BIOMASS PRODUCER GAS FUELING OF SPARK IGNITION ENGINES

by

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INTRODUCTION

Included in the search for substitutes for petroleum-based fuels have been agricultural and forestry residues and urban wastes. In the case of agricultural and forest residues, four primary methods of energy recovery are being studied by researchers: direct combustion, anaerobic digestion, gasification, and fermentation. The latter three methods are capable of supplying fuel for internal combustion engines, either in liquid or gaseous form.

It is well known that internal combustion engines are one of the main sources of power for irrigation pumps and sprinkler systems for a large number of farmers in the United States. Hay (1978) reports that in Kansas alone, about 16,000 pumping units are fueled with natural and LP gas, 65% of the Kansas total. Another 14% are fueled with diesel fuel. In terms of total energy usage, in Texas and Kansas, respectively, 83 and 81 percent of the energy used for irrigation is from natural gas (Economic Research Service, 1977).

While any of the four options of energy recovery from crop residues mentioned above may be most appropriate for a given situation, Clark et al. (1978) proposed that gasification holds the greatest potential for alleviating natural gas curtailments to irrigators in the Great Plains States.

Griffen (1944), Williams et al. (1978), and others have demonstrated the performance characteristics of a duel-fueled, compression ignition engine operating on producer gas and pilot injection of diesel fuel. While compression ignition applications can not be ignored because of their high cycle efficiency, neither can the fact pass unnoticed that so many power units already in the field are spark ignition engines. This study then, will focus on the performance of a spark ignition engine fueled by gas from a fluidized bed gasifier. Engine performance with natural gas as a fuel will be used for comparison.

REVIEW OF LITERATURE

Gasification

The use of producer gas in internal combustion engines is not new. Horsfield and Williams (1978) report that an estimated 700,000 vehicles in Europe, Australia, South America, and the Pacific Islands were converted to producer gas during World War II. These gasifiers were primarily of the updraft, crossdraft, and downdraft types. Additional histories of early gasifiers are provided by the Solar Energy Research Institute (1979) and Nowakowska and Wiebe (1945).

Groves et al. (1979) report that valuable characteristics of fluidized bed gasifiers include: high rate of heat transfer from bed material to feedstock, maintenance of an isothermal bed temperature, accurate control of bed temperature, efficient conversion reactions, wide variation in type, composition, and moisture content of acceptable feedstocks, and capability of continuous feeding. Construction details, theory of operation, and performance data of the Kansas State University fluidized bed gasifier have been documented by Ramen et al. (1980), Walawender et al. (1980), and Walawender and Fan (1978). Data are included for the gasification of feedlot manure, corn stover, cane, sewage sludge, and tire rubber.

Engine Power Output

Tatom et al. (1976) reported fueling a General Motors truck engine with a simulated pyrolysis gas consisting of 12% hydrogen, 24% carbon monoxide, 7% methane, and 57% nitrogen. Engine power output was 60-65% of that attained with gasoline. Burstall and Woods (1939) report that engine power dropped to 66% of gasoline power output when operating with a charcoal derived gas, while Spiers (1942) reported maximum power developed with producer gas was only 53% of gasoline power output. The decrease in power levels is due mainly to the lower heating value of the air-producer gas mixture compared to an air-gasoline mixture (SERI, 1979); also, the number of molecules present in the combustion chamber increases during combustion for a gasoline-air mixture, remains constant for a methane-air mixture, and decreases for a producer gas-air mixture, assuming complete combustion. These two factors decrease peak combustion pressure and hence, mean effective pressure. Most researchers reported some power loss due to a drop in volumetric efficiency. One reason for this is the absence of evaporative cooling which occurs in a gasoline-fueled engine. Also, there was a manifold pressure depression caused by the engine having to pull air into the gasifier for reaction. In fact, early gasifiers were sometimes referred to as "suction gas generators". This effect will not be experienced when using gas from a fluidized bed gasifier.

Heywood (1941) theorized that supercharging the inlet mixture to 141. kPa (20.5 psia) would result in gasoline power output. Branders (1941) found that a boost pressure of 41.4 kPa (6 psig) compensated for power losses due to the lower energy density of the producer gas.

Air-Fuel Mixture Strength and Flammability Limits

Spiers (1942) found that maximum engine power using producer gas occurred with a lean mixture—about 93.5% of the stoichiometric mixture strength for engine speeds of 1000-3000 RPM. Woods (1940) reported maximum power on producer gas occurred with mixtures 2-3% lean for a wide range of compression ratios. Accounting for dissociation of combustion products, primarily CO_2 and H_2O dissociations, Woods (1940) predicted that maximum power with a producer gas fueled engine should occur with a 2% rich mixture. Including nitric oxide formation in his calculations gave Woods the same value of maximum power mixture strength.

While the limits of flammability of simple mixtures of a single combustible gas and air have been documented with relative consistency by Coward and Jones (1952), Matheson (1966), and others, the flammability limits for mixtures of more than one combustible gas are less definite. Le Chatelier (1871) proposed an additive effect method for computing lower flammability limits of combustible mixtures. Coward and Jones (1952) demonstrate the method applied to both upper and lower limits of flammability and discuss the effects of humidity, pressure, temperature, and turbulence. Coward et al. (1919, 1926) verified the principle for mixtures of hydrogen, carbon monoxide, simple paraffins, and natural gas.

Ignition Timing and Flame Speed

Ignition timing for optimum torque is primarily a factor of the mixture flame speed and engine speed. Le Chatelier's mixture law calculates flammability limits, and it may also be used to indicate flame speed. In a relative sense, flame speed is minimum at the limits of flammability and is maximum near correct mixture strength. Optimum ignition advance, being inversely related to flame speed, should then be minimized near stoichiometric mixtures. Tatom et al. (1976) report spark timing for best torque as 30, 35, and 40 degrees before top dead center (^{o}BTDC) for engine speeds of 1500, 2000, and 2500 RPM, respectively. Fuel-air mixture strength was also adjusted for best torque, but was not measured.

Optimum ignition timing has been shown by Marias (1936), Spiers (1942), and Rammler et al. (1938) to decrease with increasing fuel hydrogen content. One understands the relative effect of hydrogen content on flame speed when realizing that the maximum flame speed for hydrogen in air is approximately 2.7 m/s, while methane and carbon monixide have maximum flame speeds of approximately 0.3 m/s.

Other fuel constituents will also have an effect on flame speed and ignition timing. Gumz (1942) presented a triangular coordinate chart for estimating flame speeds of mixtures of hydrogen, methane, and carbon monoxide. Nitrogen and carbon dioxide dilutions lower the estimated maximum flame velocity according to equation (1).

$$V = V_o \left[1 - \frac{N_2\ (\%) + 1.67\ CO_2\ (\%)}{100} \right] \qquad (1)$$

The rate of reaction is increased by higher temperatures (high particle energy), and higher density resulting in an increased number of particle collisions (Segeler, 1932). Obert (1973) and Edwards and Teague (1970) elaborate on combustion theory in spark ignition engines.

Moore and Roy (1956) reported that inlet mixture heating for very lean mixtures of propane reduces optimum ignition advance. Moore and Roy also found that indicated efficiency dropped and exhaust methane content increased for very lean mixtures in a methane fueled engine due to lower flame speeds. For rapidly burning mixtures, exhaust methane content was low.

Knocking is not a problem with producer gas at high ignition advances with compression ratios typical of gasoline fueled engines. SERI (1979) reports a producer gas octane number of 105. Spiers (1942) found that engine power dropped as timing was increased beyond the optimum value, but knocking was not audibly detected with compression ratios of 6:1 and 7:1.

Gas Cleaning

In addition to power losses, researchers also report gas clean-up as one of the most troublesome aspects of producer gas usage in engines. Many schemes have been tested, none of them being entirely successful. Bowden et al. (1942) tested a succession of two "windmill type" impingement cleaners, an oil bath, and a sisal hemp drier. Also used by Bowden was the combination of a centrifugal dry cleaner, a horizontal oil-contact cleaner, and a hemp drier. Spiers (1942) found oiled coke to be unsatisfactory. He also tested a sisal tow packed "milk churn" type filter.

The amount and type of cleaning necessary is affected not only by the type of gasifier, but also by the gasification feedstock used. It is well known that updraft gasifiers yield significantly more tar mist than downdraft types. SERI (1979) reports that updraft gasifiers yield a low tar product with fuels such as coke, anthracite, and possibly charcoal. When these types of fuels are used in downdraft gasifiers, tar production is even lower. So, the principal cleaning problem faced by early investigators was one of ash removal. When the gasification feedstock is wood, crop residue, or other biomass, the tar content of the gas is much higher.

Supercharging

Zinner (1978) gives four reasons for the increase in engine efficiency when exhaust turbocharging is utilized. Two of those reasons applicable to turbocharged spark ignition engines without charge cooling are: 1) Engine friction losses increase less rapidly than engine mean effective pressure, and 2) In four stroke engines there is a smaller negative loop work (intake and exhaust process), or possibly even a positive loop work.

INVESTIGATION

Objectives and Scope

The objectives of this research are as follows:

1. Develop an engine fueling system to utilize low energy gas produced from crop residues in a fluidized bed gasifier.

2. Evaluate engine performance when fueled with pipeline quality natural gas to provide a standard of comparison.

3. Evaluate engine performance when fueled with low energy gas produced from crop residues in a fluidized bed gasifier.

4. Determine whether supercharging the low energy gas and air mixture can be effectively used to restore engine power to natural gas power levels.

Gaseous fuel carburetors for natural gas engines meter the fuel so that 9 - 10% of the inducted air fuel mixture is fuel. These carburetors are suitable for fuel with an energy content of 30 - 37 MJ/m^3 (800 - 1000 BTU/ft^3). Other carburetors are commercially available for fuels with energy contents as low as 19 - 22 MJ/m^3 (500 - 600 BTU/ft^3), heating values typical of poor quality natural gas or gas from an anaerobic digester. Because air blown gasifiers produce a dilute fuel with heating values as low as 3.7 to 7.5 MJ/m^3 (100-200 BTU/ft^3), the carburetor must be capable of handling large volumes of fuel. The proportion of fuel to total mixture volume will usually lie in the range of 30 - 50%. Carburetors of this type are not commercially available.

In addition to handling high flow rates, the engine fueling system from a fluidized bed gasifier must provide a steady, constant pressure flow of fuel. The output from a fluidized bed is characterized by an unsteady, pulsating flow. This is caused by pockets of the fluidizing gas rising through and erupting at the surface of the bed. This inherent quality of fluidized bed gasification makes gas carburetion more difficult.

Experimental Equipment and Procedures

Gasifier

The pilot plant gasifier supplying the fuel used in this work is shown in Figure 1. The gasifier equipment may be divided into seven main components: 1) the reactor, 2) feed hopper and screw feeder, 3) high temperature cyclone for particulate removal, 4) a venturi scrubber, 5) excess fuel afterburner, 6) controls, and 7) gas sampling equipment.

The reactor was a .23 m (9 in.) I.D. cylinder constructed of 310 stainless steel with a normal operating range of 800 to 1100 K. The combustion of propane under starving air conditions in the plenum supplied the necessary heat for gasification and the fluidizing gas. Additional fluidizing gas was supplied by water sprayed into the plenum which also served to keep the plenum temperature below 1200 K. A perforated plate acted as a gas distributor. Supplemental heat was supplied by a radiant burner surrounding the bed when needed.

The feedstock was fed to the fluidized bed by a horizontal auger located under the feed hopper. The auger moved the material to a

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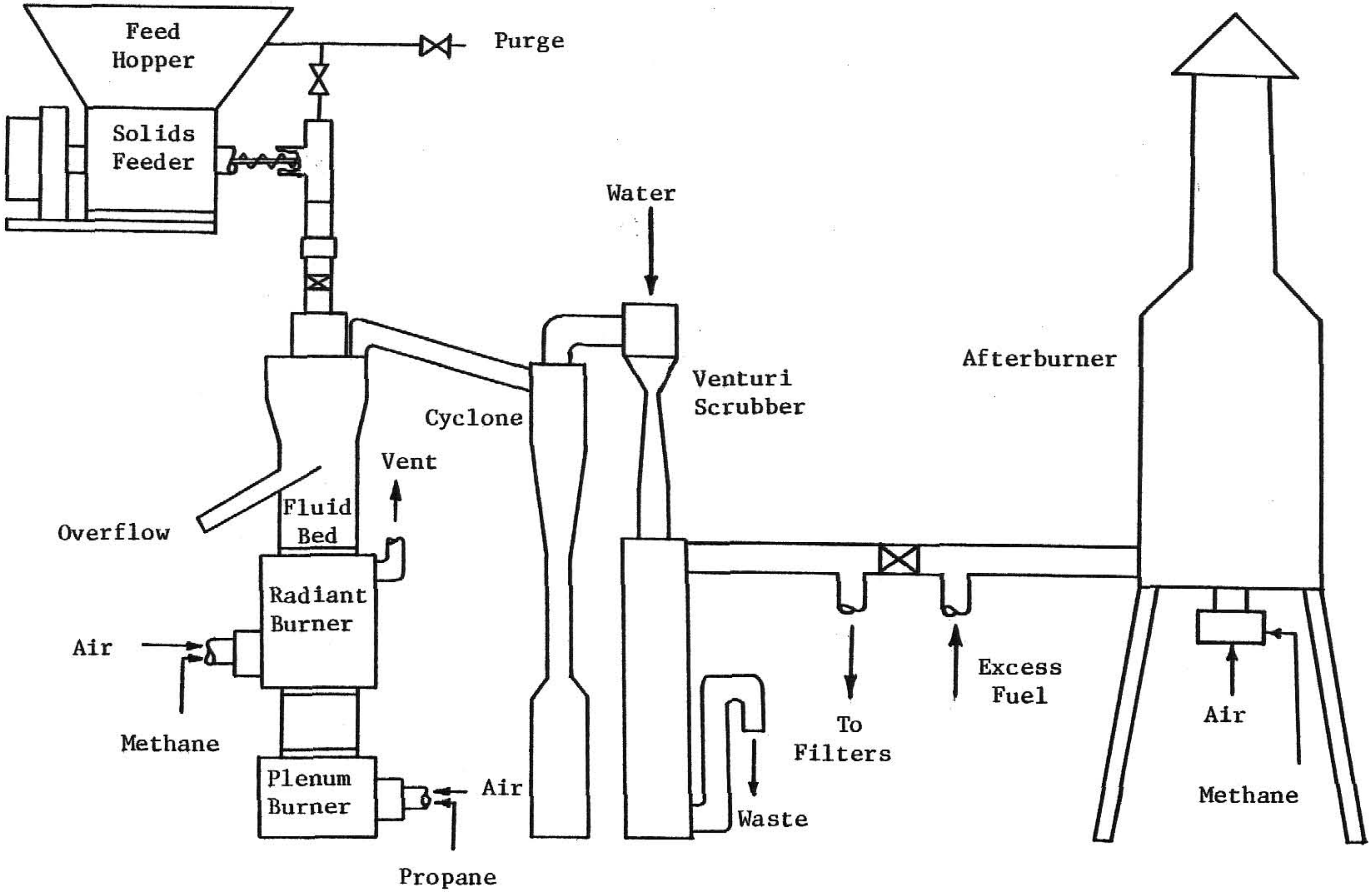

Figure 1. Schematic Diagram of Pilot Plant Gasifier.

vertical feedpipe. Gravity feed was used to deliver the feed from the auger to the bed surface. A helium purge on the feed hopper was necessary to prevent reactor gases from rising into the feed system. Condensation of vapors in these gases frequently caused feed pipe plugging because of an inadequate seal on the hopper which allowed the vapors to rise into the feedpipe.

The reactor bed contained about 46 kg of silica sand with a size range of -14 +50 mesh. When the corn stover was fed to the bed, char and volatiles were formed. Superficial velocities were maintained in the range of 27 to 46 cm/s so that the sand remained in the bed, but most of the char was entrained in the off gas.

The volatiles produced in the reaction passed upward through a disengaging zone and then on to a cyclone which removed most of the char particles. From there, the volatiles were sent to a venturi water scrubber followed by a series of packed bed filters, and then on to the engine.

Instrumentation included flow meters for air, propane, and injection water and a twelve point thermocouple temperature recorder. A Packard Model 417 Becker Gas Chromatagraph was used for gas analysis. A Perkin-Elmer 240B Elemental Analyzer was used for ultimate analysis of the feed and char.

Feedstock Description

Corn stover was used as a feedstock material for the gasifier. The ultimate analysis of the corn stover was C, 44%; H, 5.7%; O, 38.3%; N, 6.0%, and Ash 6.0%. The corn stover was hammer milled to pass through a 0.64 cm (1/4 inch) screen. It was dried with forced air to a moisture content of 7.1%.

Engine and Fuel System Description

The engine used in this study was a Continental R800-46 four-stroke, four cylinder engine with a displacement of 0.846 L (51.6 in^3) and a maximum rated speed of 5000 RPM. The engine is manufactured by Renault of France. This model was chosen because its power output matched the fuel output potential of the gasifier. The gasoline carburetor, fuel pump, governor and fan were removed for the tests. Cooling was provided by a water-to-water heat exchanger.

Initial engine trials using an Impco brand natural gas carburetor to meter the producer gas were unsuccessful because of the restricted fuel flow passage. A second series of tests using an Impco digester gas carburetor also failed for the same reason. A sleeve was machined and inserted in the air passage of the digester gas carburetor in an attempt to obtain a larger volume of flow through the gas valve. This, too, failed. A combustible mixture was finally obtained with a manually controlled carburetor assembled from pipe valves and fittings. The air flow passage was a 5.08 cm (2 in.) diameter pipe and the fuel line was a 2.54 cm (1 in.) diameter pipe. These pipe sizes, although larger than necessary, were chosen to minimize pressure drops. Air flow was controlled with a ball valve, and fuel flow was controlled with the butterfly valve removed from the choke assembly of the original gasoline carburetor. These two valves could be manipulated to change the air-fuel mixture strength and to provide a throttling effect. A third valve controlled natural gas flow for comparison tests. The fuel delivery system is shown schematically in Figure 2.

A variable volume, constant pressure fuel storage tank was constructed to supply a steady, positive pressure flow of fuel to the fuel metering valve. The storage tank was made from two diesel fuel storage tanks with diameters of .965 m (38 in.) and .914 m (36 in.) and a length of about 1.5 m (5 ft.). One end was removed from each tank, and the smaller tank was inverted inside of the larger tank. The unit was filled with water to provide a seal. When the delivery system pressure exceeded 1.0 kPa (4 in. H_2O), the inner tank lifted to accomodate 0.6 m^3 (21 ft^3) of fuel. After filling was completed, the gas pressure increased to 1.5 kPa (6 in. H_2O). At this point, excess gas escaped through a water trap to an afterburner. In order to simulate turbocharged conditions, a General Motors 2-53 Roots type positive displacement blower was used to boost intake manifold pressure. Control over manifold pressure at all times was maintained by driving the blower independently of the engine with an electric motor and a variable speed v-belt drive. A bypass valve connecting the blower inlet and surge tank was opened during tests under naturally aspirated conditions.

A variety of schemes have been used for gas cleaning. A cyclone and venturi-type water scrubber have been in place for all tests. In addition to these components, combinations of packed columns of various cross-sectional areas containing glass wool, steel wool, and furnace filters have been tried. Tests have also been run with a helical path centrifugal cleaner. Because the venturi scrubber will tolerate a

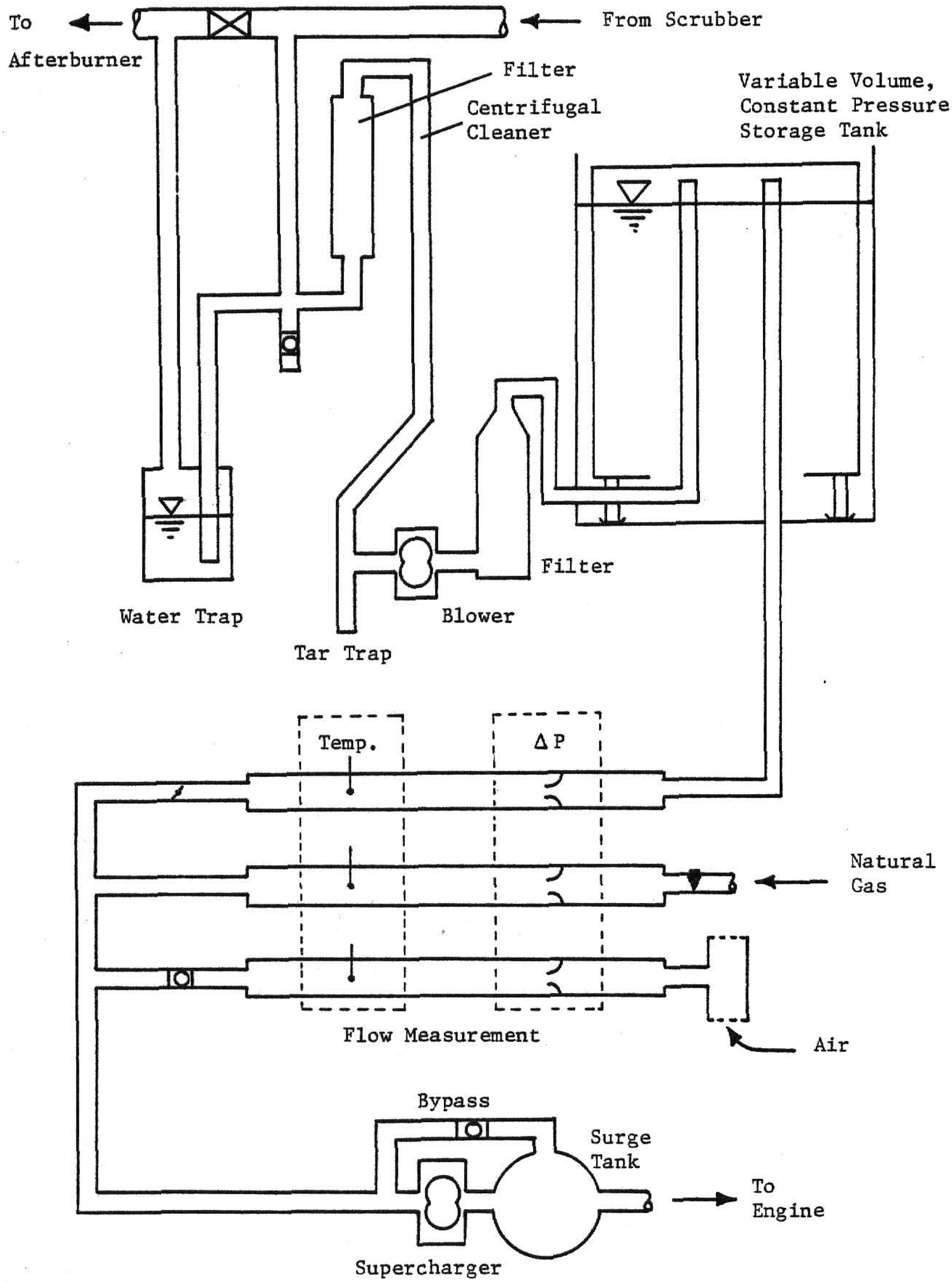

Figure 2. Schematic Diagram of Air-Fuel Measurement and Delivery System.

maximum pressure of approximately 4.0 kPa (16 in. H_2O), a smaller Roots blower has been installed to overcome pressure drops in the fuel filtering system.

Dynamometer and Instrumentation

The engine was loaded with a Mid-West Dynamometer Model 1014A eddy current dynamometer. The dynamometer, modified with a Control Engineering CE229 controller, can be operated in either a constant speed or constant torque mode. While operating in the constant speed mode for these tests, the dynamometer also acted as a governor for the engine. A dual-bridge load cell, Transducers Inc. Model T63H-200 was inserted into the linkage of the dynamometer scale so that both electronic and visual indication of engine torque would be provided. The load cell signal is received by a Daytronic 3270 strain gage conditioner and indicator. Engine speed was determined by a 60-tooth gear, an Electro 3010AN magnetic pick-up, and a Daytronic 3240 frequency conditioner and indicator.

Air and fuel temperatures, inlet mixture temperature, water jacket temperatures, and pan oil temperature were measured with copper-constantan thermocouples and a Hy Cal Model 300 reference cell. Exhaust temperature was measured with an iron-constantan thermocouple and an ice-water reference. Air and fuel flow were measured with long radius flow nozzles and Setra Systems Inc. Model 239 variable capacitance differential pressure transducers. The differential pressure transducers were calibrated against a Meriam Model 34FB2 micromanometer. The discharge coefficients for the flow nozzles were determined using a discharge coefficient-Reynold's number relationship proposed by Benedict

(1966). Absolute air, fuel, and inlet manifold pressures were measured with Setra Systems Inc. Model 204 variable capacitance absolute pressure transducers. All electronic data signals were recorded by a Digital Model 1268 data logger. Atmospheric wet-bulb and dry-bulb temperatures were measured with a sling psychrometer. Engine power was corrected to standard atmospheric conditions according to SAE procedures (SAE Handbook, 1978).

Engine Testing Procedure

The engine test sequence was divided into three phases. The first phase of testing involved the determination of engine performance and behavior when fueled with natural gas under naturally aspirated conditions. After the engine was warmed up and the instruments were calibrated, the air valve was opened fully to simulate full throttle conditions. The natural gas valve was then set at a position to provide a particular air-fuel mixture strength. Next, the ignition spark advance was manually adjusted to find the minimum advance for best engine torque. The spark timing was recorded; after engine temperatures had stabilized, all other data points were recorded with the data logger. This process was continued until data had been collected for several air-fuel mixture strength values ranging from very lean to very rich. The dynamometer was operated in the constant speed mode. Natural gas data was collected at engine speeds of 1800, 2200, 2600, and 3000 RPM.

The second phase of testing was identical to the first, except that the fuel was from gasified corn stover. Tests for naturally aspirated producer gas fueling were performed at 2200 and 2600 RPM. The third phase of testing used gas from corn stover while operating under supercharged conditions. Preliminary tests indicated that a manifold pressure of about 1.4 times atmospheric pressure was required to achieve natural gas power levels. Tests were performed at 2200 RPM with manifold pressures of 1.25, 1.3, and 1.4 times atmospheric pressure. Because of a malfunction in the fuel gas analysis equipment, the results at the 1.25 pressure level were discarded. During testing, the variable speed drive was adjusted so that the boost pressure was slightly higher than the desired level. The bypass valve was used to fine tune the boost level.

Results

Fuel Analysis

A typical analysis of the fuels during testing is shown in Table 1. The composition of the natural gas remained relatively constant with a lower heating value of 31.7 MJ/m^3 (852 BTU/ft^3). Because of the many variables governing the behavior of the gasifier, it was impossible to obtain producer gas of constant quality for the duration of a test session.

Table 1. Average Fuel Composition

Component	Natural Gas Vol. %	Producer Gas Old Stover Vol. %	Producer Gas Old Stover Std. Dev.	Producer Gas New Stover Vol. %	Producer Gas New Stover Std. Dev.
Hydrogen		11.4	1.7	10.5	0.5
Nitrogen	12.2	52.0	2.8	56.2	1.2
Methane	76.5	4.3	0.7	3.7	0.2
Carbon Monoxide		13.0	1.7	11.1	0.5
Carbon Dioxide		15.5	1.2	15.7	0.4
Ethylene		1.4	0.2	1.2	<0.1
Ethane	7.4	0.3	<0.1	0.2	<0.1
Propylene		0.4	0.1	0.3	<0.1
Propane	3.8				

Three important variables in this group are bed temperature, air injection rate, and feed rate. With careful operation, bed temperature and air injection rate could be maintained fairly constant during a daily sequence and over a weekly or monthly collection of single-day runs. The biomass feed rate, on the other hand, was quite difficult to control. The first batch of corn stover processed (designated as "old stover") was collected with a large stackmaker. Because of cool, damp weather, the moisture content was higher than desirable. For storage the stover was run through a tub grinder and kept in plastic trash bags. Before use, the stover was more finely ground in a cutterbar hammer mill with a one-half inch screen. Due to the moisture content of the stover and the method of feeding the hammer mill, the composition of the stover from the hammer mill was not consistent. For that reason, the feed rate was not always the same for a given feeder control setting even though the stover was dried prior to gasification. To complicate this problem, difficulties were experienced in maintaining the helium purge on the feed hopper. When purge pressure was lost, hot gases from the bed rose into the vertical feed pipe and the horizontal auger section. As the hot gases encountered the relatively cold metal and corn stover, the vapors condensed and caused the feed pipe to plug. When this occurred, the nitrogen content of the off gas increased, resulting in a decreased heating value. Sometimes this was a gradual process, while at other times it happened suddenly. The composition of the gases produced from the stover as recorded in Table 1 are averages computed from gas analyses taken over the duration of the test period.

The second batch of corn stover processed (designated as "new stover"), because of a warm dry fall, was more completely dried in the field. The stover was cut with a sickle mower, raked into windrows, and baled with a conventional square baler. The bales were stored under cover. For processing, the bales were fed directly into the hammer mill.

Because the stover moisture content was much lower than the first batch of stover processed, the resulting grind was finer and more consistent using the same hammer mill screen size. In addition, the feed hopper was modified so that the purge was more reliable, and the feed pipe plugging was eliminated. The combination of these two factors yielded a much more consistent producer gas composition as shown in Table 1. The use of the new stover as a gasification feedstock roughly corresponds to the period of supercharged engine trials. R-squared values for fitted equations describing all modes of engine operation listed in Appendix B, Table 7 give an indication of the effect of the changing gas composition on engine performance.

The average fuel lower heating values as calculated from the gas analyses are recorded in Table 2. The reduction of variation in the producer gas composition when using the new stover as the gasification feedstock is clearly reflected in the reduction of variation in the energy content of the fuels. Some characteristics of the fuels are listed in Table 3. These values were calculated using the values of average fuel composition.

Table 2. Average Calculated Fuel Lower Heating Values

Fuel	MJ/m^3	(BTU/ft^3)	Standard Deviation
Natural Gas	31.7	(852)	
Producer Gas (Old Stover)	5.12	(138)	0.57 (15.0)
Producer Gas (New Stover)	4.46	(120)	0.20 (5.0)

Table 3. Characteristics of Fuels Listed in Table 1.

	Calculated Flammability Limits (% in Air)	Stoichiometric Mixture Fuel Content (%)	Change in Number of Moles for Stoichiometric Combustion (%)
Natural Gas	5.1 - 15.8	9.6	+0.7
Producer Gas (Old Stover)	18.9 - 59.9	42.6	-5.1
Producer Gas (New Stover)	22.1-61.4	46.6	-5.0

The results of this work are being presented as a function of the fuel-air equivalence ratio, defined as the ratio of the actual fuel-air ratio to the chemically balanced fuel-air ratio. So relatively speaking, values greater than one represent rich mixtures, and values less than one represent lean mixtures.

Figure 3 is a family of curves showing the relationship between the fuel-air equivalence ratio and the fraction of the inducted air-fuel mixture that is occupied by fuel. Each curve represents a fuel with a particular lower heating value. Only data for air-producer gas mixtures are presented.

The heating values of the air-fuel mixture as a function of mixture strength for the different fuels are displayed in Figure 4. Included for each fuel are least-square linear regression equations and correlation coefficients.

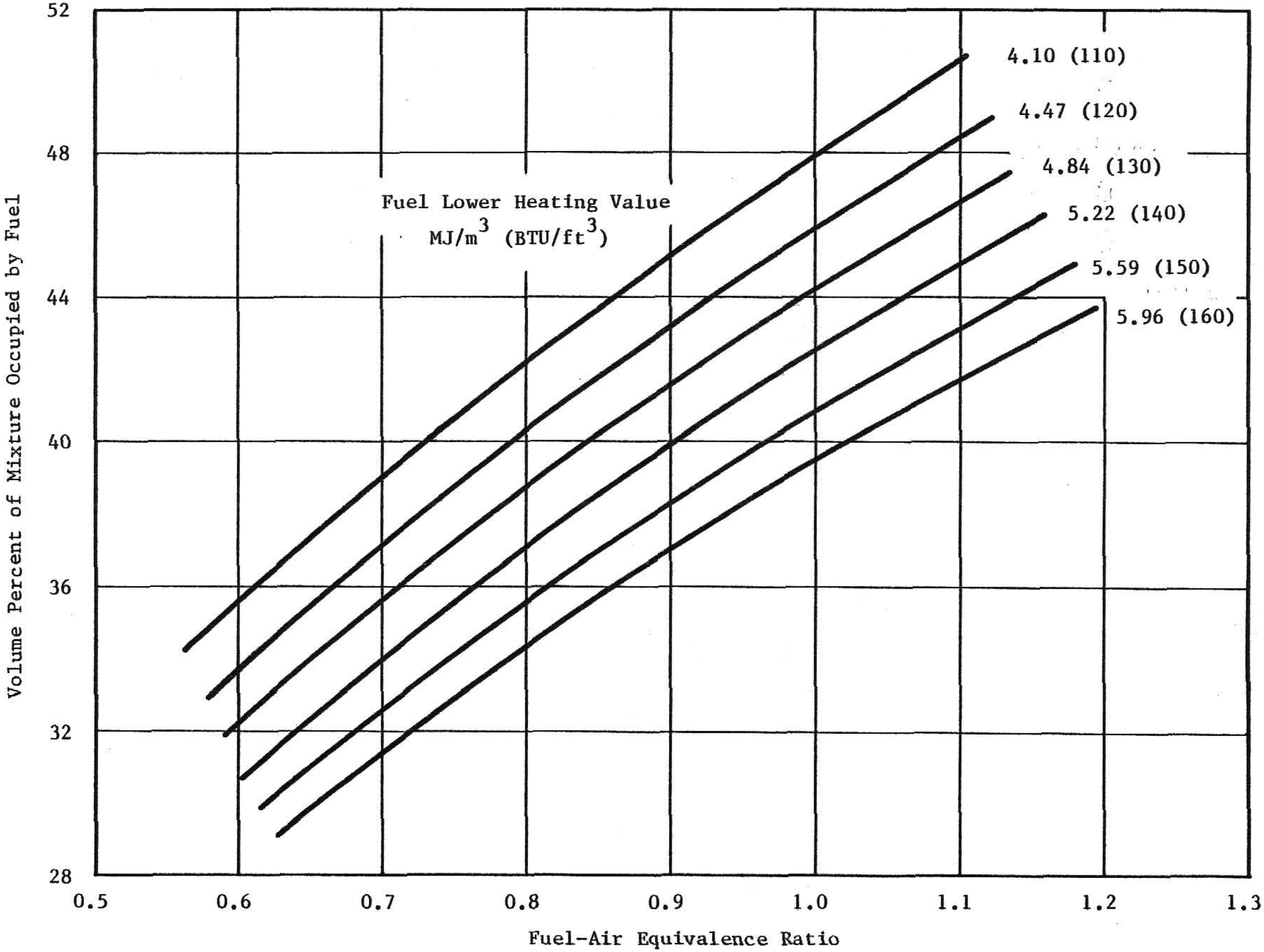

Figure 3. Amount of Mixture Occupied by Fuel for Fuels of Various Heating Values.

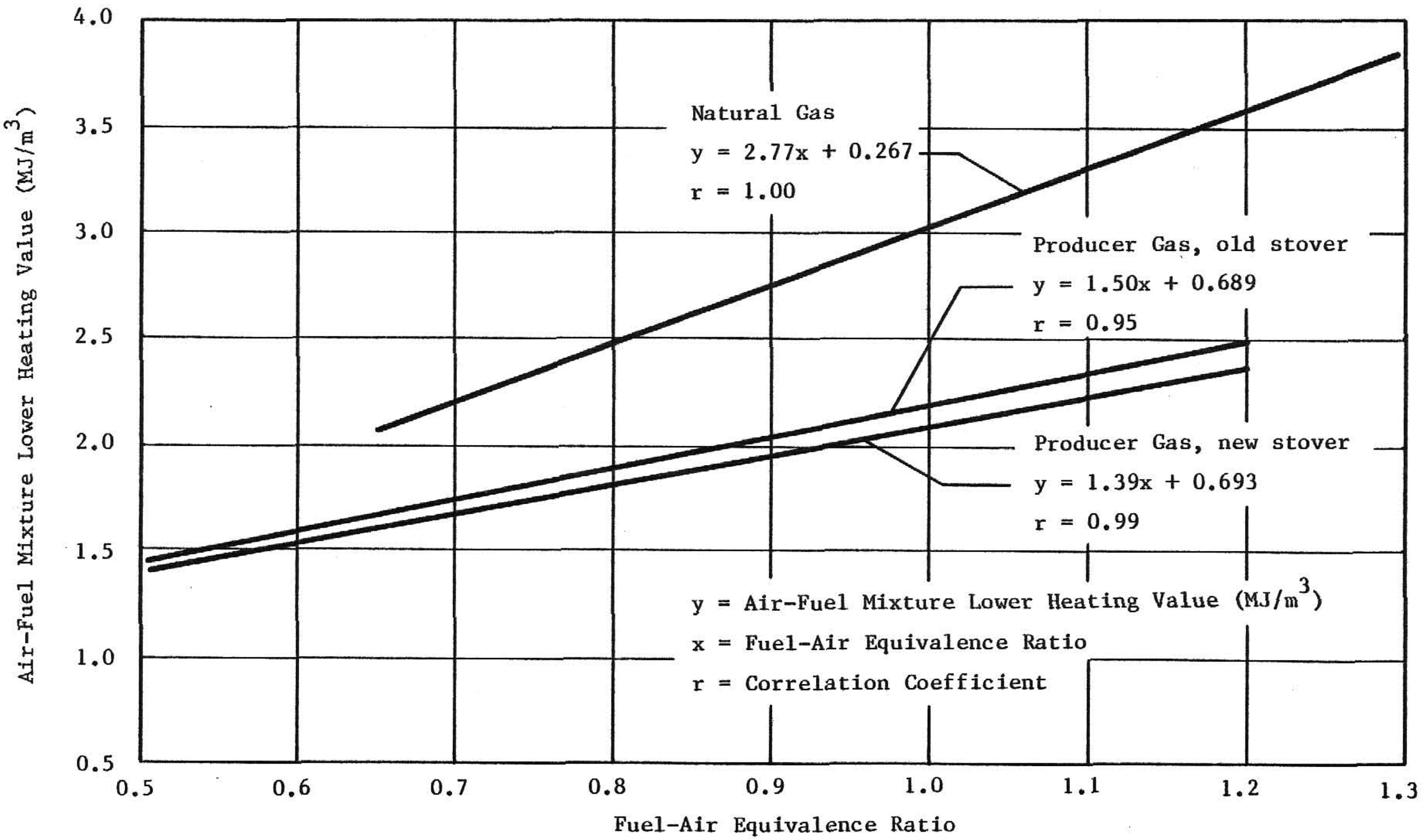

Figure 4. Mixture Heating Value as a Function of Equivalence Ratio for Various Fuels.

Engine Performance

Test results including corrected brake power, brake thermal efficiency, exhaust temperature, and minimum ignition timing for maximum torque are plotted in Appendix A, Figures 14 through 23. Because of the obvious consistency of the natural gas data presented in the figures, the accompanying curves were drawn without the use of statistical procedures. The amount of scatter in the producer gas data dictated that equations be fit to the data to aid in graphical presentation of the results. Quadratic equations describing each of the above four parameters for producer gas operation along with R-squared values are presented in Appendix B. Numerical values describing producer gas operation in the remainder of the text were generated with the use of the quadratic equations.

DISCUSSION

Naturally Aspirated Tests at 2600 RPM

Power and Efficiency

Naturally aspirated power and brake thermal efficiency at 2600 RPM are displayed in Figure 5. Maximum natural gas power was 12.4 kW at an equivalence ratio of 1.025. Maximum producer gas power was 8.1 kW at a lean equivalence ratio of 0.931, which agrees with the findings of Spiers (1942). Therefore, the producer gas to natural gas power ratio at 2600 RPM is 0.653. By inspection of Figure 4, we see that the ratio of air-fuel mixture lower heating values at the corresponding maximum power equivalence ratios is 0.671 (using the old stover producer gas curve from Figure 4). So, the loss in power output during producer gas operation closely corresponds to the reduction in the lower heating value of the inducted air-fuel mixture.

The engine exhibited lower brake thermal efficiencies when fueled with producer gas compared to the efficiencies achieved during natural gas operation. Brake thermal efficiency is defined as the ratio of useful mechanical energy output to the chemical energy inducted in the air fuel mixture calculated with the lower heating value of the fuel. At the points of maximum power, producer gas brake thermal efficiency was 26.4%, and natural gas had an efficiency of 28.4%. This could be due to the slower flame travel in the combustion chamber. Because of the decreased flame speed, the actual process deviates more from the theoretical Otto air standard cycle which has an instantaneous, constant volume heat addi-

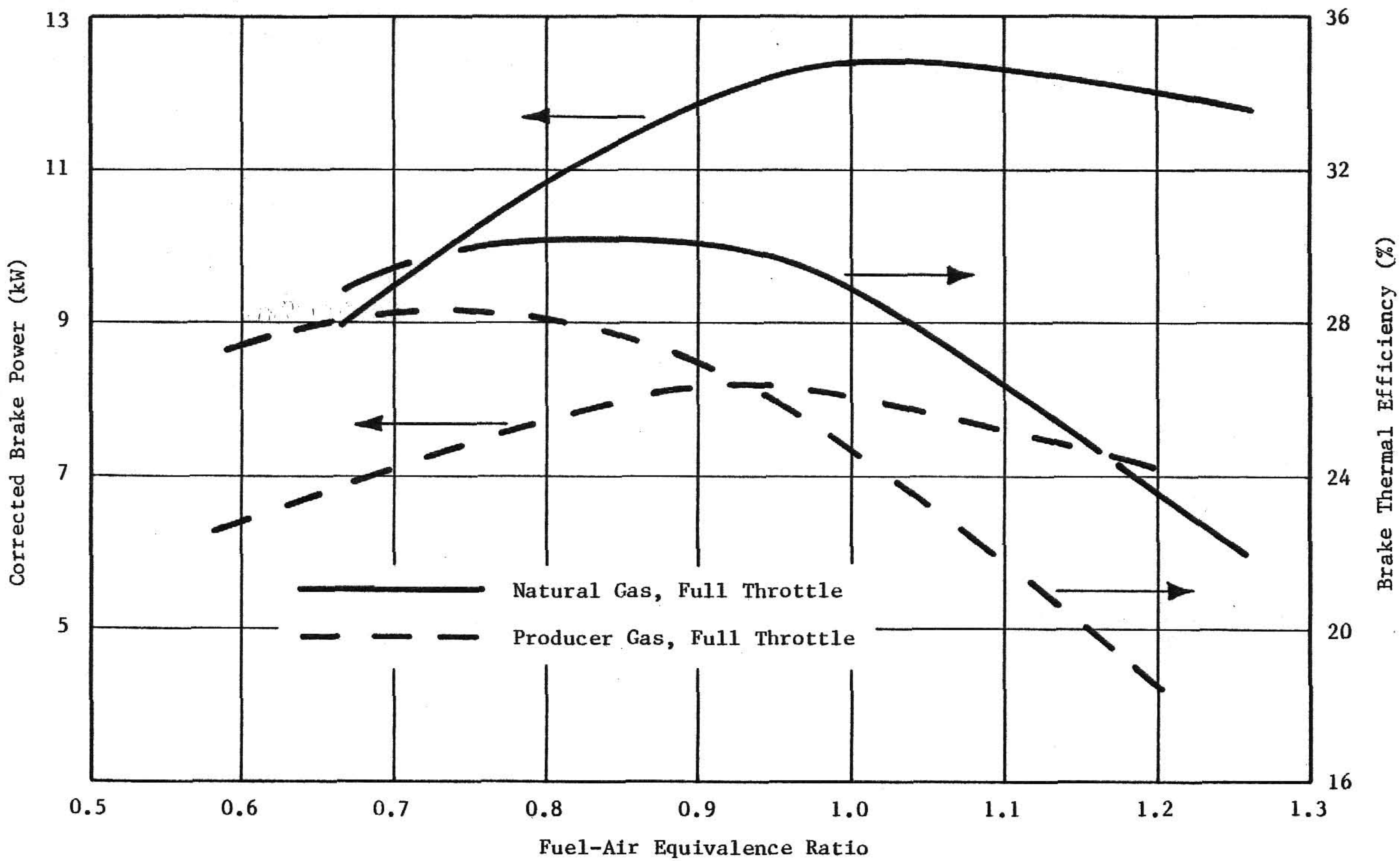

Figure 5. Brake Power and Brake Thermal Efficiency at 2600 RPM (Abstracted from Figures 14 and 15).

tion.

As discussed earlier and shown in Table 3, the number of molecules present in the combustion chamber decreases during combustion of producer gas. Those values are for stoichiometric conditions. For a lean mixture with an equivalence ratio of 0.931, the number of molecules still decreases by 5.0%, assuming complete combustion of the fuel. What happens with a rich mixture of either fuel is more difficult to predict because of the uncertainty of the equilibrium reached between products, reactants, and intermediate species. In either case, it is probably pointless to attempt to account for performance differences due to this phenomenon in this series of tests. The reason being that the composition of the fuel was not exactly known. True, the gaseous fraction was accounted for. However, as will be discussed in a later section, the producer gas contained a tar-oil fraction that was not analyzed. Even though this fraction was ignored (because its quantity and quality were unknown) during efficiency and equivalence ratio calculations, it may have been partially or completely combusted in the cylinder. In that case, the brake thermal efficiency curve for producer gas would be lower because the engine received additional energy in the form of oils and tars that was not considered.

Also, knowing the contribution of the tar-oil fraction to the equivalence ratio may have caused the power curve to shift to the right. Any combustion of tar and oil that occurred meant that the air-fuel mixture being utilized was richer than was calculated. In order for the contribution of the tar-oil fraction to be known, additional tests must be performed using producer gas completely free of suspended tar and oil.

Then, exhaust gas analysis would be useful in determining the equivalence ratio at which combustion is most complete, that is, where the amount of unburned hydrocarbons is a minimum. Recall that Moore and Roy (1956) found that exhaust methane content in a methane fueled engine increased for both rich and very lean mixtures. In addition to causing horizontal displacement of the power curve, the tar-oil fraction may have affected the magnitude of the power curve. Again, it was impossible to quantify this effect using the results of the tests completed.

By inspection of the data plotted in Appendix A, Figures 14 through 19, one sees that the results for natural gas tests are much more consistent than those obtained during producer gas tests. The author believes this can be traced to the changing producer gas quality. Referring to Figure 3, assume the engine is operating at an equivalence ratio of 0.9 with a fuel having a heating value of 5.59 MJ/m^3 (150 BTU/ft^3). About 38.4% of the total mixture is occupied by fuel. Now, suppose the gasifier feed rate decreases and the heating value quickly drops to 4.47 MJ/m^3 (120 BTU/ft^3) before another gas sample is taken. If the air and fuel metering valves are left unchanged, the proportion of fuel in the mixture is still the same. However, the engine is now operating at an equivalence ratio of 0.74. From Figure 5, the engine power will have dropped from 8.0 kW to 7.5 kW, a drop of 6%. This is not to say that the behavior of the engine is inconsistent, but rather that knowledge of the fuel composition did not keep pace with the dynamics of the gasifier output.

Ignition Timing and Exhaust Temperature

While maximum power output for natural gas occurred with a mixture 2-3% rich (Figure 5), maximum exhaust temperature and minimum ignition timing plotted in Figure 6 occurred with mixtures 2-3% lean. The ignition timing curve indicates the value of total spark advance giving maximum torque. Low values of spark advance are an indication of a fast burning air-fuel mixture. Maximum exhaust temperature was about 910 K and minimum ignition advance was 29 degrees before top dead center (^{o}BTDC). Maximum exhaust temperature for naturally aspirated producer gas operation at 2600 RPM was about 801 K and minimum ignition advance was 39^{o} BTCD. These differences were most likely due to the fact that the producer gas was diluted with inert nitrogen and carbon dioxide at levels of 65-70%. Like natural gas operation, maximum exhaust temperature and minimum ignition advance for producer gas operation occurred with air-fuel mixtures leaner than the maximum power mixture. A summary of the significant results for naturally aspirated engine operation at 2600 RPM is presented in Table 4.

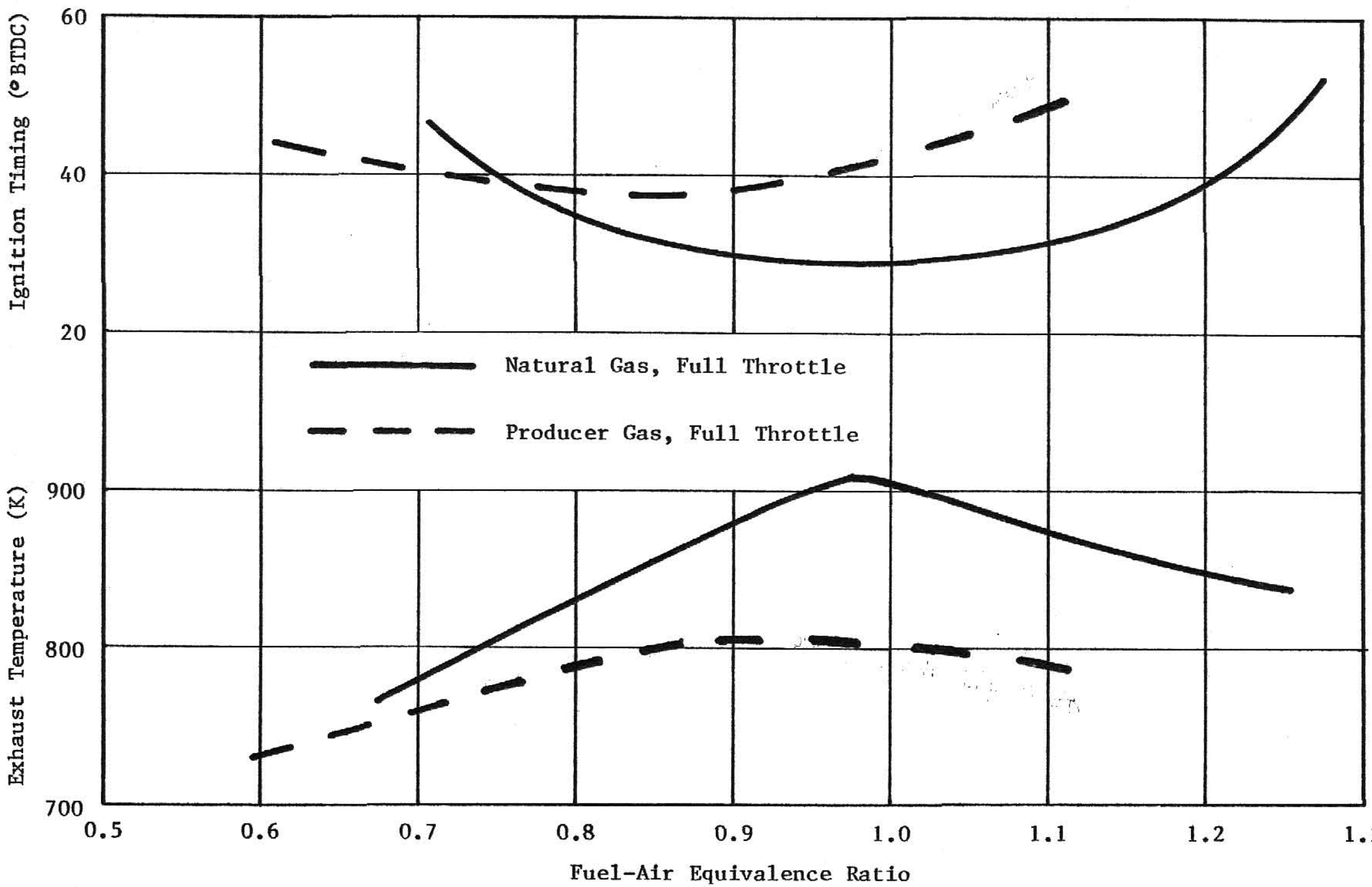

Figure 6. Exhaust Temperature and Ignition Timing at 2600 RPM (Abstracted from Figure 16).

Table 4. Significant Results for Naturally Aspirated Engine Operation at 2600 RPM

Parameter	Result
Maximum Natural Gas Power	12.4 kW at ϕ = 1.025*
Maximum Producer Gas Power	8.1 kW at ϕ = 0.931
Ratio of Producer Gas to Natural Gas Power	0.653
Ratio of Air-Fuel Mixture Lower Heating Values at Maximum Power Conditions	0.671
Brake Thermal Efficiency at Maximum Power Condition	
Natural Gas	28.4%
Producer Gas	26.4%
Maximum Natural Gas Thermal Efficiency	30.1% at ϕ = 0.80 to 0.90
Maximum Producer Gas Thermal Efficiency	28.3% at ϕ = 0.74
Maximum Exhaust Temperature	
Natural Gas	910 K at ϕ = 0.98
Producer Gas	801 K at ϕ = 0.95
Minimum Ignition Advance	
Natural Gas	29^{o} BTDC at ϕ = 0.925 to 1.025
Producer Gas	39^{o} BTDC at ϕ = 0.81

*ϕ = Fuel-Air Equivalence Ratio

Supercharged Tests at 2200 RPM

Basis for Comparison

Before performing tests under supercharged conditions at 2200 RPM, naturally aspirated tests were performed at this speed to serve as a basis for comparison. Results of naturally aspirated tests at 2200 RPM are summarized in Table 5.

Natural gas results at 2200 RPM are very similar to those obtained at 2600 RPM. In both instances, maximum power occurred at an equivalence ratio of 1.025. Maximum natural gas brake thermal efficiencies fell between equivalence ratios of 0.8 to 0.9 at both speeds. In both cases the maximum exhaust temperature occurred with a mixture 2% lean, and finally, minimum ignition advances fell in the range of equivalence ratios of from 0.9 to 1.0.

Results for producer gas operation are also consistent, but there are some differences. For example, at 2600 RPM, brake thermal efficiency peaked at an equivalence ratio of 0.74. At 2200, though, the maximum efficiency was at an equivalence ratio of 0.52. Because of rough engine operation for very lean mixtures, the peak was difficult to locate. One may notice by inspection of Figures 15 and 18 that the consistency of the results for brake thermal efficiency decreases below equivalence ratios of 0.8 to 0.9 for naturally aspirated producer gas operation. No reasonable explanation for this particular reaction is known to the author.

One producer gas result contradicting logic is that the minimum ignition advance is greater for the slower engine speed. One would

Table 5. Significant Results for Naturally Aspirated Engine Operation at 2200 RPM

Parameter	Result
Maximum Natural Gas Power	11.1 kW at ϕ = 1.025*
Maximum Producer Gas Power	7.5 kW at ϕ = 0.88
Ratio of Producer Gas to Natural Gas Power	0.676
Ratio of Air-Fuel Mixture Lower Heating Values at Maximum Power Conditions	0.649
Brake Thermal Efficiency at Maximum Power Condition	
Natural Gas	28.6%
Producer Gas	28.6%
Maximum Natural Gas Thermal Efficiency	30.5% at ϕ = 0.8 to 0.9
Maximum Producer Gas Thermal Efficiency	32.1% at ϕ = 0.52
Maximum Exhaust Temperature	
Natural Gas	880 K at ϕ = 0.98
Producer Gas	806 K at ϕ = 0.92
Minimum Ignition Advance	
Natural Gas	25° BTDC at ϕ = 0.9 to 1.0
Producer Gas	42° BTDC at ϕ = 0.84

*ϕ = Fuel-Air Equivalence Ratio

expect the opposite to be true. A possible explanation is that the lower engine speed had less turbulence in the combustion chamber which resulted in a lower flame speed. Difficulty was experienced in pinpointing the correct ignition timing for a given mixture. The large amount of scatter at 2600 RPM resulted in a low R-squared value for the fitted equation in

Appendix B, Table 7. The corresponding curves in Figures 6 and 16 were drawn with the aid of the equation and prior knowledge of the ignition advance requirements for various fuels and air-fuel mixtures.

Selection of Boost Levels

Before any supercharging was performed, an estimate of the boost level required to regain natural gas power levels was made by considering the amount of potential chemical energy present in the combustion chamber prior to ignition. Using typical fuel compositions, the ratio of producer gas to natural gas heating values on a volumetric basis for stoichiometric conditions was about 0.72. In order to get an equivalent amount of energy into the cylinder, the density of the mixture would have to be increased to a level equal to the reciprocal of the heating value ratio. If the mixture density were increased isothermally by pressure alone, that would indicate a required pressure level of about 1.4 times atmospheric pressure. Preliminary tests confirmed that a boost level of 0.4 atmospheres (corresponding to a pressure level or pressure ratio of 1.4) achieved natural gas power levels.

However, the compression of the air-fuel mixture was definitely not an isothermal process. At a boost level of 0.3, there was a 40K (72° F) temperature rise across the blower, and at a boost level of 0.4 there was a temperature rise of 58K (105° F) across the blower. This of course, reduces the density of the air-fuel charge. In the end, the volumetric efficiency of the engine at a pressure ratio of 1.4 atmospheres was about 120% while during naturally aspirated conditions the volumetric efficiency was about 84%. Since volumetric efficiency is calculated on a mass basis, and since for a given fuel the amount of inducted mass is directly proportional to the amount of inducted potential energy, the ratio of volumetric efficiencies is an indication of the effectiveness of supercharging for power gain. In this case, the ratio of 120 to 84 is very close to the predicted pressure ratio of 1.4 necessary for natural gas power level recovery.

Supercharged Engine Power Output

Corrected engine brake output for boost levels of 0.3 and 0.4 atmospheres is shown graphically in Figure 7. Boosting the producer gas air-fuel mixture to a manifold pressure of 1.3 times atmospheric resulted in a corrected brake power of 10.4 kW, 93.7% of the natural gas power output. Notice that the maximum power output was at an equivalence ratio of 0.86, which is leaner than the maximum power point under naturally aspirated conditions. It is not known whether this was due to changes in the chemistry of combustion caused by supercharging, or if it was due to a slightly different fuel gas composition. Remember that a different batch of corn stover was used during naturally aspirated tests. The method of processing the stover for the supercharging tests required that a different feed rate be used which resulted in the composition change.

A pressure ratio of 1.4 atmospheres gave a corrected power output of 11.2 kW, 101% of the maximum natural gas power level. Except for magnitude, the power curve for a 0.4 boost level closely resembles the 0.3 boost level power curve, with maximum power occurring at an equivalence ratio of 0.87.

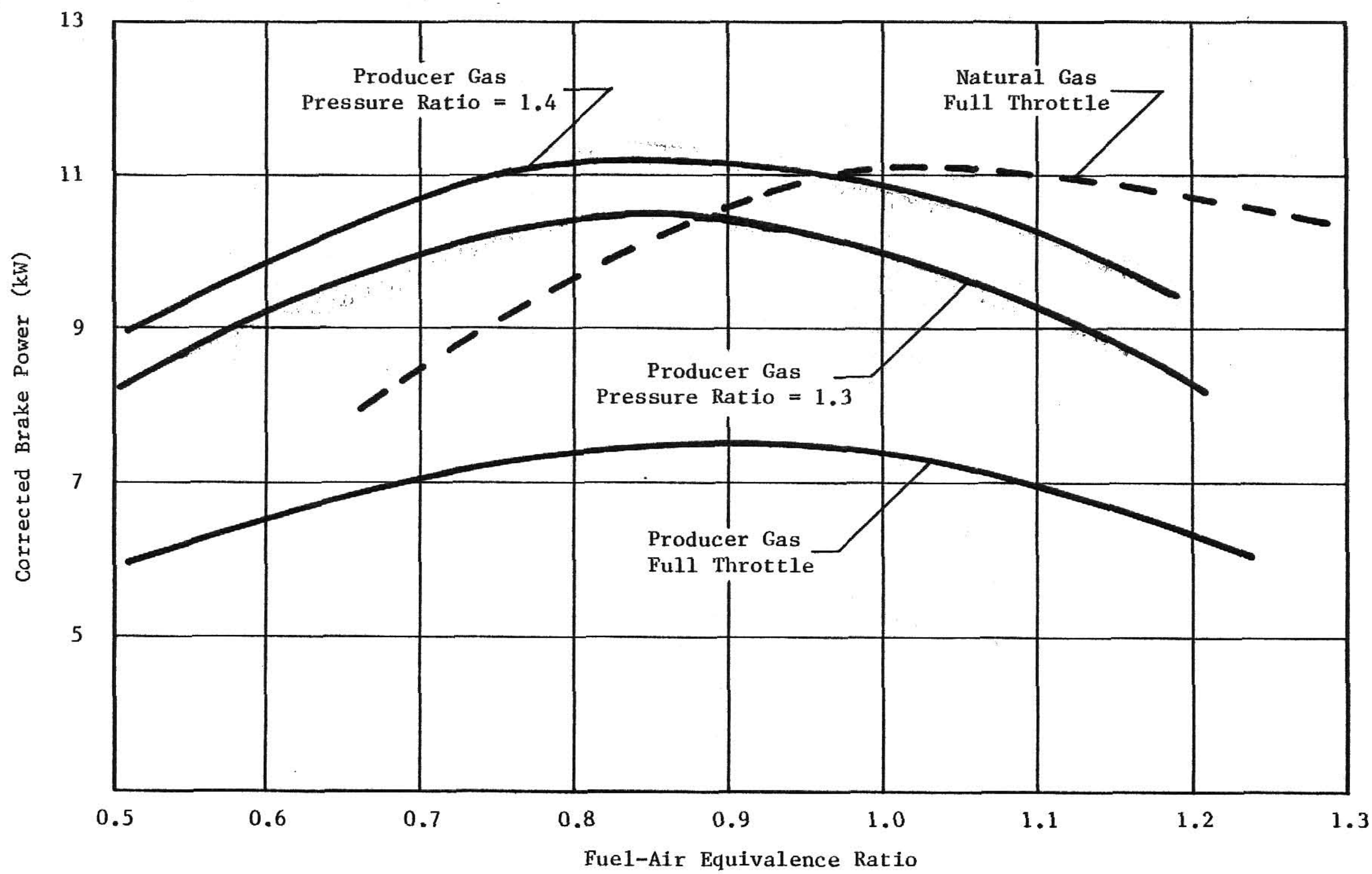

Figure 7. Brake Power at 2200 RPM for Naturally Aspirated and Supercharged Conditions (Abstracted from Figures 17 and 20).

In this case of an independent, mechanically driven supercharger, concern was expressed about the amount of engine power output that resulted from the pumping effect of the supercharger. A power output of 11.2 kW for this engine was equivalent to a brake mean effective pressure of 723 kPa (105 psi). The boost pressure of 0.4 atmospheres was equivalent to 39.4 kPa (5.7 psi) at an atmospheric pressure of 98.5 kPa (29.1 in Hg). Thus, the contribution to engine power output due to the pumping effect of the supercharger may be estimated as the ratio of 39.4 to 723, or 5.4%. So, if this supercharger were indeed driven mechanically by this engine, one would expect the useful power output at a pressure ratio of 1.4 to drop by 5.4% or more, depending on the efficiency of the supercharger. But, one purpose of this portion of the testing program was to simulate turbocharged conditions. Because an exhaust driven supercharger, a turbocharger, is operated by otherwise wasted exhaust energy, little if any of the load required to operate the turbocharger is placed on the engine. So, no adjustment will be made on the supercharged engine power curve due to the pumping effect of the supercharger.

Supercharged Engine Efficiency

Brake thermal efficiency curves for engine operation at 2200 RPM are shown in Figure 8. As expected, supercharging increased the efficiency of the engine over that experienced during naturally aspirated trials. Notice that, especially for the 1.4 pressure ratio, the equivalence ratios yielding maximum power also resulted in the largest increases in brake thermal efficiency. As mentioned in the literature review, the power output increased faster than did the friction power. Using the SAE standard procedure, if friction power is found by calculation rather than

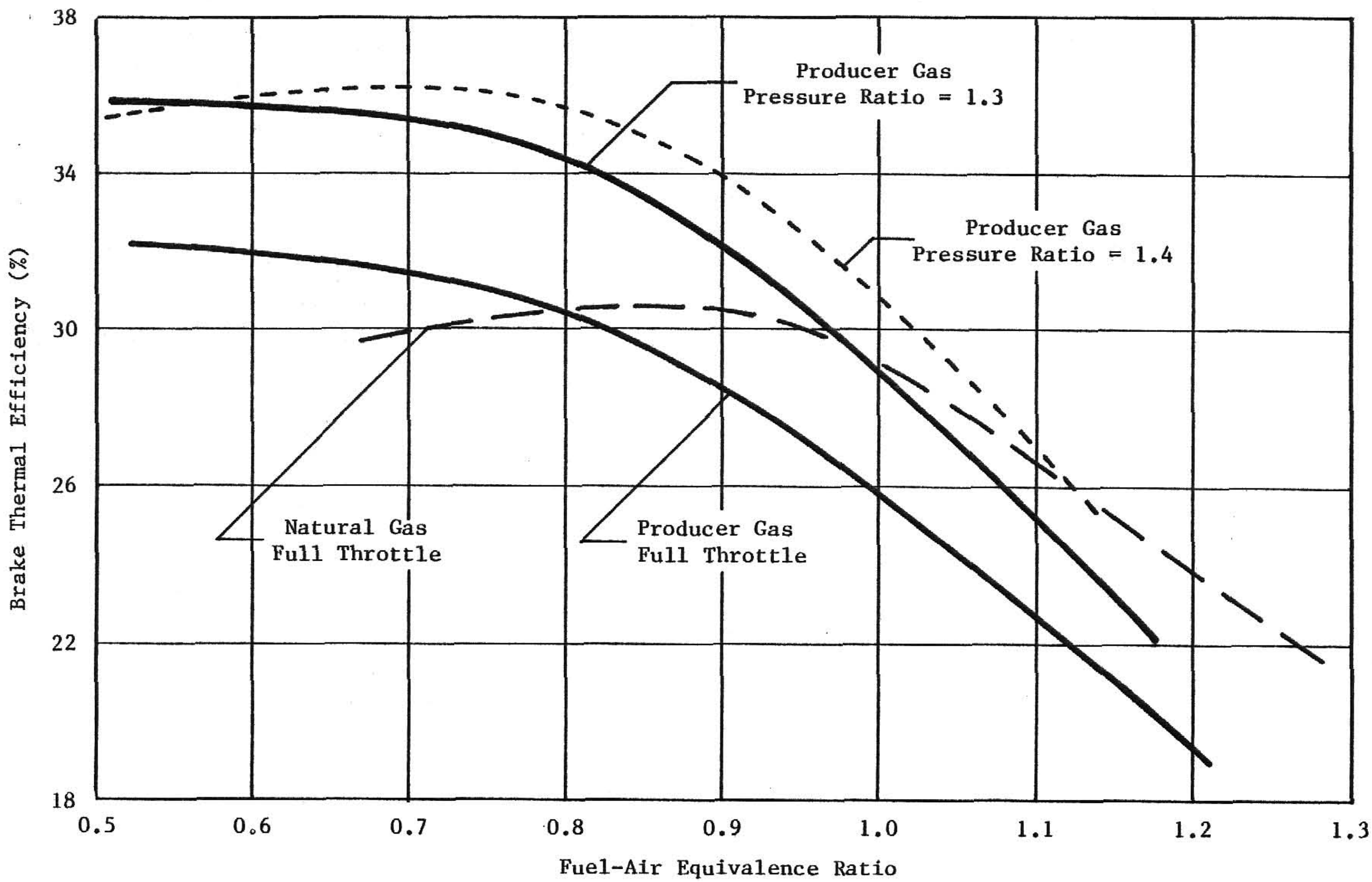

Figure 8. Brake Thermal Efficiency at 2200 RPM for Naturally Aspirated and Supercharged Conditions (Abstracted from Figures 18 and 21).

the hot-motored method, the mechanical efficiency equation used is most sensitive to engine speed, but it is also dependent on brake mean effective pressure. For the naturally aspirated tests in the maximum power range, the engine's mechanical efficiency was about 77%. At the 1.4 pressure ratio, the mechanical efficiency increased to 85%.

Exhaust Temperature and Ignition Timing

Peak exhaust temperatures during supercharged trials were 29-32 K higher than during naturally aspirated trials, but they occurred at essentially the same equivalence ratio (See Figure 9). This increase was about the same as the increase in the inlet mixture temperature due to the supercharger. But, the exhaust temperature is still about 44 K below that experienced during the natural gas trials. This should lead to prolonged valve life.

As expected, the minimum ignition advance for maximum torque decreased for supercharged conditions. The reason is that the increased temperature and pressure in the combustion chamber at ignition caused a higher reaction rate. Since the flame speed increased, the engine was operating closer to the constant volume heat addition of the Otto cycle which also may have contributed to the increased brake thermal efficiencies. But, the flame speed with supercharged producer gas was still below that obtained with natural gas as shown in Figure 9. Also, the interaction of pressure, temperature, and mixture composition caused the minimum ignition timing to occur with a leaner mixture.

By inspection of the data plotted in Figures 22 and 23, one can see that the curves for exhaust temperature and ignition timing are

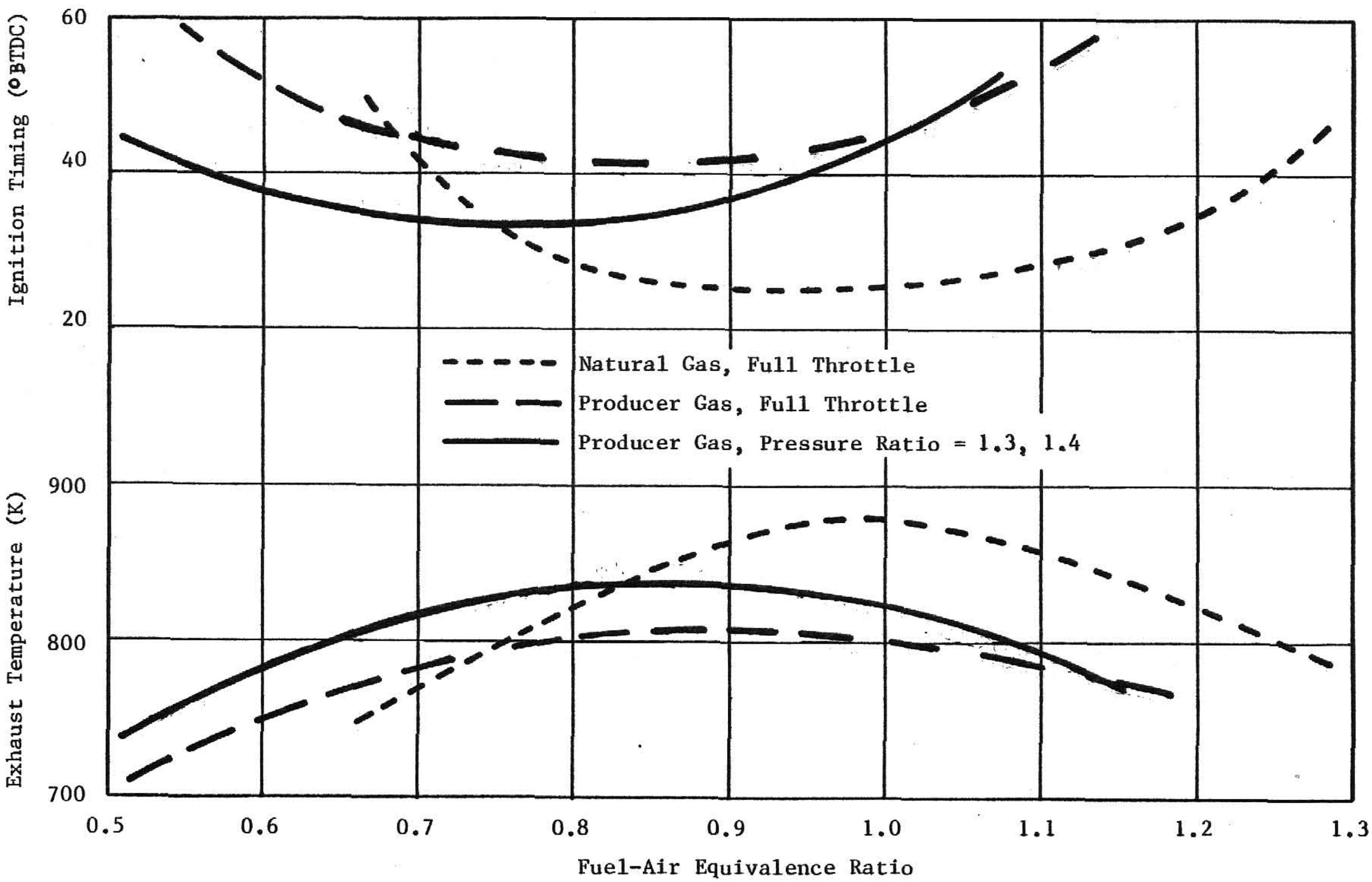

Figure 9. Exhaust Temperature and Ignition Timing at 2200 RPM for Naturally Aspirated and Supercharged Conditions (Abstracted from Figures 19, 22, and 23).

essentially the same at pressure ratios of 1.3 and 1.4. These curves are shown as one in Figure 9. Results for supercharged producer gas operation are summarized in Table 6.

Table 6. Significant Results for Supercharged Engine Operation at 2200 RPM

Maximum Power at 1.3 Pressure Ratio	10.4 kW at Ø = 0.86*
Maximum Power at 1.4 Pressure Ratio	11.2 kW at Ø = 0.87
Ratio of Supercharged Producer Gas Power to Natural Gas Power	
1.3 Pressure Ratio	0.937
1.4 Pressure Ratio	1.01
Ratio of Supercharged to Naturally Aspirated Producer Gas Power	
1.3 Pressure Ratio	1.39
1.4 Pressure Ratio	1.49
Brake Thermal Efficiency at Maximum Power Condition	
1.3 Pressure Ratio	33.2%
1.4 Pressure Ratio	34.6%
Maximum Brake Thermal Efficiency	
1.3 Pressure Ratio	35.9% at Ø = 0.60
1.4 Pressure Ratio	36.3% at Ø = 0.68
Maximum Exhaust Temperature	
1.3 Pressure Ratio	838 K at Ø = 0.86
1.4 Pressure Ratio	835 K at Ø = 0.88
Minimum Ignition Advance	
1.3 Pressure Ratio	33° BTDC at Ø = 0.76
1.4 Pressure Ratio	34° BTDC at Ø = 0.76

*Ø = Fuel-Air Equivalence Ratio

Gas Cleaning

All efforts to clean the producer gas prior to use met with limited success. The problem was not with entrained solid particles, but rather with suspended tar-oil mists. Though not successful, the best filter medium tested was glass wool in a 7.6 cm diameter column. Larger columns were not as effective because the slower gas velocity reduced impingement of the mist on the filter fibers.

Figures 10, 11, and 12 are photographs of the intake valves, exhaust valves, and combustion chamber after only one hour of initial testing. At the time, no filters were in place downstream from the venturi scrubber. Figure 13 is a photograph of the intake valves after accumulating about twenty hours of engine operation while the filter train shown in Figure 2 was in use. Obviously, the system was ineffective in completely removing the tar-oil mist. Further comments on this subject are included in the section "Suggestions for Further Research".

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Figure 10. Intake Valves after One Hour of Engine Operation, No Gas Filtering.

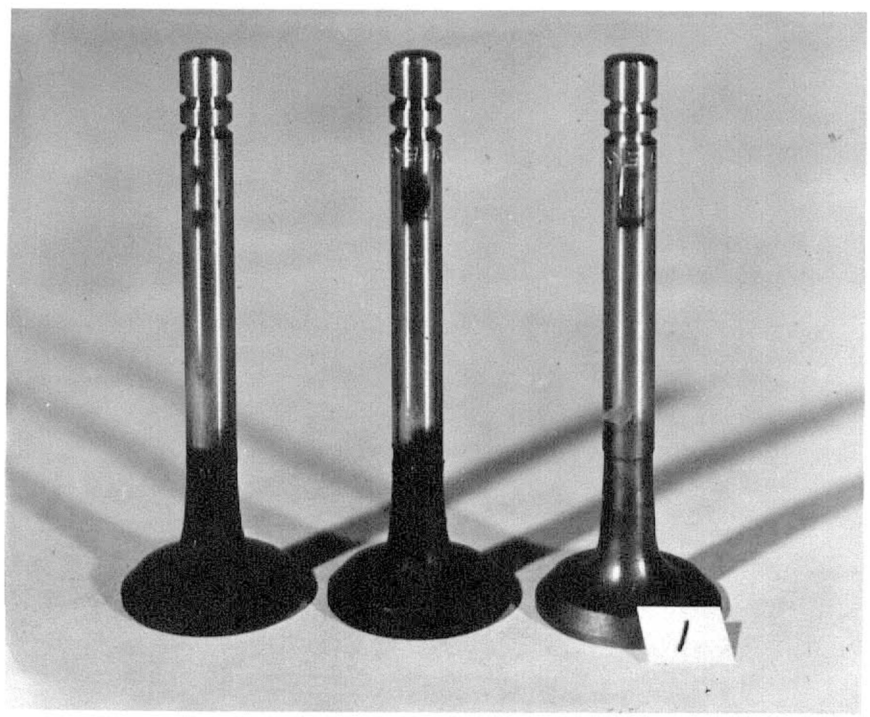

Figure 11. Exhaust Valves after One Hour of Engine Operation, No Gas Filtering.

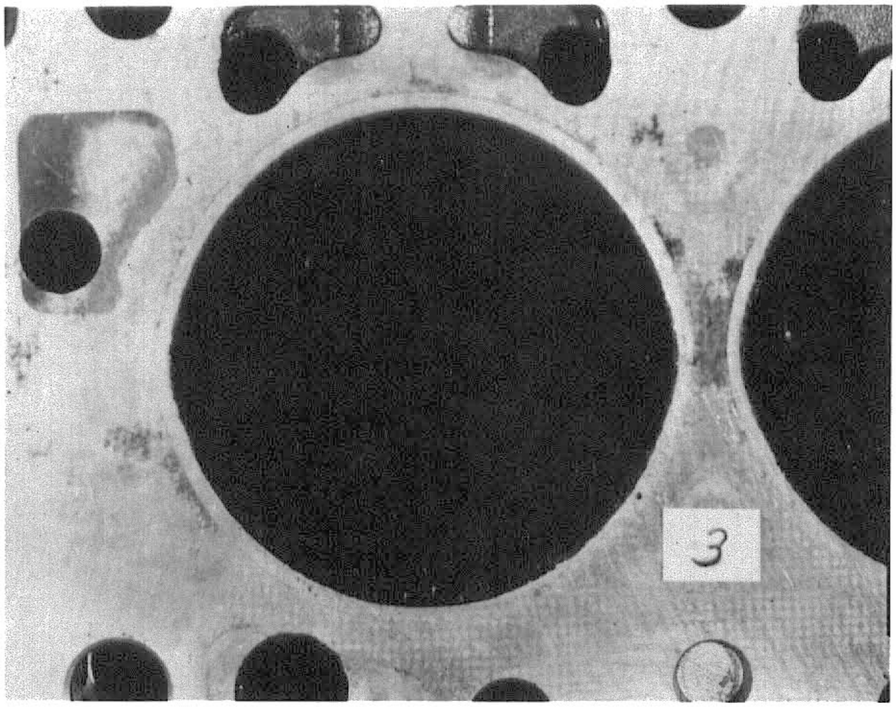

Figure 12. Combustion Chamber after One Hour of Engine Operation, No Gas Filtering.

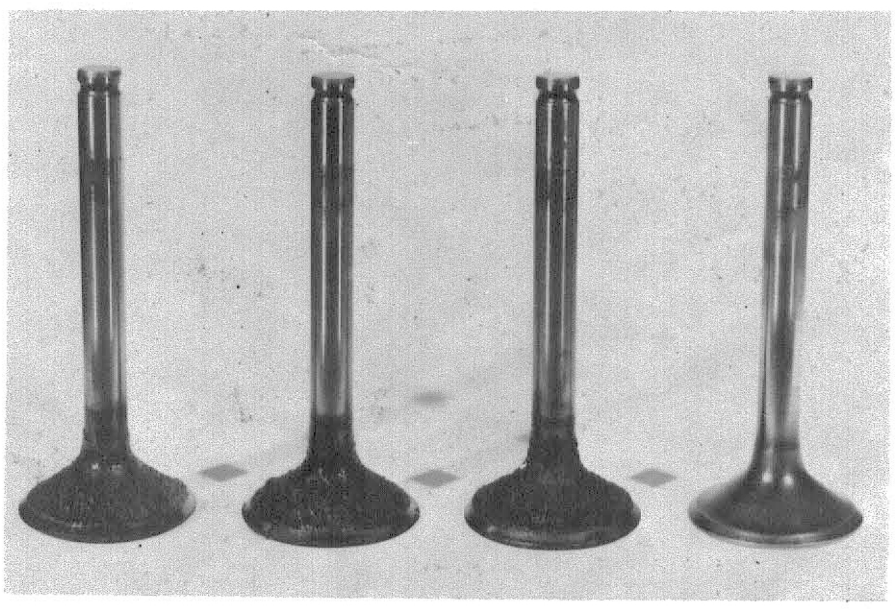

Figure 13. Intake Valves after Twenty Hours of Engine Operation, Packed Column Filtering System.

CONCLUSIONS

1. When low energy gas from a fluidized bed gasifier is to be utilized in an internal combustion engine, provisions must be made in the fuel delivery system to convert the unsteady, pulsating flow to a constant pressure flow.

2. The carburetion system must be capable of supplying a volume flow rate of fuel equal to 30-50% of the total air-fuel mixture.

3. When a gaseous fuel has a low volumetric energy density, small variations in the fuel quality will have a noticable effect on engine performance.

4. The reduction in engine power output from natural gas levels during producer gas operation is closely related to the ratio of air-fuel mixture heating values. For the particular fuels used in these tests, that ratio was about 66%.

5. During naturally aspirated conditions, maximum natural gas power occurred with a mixture 2-3% rich, while maximum producer gas power occurred with mixtures 7-12% lean. The values for producer gas operation may have been affected by a tar-oil mist entrained in the fuel.

6. During naturally aspirated conditions and while operating at the maximum power air-fuel mixture, natural gas gave a 0-2% increase in brake thermal efficiency over producer gas. The difference in efficiency decreased for leaner mixtures and increased for richer mixtures.

7. The dilution of the producer gas with nitrogen and carbon dioxide volumetric percentages totaling around 70% caused a reduction in exhaust temperature and flame speed in the combustion chamber. The reduced flame speed required a greater ignition advance in order to achieve maximum torque.

8. A manifold pressure of 1.4 atmospheres during producer gas operation resulted in full recovery of natural gas power levels. Maximum power during supercharged operation occurred with a mixture 13-14% lean.

9. As expected, supercharged operation resulted in higher brake thermal efficiencies. The ranges of air-fuel mixtures giving maximum power also produced the largest magnitude increase in efficiency.

10. Supercharged exhaust temperatures were higher than those obtained during naturally aspirated tests, but not as high as natural gas exhaust temperatures. The increase in exhaust temperature reflected the temperature increase of the air-fuel mixture during supercharging.

11. The higher temperature and pressure in the combustion chamber during supercharging contributed to the lowering of required ignition advance for maximum power.

12. More effective methods of gas cleaning than those used in this study must be used. This task should not only include removal of the tar-oil mist present in the gas, but also design changes in the gasifier to produce smaller amounts of the mist.

SUMMARY

The results of this research project show that a spark ignition engine will perform satisfactorily when fueled by a fluidized bed gasifier. Although power output is reduced compared to natural gas levels, full power recovery can be achieved with a relatively low level of supercharging.

Important considerations are an adequate fuel delivery system and adequate gas cleaning. Gas cleaning should be approached from a total system point of view, beginning with a new generation design for the gasifier.

SUGGESTIONS FOR FURTHER RESEARCH

These tests have shown the technical feasibility of using low energy gas from crop residues as a replacement for natural gas. Significant problems were encountered with engine deposits caused by insufficient removal of the tar-oil mist in the gas. Before long term engine tests and exhaust gas analysis are performed, the author believes it essential to have a gas with far fewer impurities. Although packed column filters capable of thorough mist removal may be developed, it is impractical to reason that they could be a long term solution for a marketable gasifier. The KSU pilot plant gasifier has served well as a research tool, but a prototype gasification plant could be designed to possibly eliminate some of the problems.

First, it appears that the feedstock material should be fed directly into the bed just above the distributor plate. While retaining all the advantages of the fluidized bed, it would allow the feedstock to have a longer residence time in the reactor and assure that the feedstock reaches the bed temperature rapidly. Additionally, this would eliminate the problem of plugging at the junction of the horizontal auger and the vertical feedpipe.

Next, a high temperature condenser should be placed ahead of the venturi scrubber so that most of the tar vapors could be removed before the gas is sprayed with water. An electrostatic precipitator after the scrubber would remove entrained droplets of tar and water. Finally, the gas should be cooled to lower its water vapor content.

After improved cleaning techniques have been demonstrated, short term engine tests, including exhaust gas analysis should be performed to determine any changes in engine performance characteristics caused by removal of the tars. Then, long term tests should be made in conjunction with an in-the-field demonstration unit for irrigation pumping and electricity generation.

REFERENCES CITED

Benedict, R. P. 1966. Most probable discharge coefficients for ASME flow nozzles. Transactions of the ASME 88(2): 734-744. Journal of Basic Engineering.

Bowden, A. T., E. E. Freeth, and A. D. Rutherford. 1942. Bench and field tests of vehicle gas producer plant as applied to farm tractors. Proceedings of the Institute of Mechanical Engineers. 146: 193-207.

Branders, H. A. 1941. Producer gas is the motor fuel of Finland. Automot. Ind. 84: 482-485.

Burstall, A. F. and M. W. Woods. 1939. Experiments on a high speed producer gas engine. The Engineer. (May 26).

Clark, S. J., R. K. Koelsch, L. R. Whitmer, and W. P. Walawender. 1978. Crop residues: energy for irrigation pumping and crop drying. Paper No. 78-2550 presented at ASAE Winter Meeting. Chicago, Illinios.

Coward, H. F., C. W. Carpenter, and W. Payman. 1919. The dilution limits of inflammability of gasious mixtures. III. The lower limits of some mixed inflammable gases with air. IV. The upper limits of some gases, singly and mixed, in air. Journal Chem. Society. 115: 27-36.

Coward, H.F. and G.W. Jones. 1952. Limits of flammability of gases and vapors. U.S. Bureau of Mine. Bul. 503.

Coward, H. F., G. W. Jones, C. G. Dunkle, and B. E. Hess. 1926. The explosibility of methane and natural gas. Bureau of Mines and Carnegie Inst. Technol. Min. and Met. Investigations. Bul. 30.

Economic Research Service. 1977. Energy and U. S. Agriculture: 1974 Data Base, Vol. 2. Federal Energy Administration, USDA. FEA/D-77/140.

Edwards, J. B. and D. M. Teague. 1970. Unraveling the chemical phenomena occurring in spark ignition engines. SAE Paper No. 700489 presented at mid-year meeting. Detroit, Michigan.

Griffen, E. 1944. The conversion of compression ignition engines to producer gas operation. Report of the Auto. Res. Comm. of the Inst. of Aut. Eng. (August).

Groves, J., J. Craig, W. A. LePori, and R. G. Anthony. 1979. Fluidized bed gasification of cotton gin waste. Paper No. 79-4547 presented at the ASAE Winter Meeting, New Orleans, Louisiana.

Gumz, W. 1942. Kurzes handbook der brennstoff-und feuerungstechnik. Julius Springer. Berlin. (Also, Edwards Brothers, Ann Arbor, Michigan).

Hay, D. R. 1978. Kansas irrigation survey. Engineering Newsletter. Cooperative Extension Service. Kansas State University, Manhattan, Kansas.

Heywood, H. 1941. Loss of power in petrol engines running on producer gas. Engineering (January 24).

Horsfield, B. and R. O. Williams. 1978. Energy for agriculture and the gasification of crop residues. Energy Sources. Vol. 3, Nos. 3 and 4.

Le Chatelier, H. 1891. Estimation of firedamp by flammability limits. Ann. Mines 19. Ser. 8. 388-395.

Marias, J. K. 1936. Some researches on the use of producer gas for motor vehicles. Jour. of S. African Inst. Engs. 34.

Matheson gas data book. 1966. The Matheson Co., Inc. Ontario.

Moore, N. P. W. and B. N. Roy. 1956. Comparative studies of methane and propane as engine fuels. Proc. of Inst. of Mech. Engs. 170. No. 140.

Nowakowska, J. and R. Wiebe. 1945. Bibliography on construction, design, economics, performance, and theory of portable and small stationary gas producers. Report No. AIC-103. USDA. Peoria, Illinios.

Obert, E. F. 1973. Internal combustion engines and air pollution. Intext, Harper, and Row. New York. 87-143.

Raman, K. P., W. P. Walawender, Y. Shimizu, and L. T. Fan. 1980. Gasification of corn stover in a fluidized bed. Paper presented at 1980 ASAE National Energy Symposium.

Rammler, E., K. Breitling, and J. Gall. 1938. Report No. D.83 of the Reichskohlrat. Berlin. Translation: B.C.U.R.A. Document No. C/727. (July 17, 1939).

Segeler, C. George. Editor. 1932. Combustion. 3rd Edition. American Gas Association. New York.

SAE Handbook, Part 2. 1978. Engine rating code - spark ignition. Standard J245. Published by the Society of Automotive Engineers. 24.13-24.18.

Solar Energy Research Institute. 1979. Generator gas: the Swedish experience from 1939-1945. SERI/SP-33-140. Golden, Colorado.

Spiers, J. 1942. The performance of a converted petrol engine on producer gas. Proc. of the Inst. of Auto. Engs. Vol. XXXV, 105. Report of Auto. Res. Comm.

Tatom, J. W., A. R. Colcord, W. M. Williams, K. R. Purdy, and D. Beinstock. 1976. Development of a prototype system for pyrolyis of agricultural and forestry wastes into fuels and other products. Prepared for

EPA. Grant No. R803430-01. Program element EHE-624.

Walawender, W. P. and L. T. Fan. 1978. Gasification of dried feedlot manure in a fluidized bed - preliminary pilot plant tests. Paper presented at 84th National Mtg. of AIChE. Atlanta, Georgia.

Walawender, W. P., K. P. Raman, and L. T. Fan. 1980. Gasification of carbonaceous materials in a fluidized bed reactor. Paper presented at Bio-Energy 80. Atlanta, Georgia.

Williams, R. O., J. R. Goss, J. J. Mehlschau, B. Jenkins, and J. Ramming. 1978. Development of pilot plant gasification system for conversion of crop and wood residues to thermal and electrical energy. American Chem. Soc. Symposium Series No. 76. Solid Wastes and Residues.

Woods, M. W. 1940. An investigation of the high speed producer gas engine. The Engineer. (May 17).

Zinner, K. 1978. Supercharging of internal combustion engines. Translated from German by G. Winkler. Springer-Verlag. New York. 123-132.

APPENDIX A

GRAPHICAL PRESENTATION OF TEST RESULTS

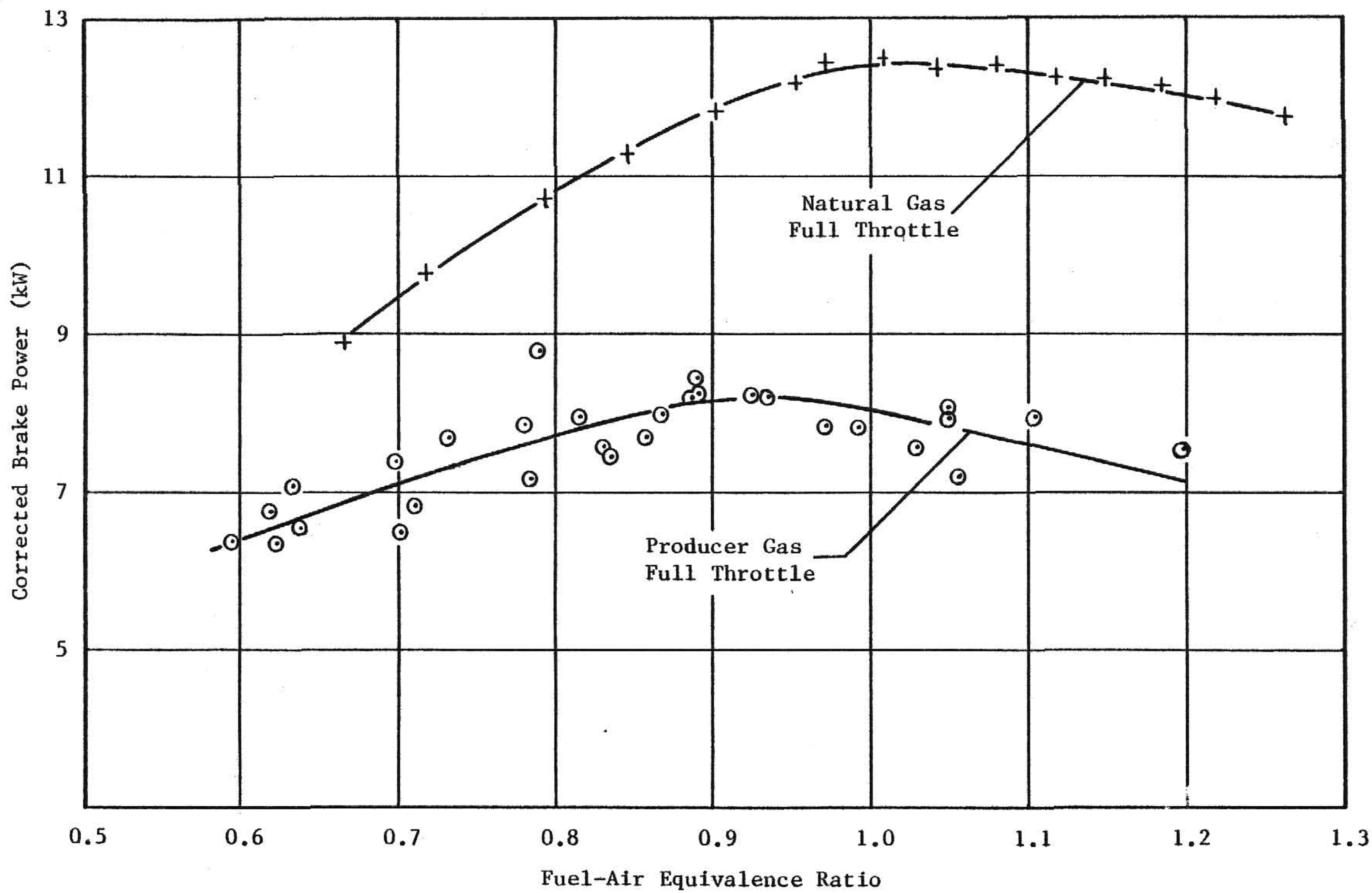

Figure 14. Brake Power versus Equivalence Ratio at 2600 RPM.

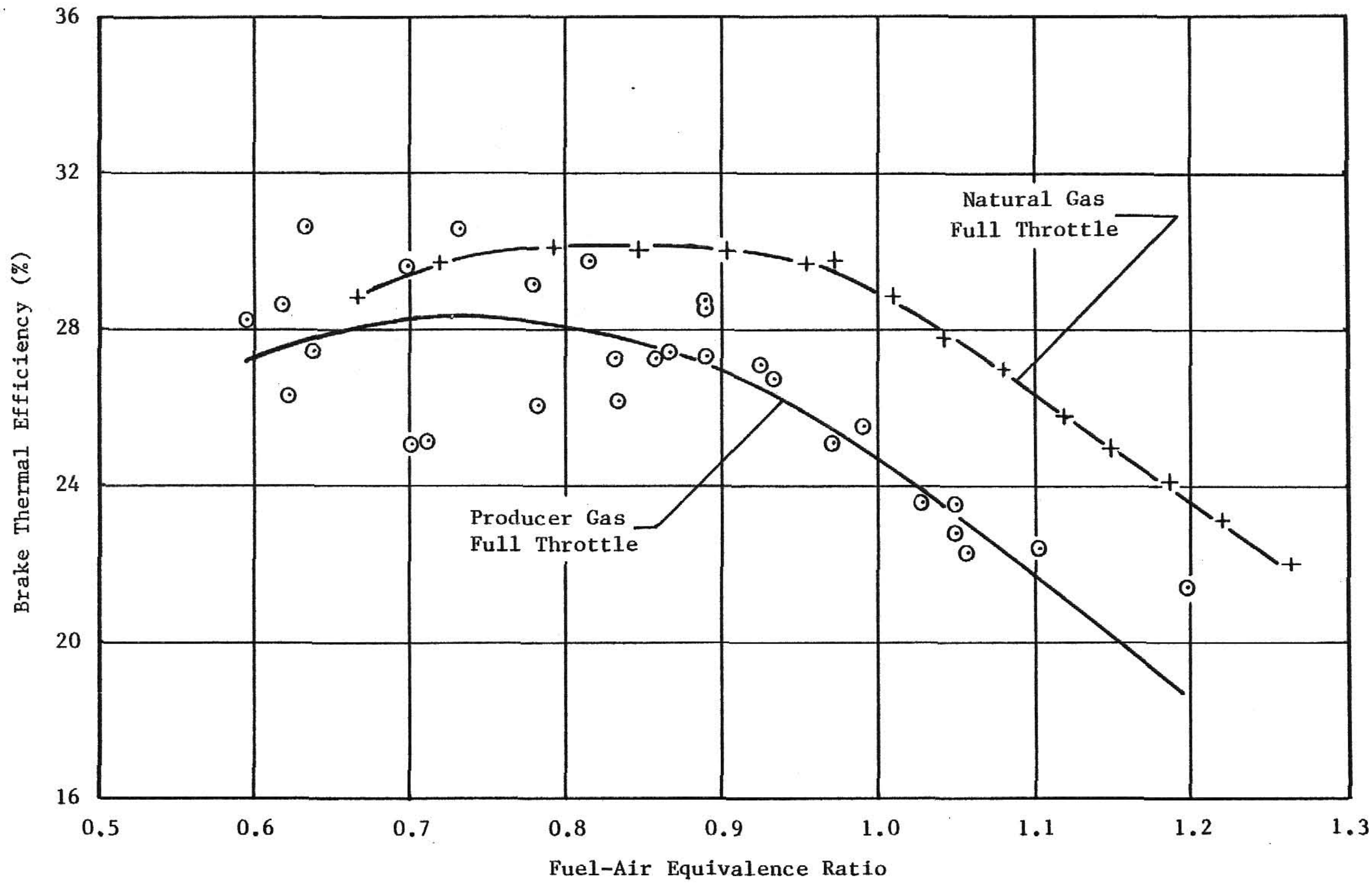

Figure 15. Brake Thermal Efficiency versus Equivalence Ratio at 2600 RPM.

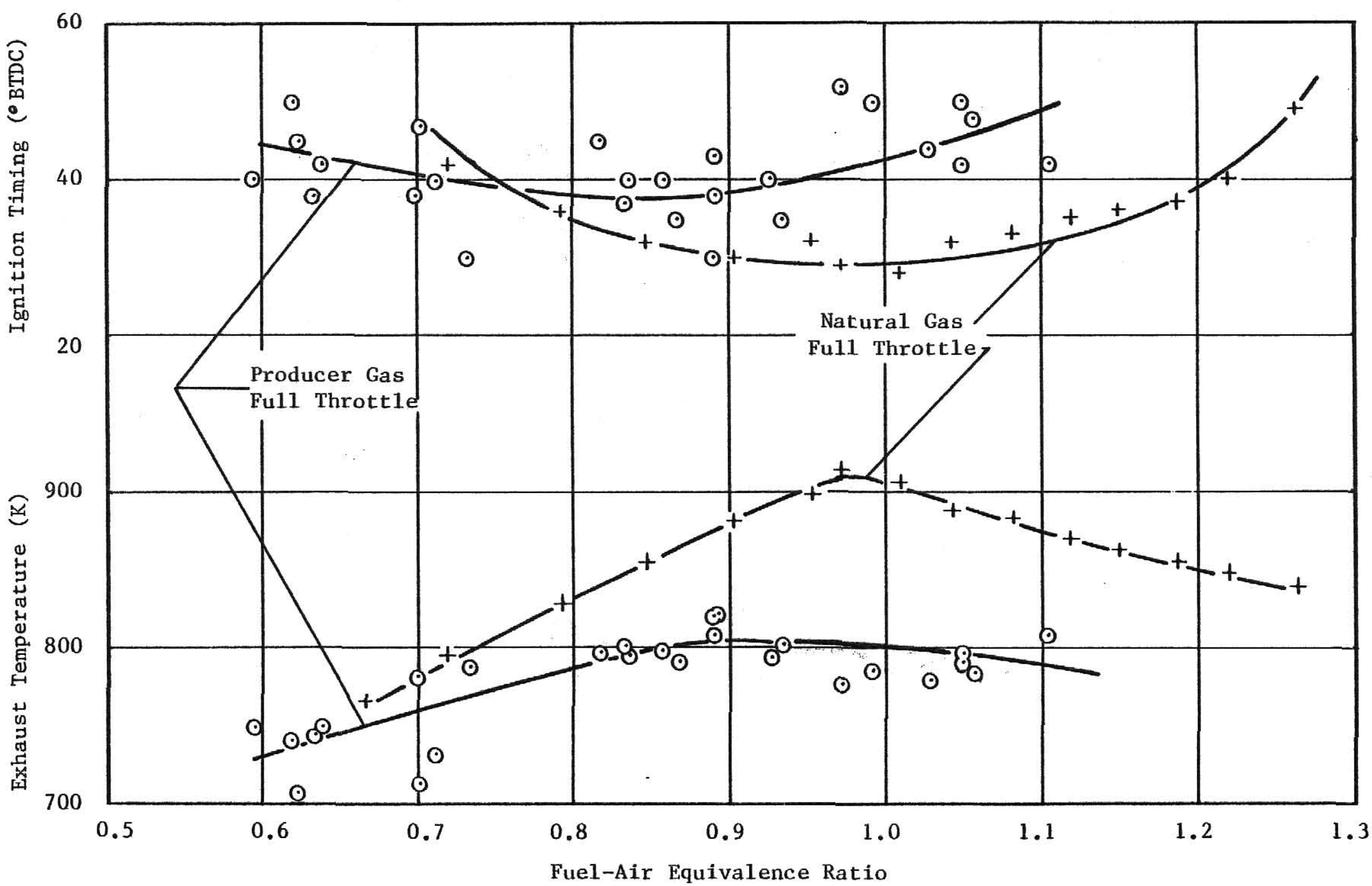

Figure 16. Exhaust Temperature and Ignition Timing versus Equivalence Ratio at 2600 RPM.

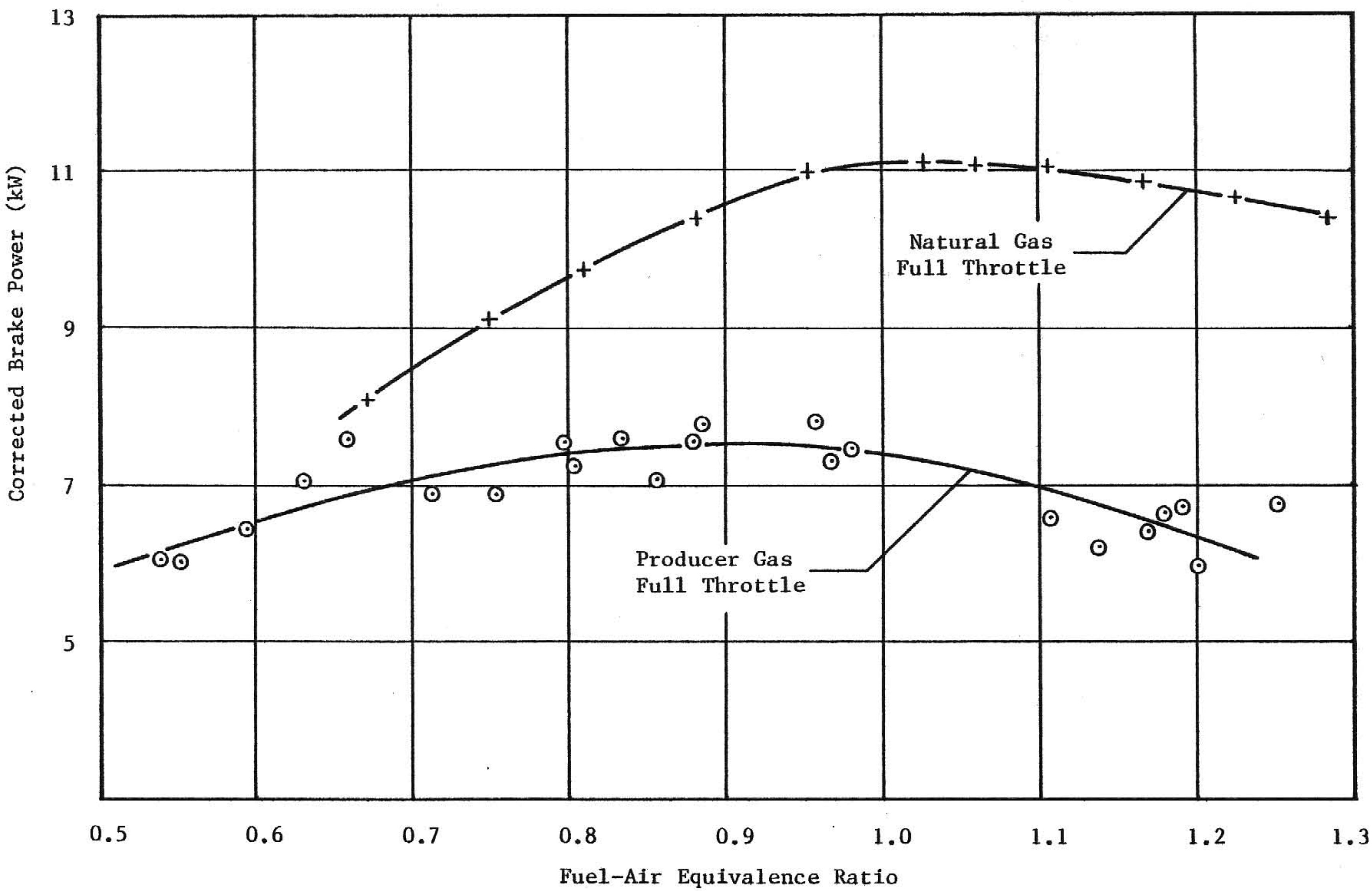

Figure 17. Brake Power versus Equivalence Ratio at 2200 RPM.

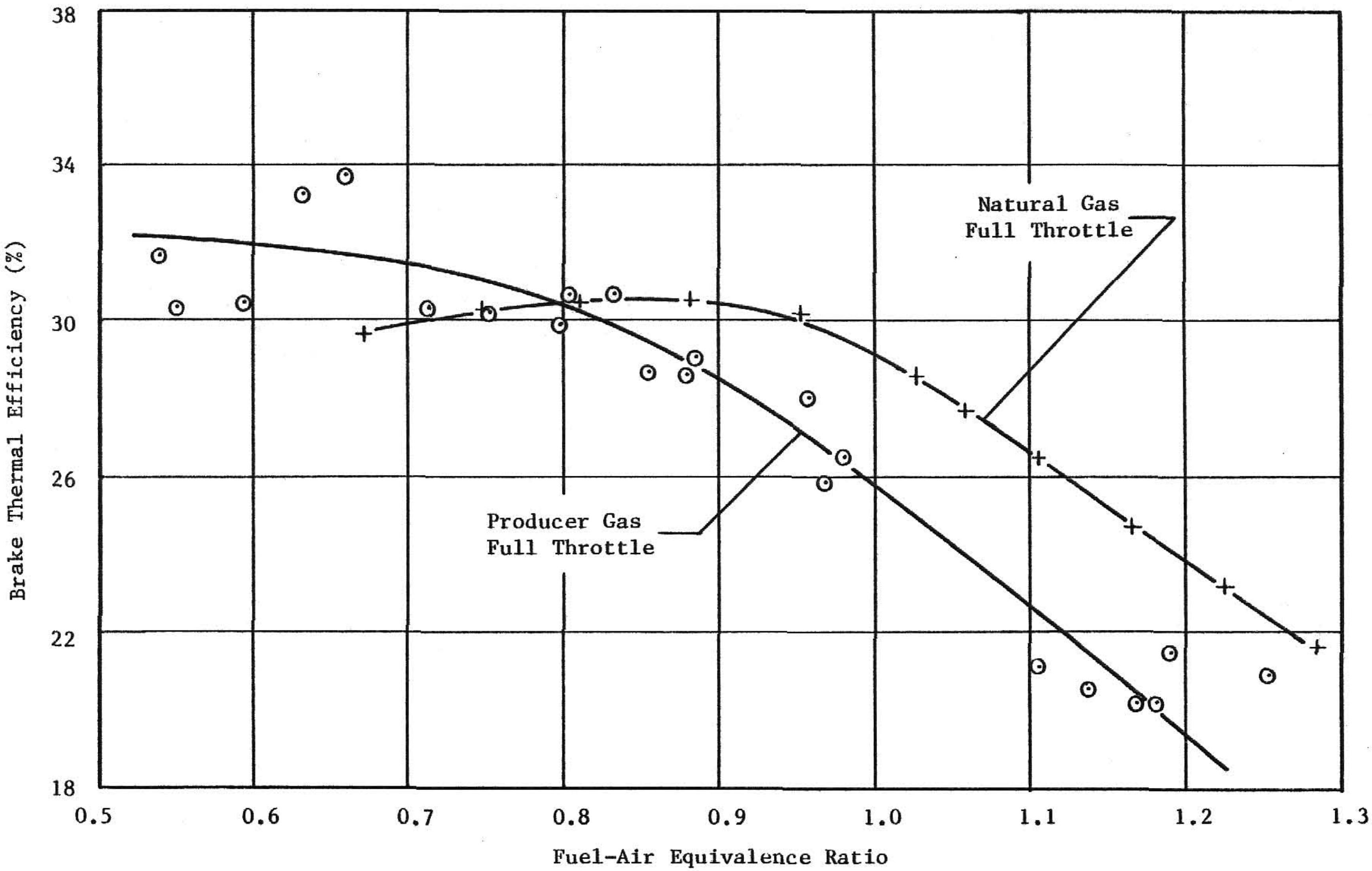

Figure 18. Brake Thermal Efficiency versus Equivalence Ratio at 2200 RPM.

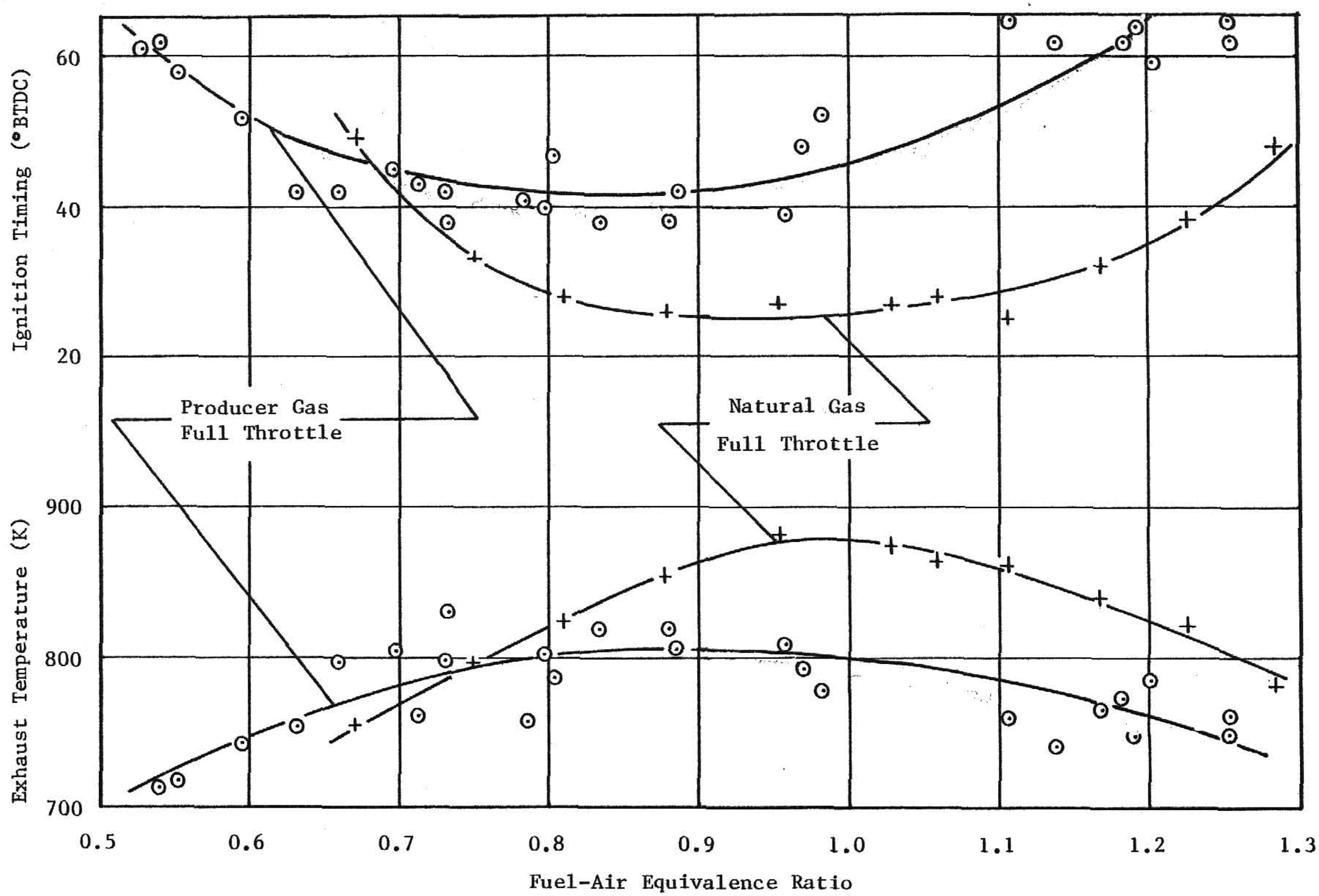

Figure 19. Exhaust Temperature and Ignition Timing versus Equivalence Ratio at 2200 RPM.

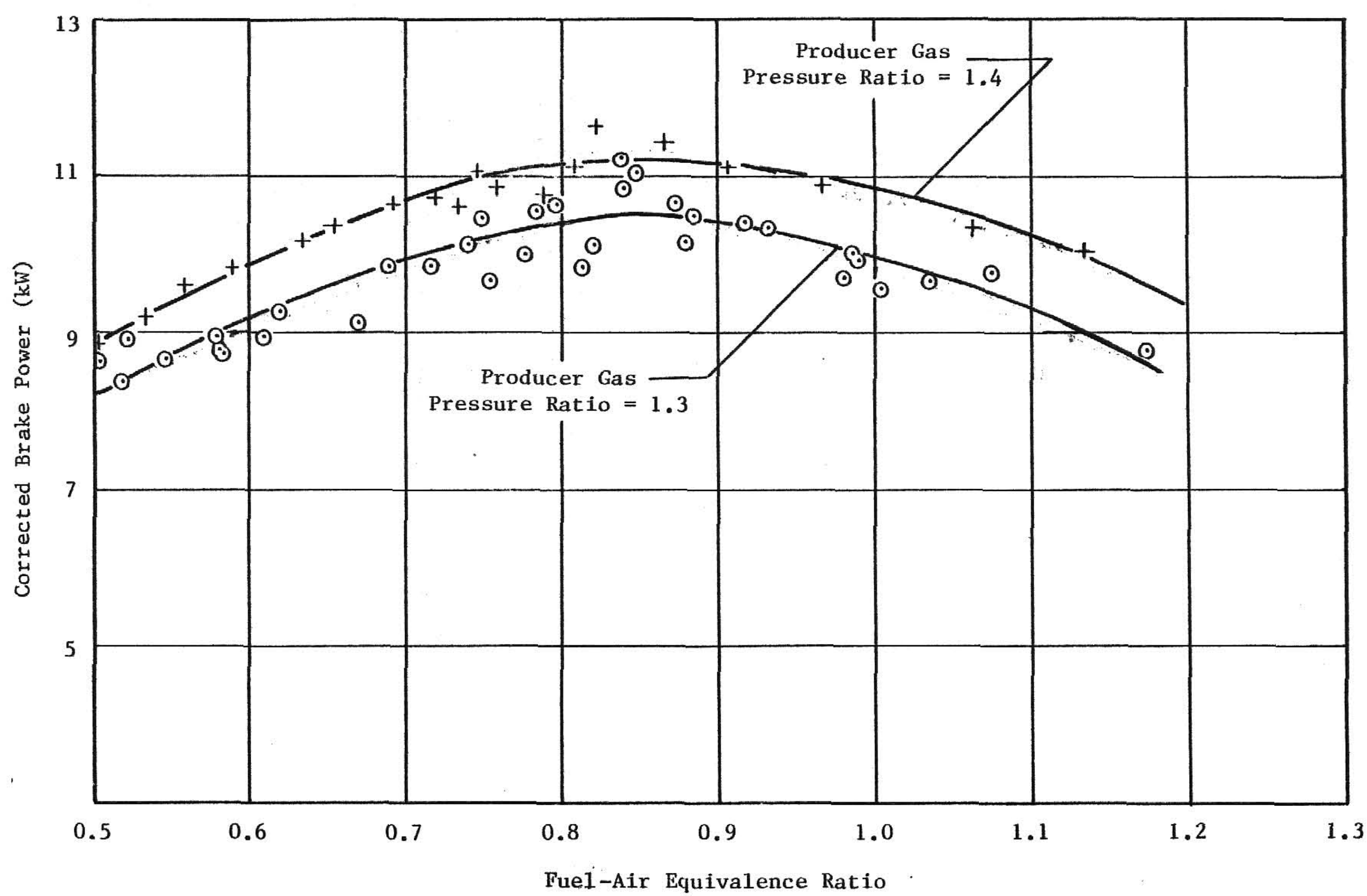

Figure 20. Supercharged Brake Power versus Equivalence Ratio at 2200 RPM.

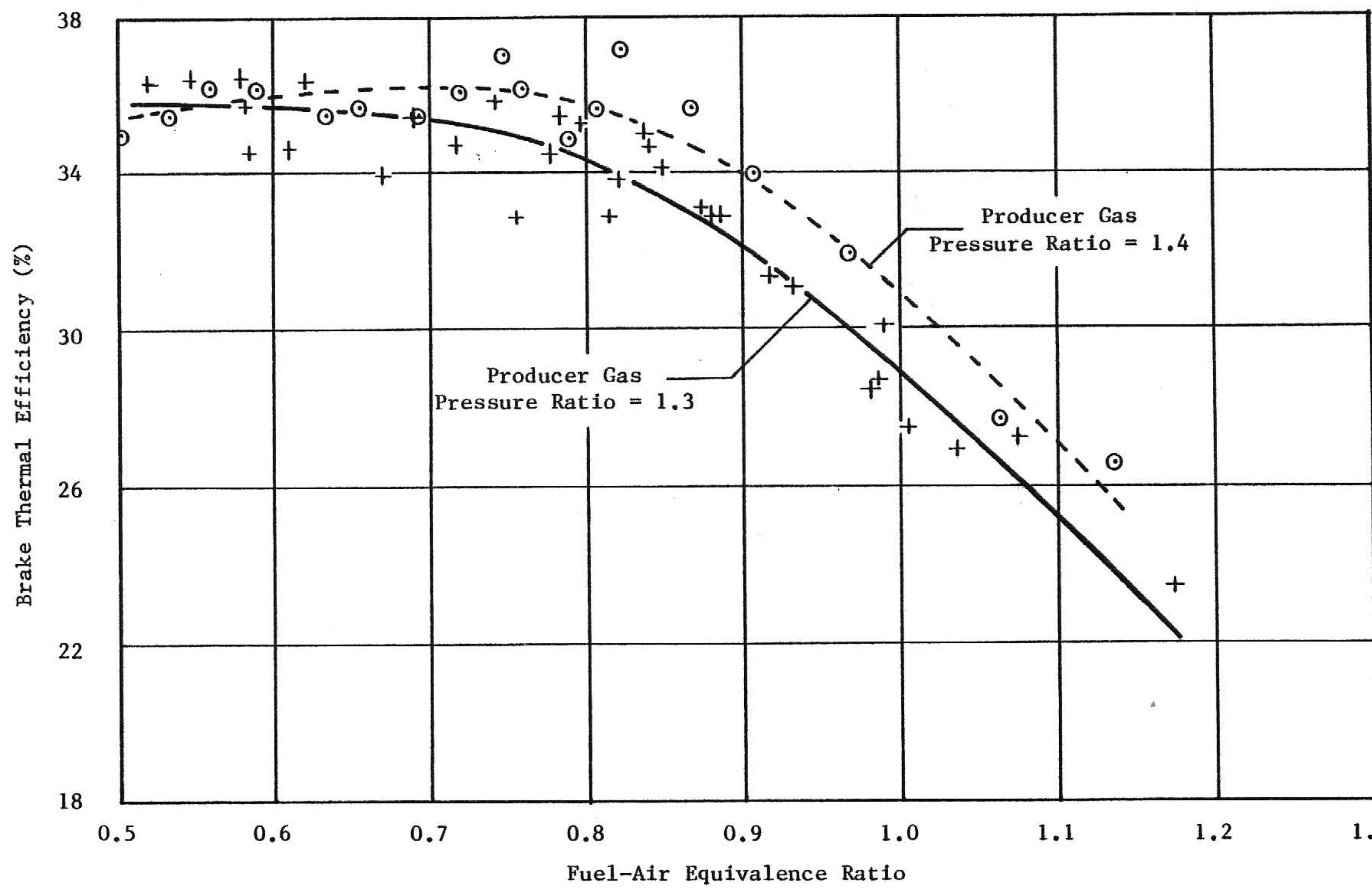

Figure 21. Supercharged Brake Thermal Efficiency versus Equivalence Ratio at 2200 RPM.

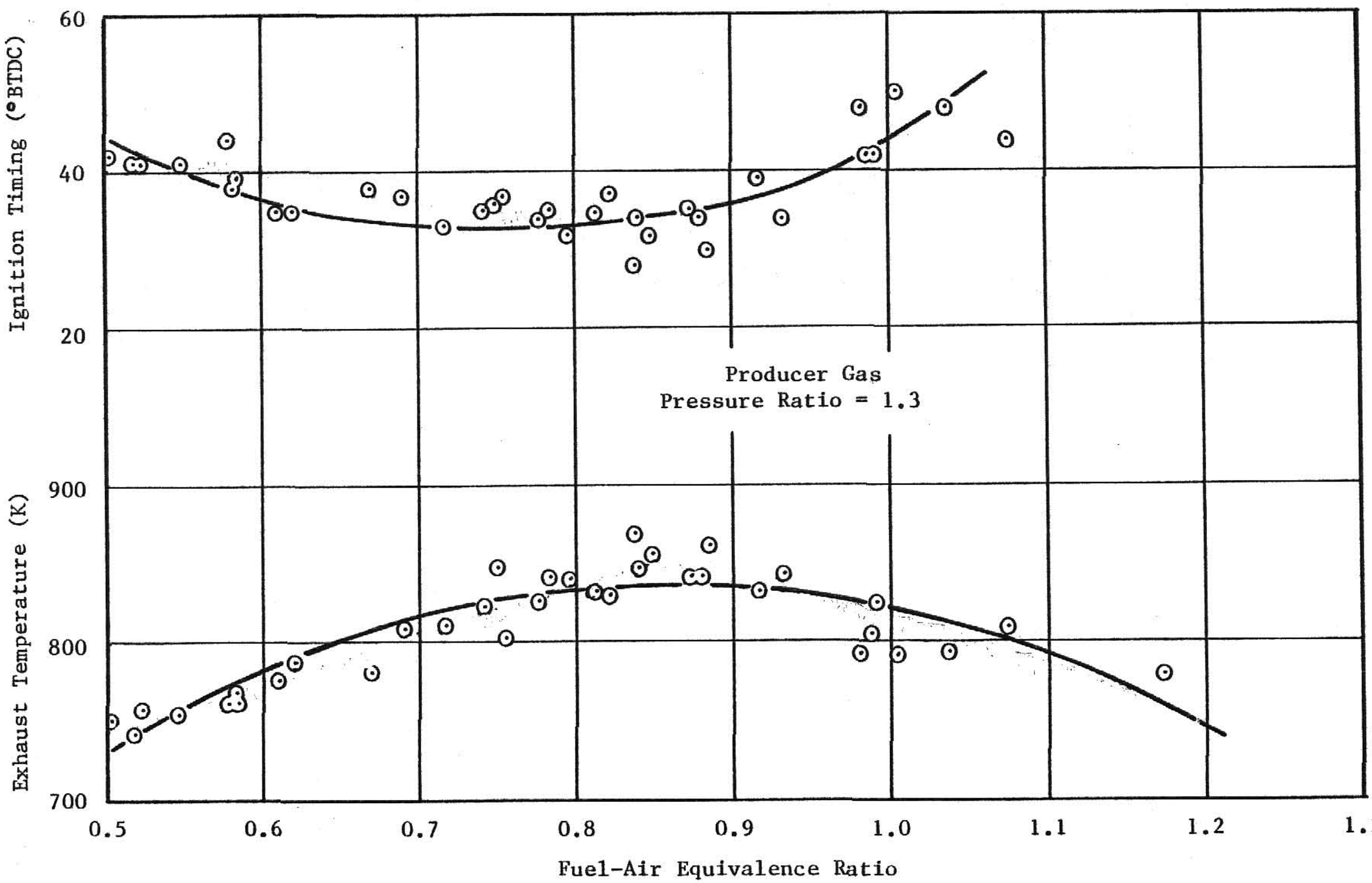

Figure 22. Exhaust Temperature and Ignition Timing for 1.3 Pressure Ratio at 2200 RPM.

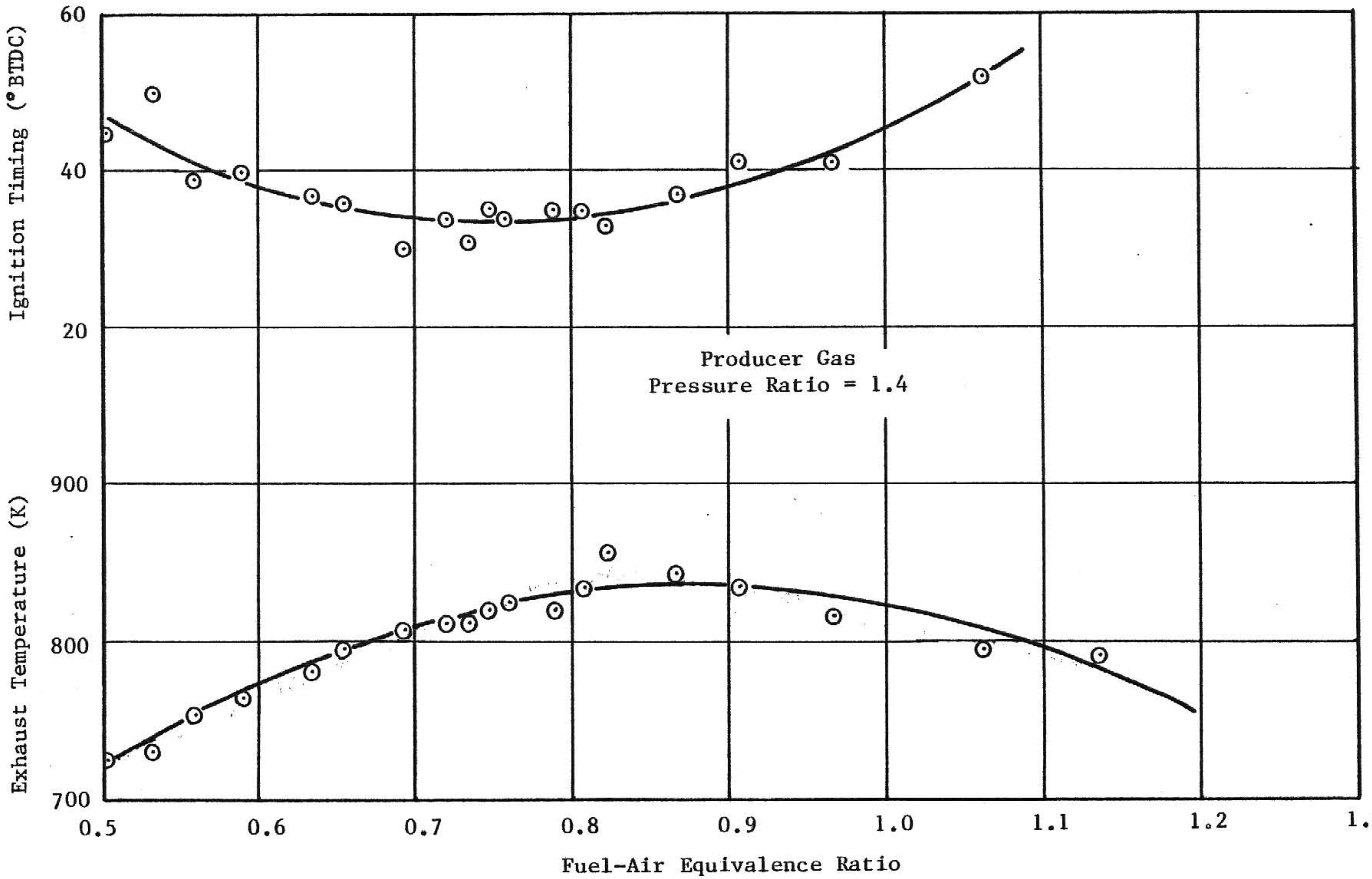

Figure 23. Exhaust Temperature and Ignition Timing for 1.4 Pressure Ratio at 2200 RPM.

APPENDIX B

QUADRATIC EQUATIONS DESCRIBING ENGINE PERFORMANCE

Predicted Value = $Ax^2 + Bx + C$, where

A = second order coefficient

B = first order coefficient

C = intercept

x = fuel-air equivalence ratio

Predicted Value = Power (kW), or

Brake Thermal Efficiency (%), or

Ignition Advance (°BTDC), or

Exhaust Temperature (K).

Table 7. Coefficients and Intercepts for Quadratic Equations

Engine Speed	Inlet Condition	A	B	C	R-Square
		Corrected Brake Power			
2200	Naturally Aspirated	-10.29	18.20	-0.593	.682
2200	1.3 Atmospheres	-18.02	30.85	-2.804	.734
2200	1.4 Atmospheres	-17.67	30.67	-2.120	.939
2600	Naturally Aspirated	-15.69	29.24	-5.568	.664
		Brake Thermal Efficiency			
2200	Naturally Aspirated	-26.85	28.12	24.73	.928
2200	1.3 Atmospheres	-40.23	48.02	21.55	.907
2200	1.4 Atmospheres	-49.25	66.85	13.65	.910
2600	Naturally Aspirated	-49.51	73.08	1.29	.663
		Ignition Advance			
2200	Naturally Aspirated	168.8	-282.4	159.9	.764
2200	1.3 Atmospheres	188.8	-285.3	140.5	.801
2200	1.4 Atmospheres	206.2	-312.7	152.1	.921
2600	Naturally Aspirated	113.0	-182.2	112.2	.211
		Exhaust Temperature			
2200	Naturally Aspirated	-583.9	1072.	313.1	.658
2200	1.3 Atmospheres	-806.5	1392.	236.9	.782
2200	1.4 Atmospheres	-815.2	1434.	205.2	.943
2600	Naturally Aspirated	-593.0	1131.	262.5	.628

BIOMASS PRODUCER GAS FUELING
OF SPARK IGNITION ENGINES

by

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B.S., Kansas State University, 1979

AN ABSTRACT OF A MASTER'S THESIS

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MASTER OF SCIENCE

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Since about 80% of the total energy consumed for irrigation in Kansas is derived from natural gas, irrigators are particularly vulnerable to any gas supply curtailments. Among several alternatives presently being investigated to supplement fossil fuels are biomass based fuels. One of these is a low energy gas produced by the gasification of crop residues.

A pilot plant fluidized bed gasifier, using corn stover as a feedstock, has been used to fuel a spark ignition engine. Engine performance when fueled with pipeline quality natural gas served as a basis for comparison. The most evident result of using the low energy gas (producer gas) was that engine power output dropped. The decrease in power output was almost entirely accounted for by the smaller amount of potential energy inducted in the air-fuel charge. The drop in the air-fuel mixture heating value was caused by a high level of diluents delivered by the gasifier along with the useful fuel components.

In addition to lowering the fuel heating value, the presence of the diluents resulted in a lower exhaust temperature and a slower flame speed. The slower flame speed necessitated an increased ignition advance.

Large quantities of this dilute fuel were consumed by the engine; 30-50% of the air-fuel mixture was occupied by fuel. When fueled with natural gas, 9-10% of the mixture was occupied by fuel. Because of this, a commercial gaseous fuel carburetor could not be used with producer gas. In addition to a carburetion system adequate for laboratory testing, a method was developed to convert the unsteady, pulsating output of the fluidized bed gasifier to a steady, constant pressure flow.

Supercharging was used to regain natural gas power output. A boost pressure of 0.4 atmospheres was adequate for this purpose. Supercharging also reduced the required ignition timing, but not to natural gas values, and increased exhaust temperature, but again, not to natural gas temperatures.

Gas cleaning was a major problem. Unlike earlier research efforts, particulate removal was no problem. However, the producer gas contained a tar-oil mist which was not removed by packed column filters. Although the mist may have contributed to the power output of the engine, serious deposits developed on the intake valves and in the combustion chamber.