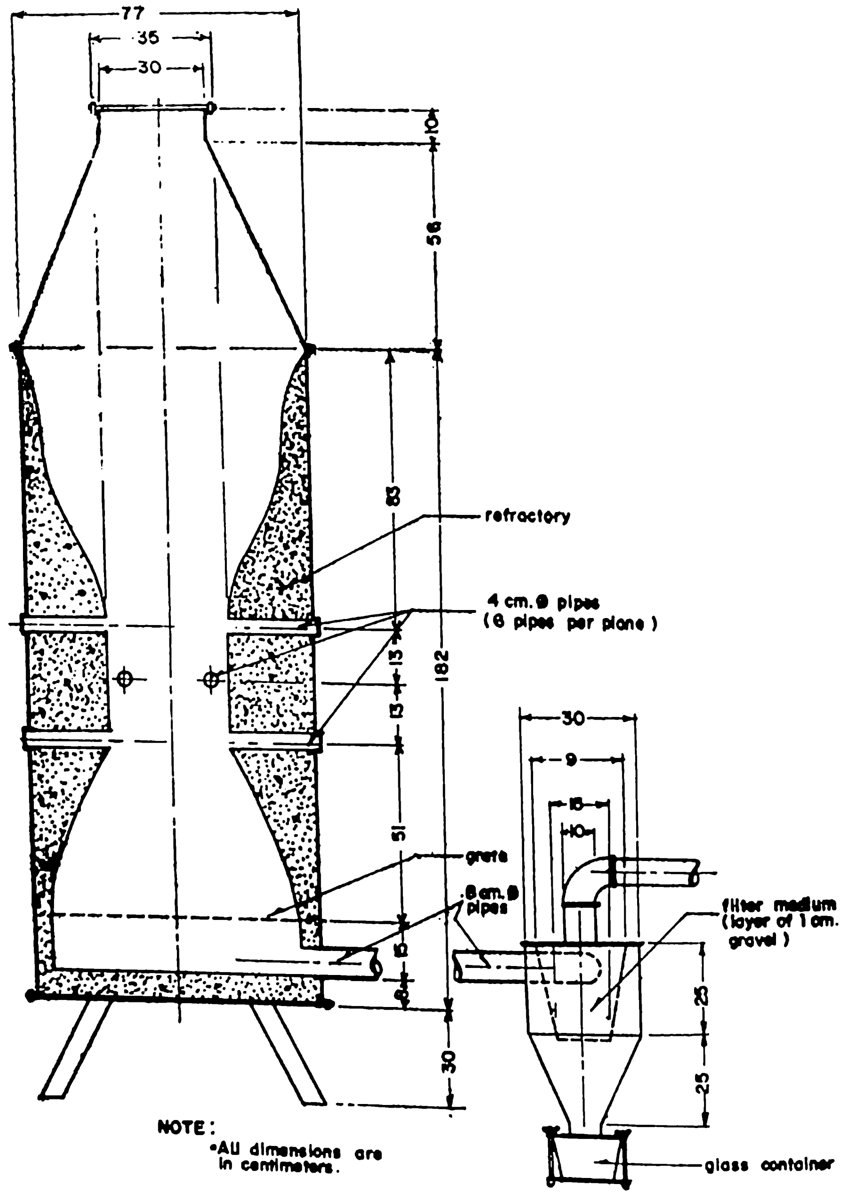

Figure 3: CROSS-SECTION OF DOWN-DRAFT SUCTION-TYPE GAS PRODUCER