# BEEF AND SWINE DIGESTER GASSES-
# EVAULUATION AS FUELS FOR
# SPARK IGNITION ENGINES

by

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# INTRODUCTION

The oil embargo of the 1970's revealed the dependence of the industrial world on a potentially unstable source of petroleum fuels. Since that time, increased emphasis has been placed on alternate and renewable fuel sources to lessen the impact of another disruption of petroleum supplies.

One of the sources of these alternate fuels that is currently being studied and developed includes biogas produced by anaerobic digestion of livestock wastes and crop residues.

Numerous papers have been written on the theory and actual operation of anaerobic digestion and anaerobic digesters which use mesophilic and thermophilic bacteria to break down the complex wastes and residues to form methane and carbon dioxide. Comparatively few studies available have researched the use of the digester gas as an alternate fuel for internal combustion engines.

Anaerobic digestion of agricultural wastes and residues can provide relatively high energy gas for heating and fueling purposes. Engines, utilizing digester gas, can be used to power other systems such as irrigation pump engines or engine generators to provide electrical power for a variety of uses.

Previous studies have shown that while mobile vehicles could use the biogas as a fuel, most vehicles studied lacked adequate storage capacity for extended operation. For this reason it is assumed that most applications using the digester gas will be stationary units.

The purpose of this study was to determine the necessary engine modifications to optimize engine performance and to identify and solve some of the problems associated with using the digester gas as an engine fuel.

# REVIEW OF LITERATURE

## Methane as an Engine Fuel

Neyeloff and Gunkel(1975) reported that the use of methane as a fuel for internal combustion engines has several advantages. It has a high octane rating which gives excellent anti-knock qualities as well as providing for more efficient combustion in engines if the compression ratio is increased. Gaseous fuels generally cause minimum engine carbon deposits. The gaseous fuels also mix more thoroughly with air to burn more completely than liquid fuels. Other advantages include a small amount of contaminating pollutants, less sludge in the lubricating oil, no wash down of cylinder wall lubrication during engine starting, no tetra-ethyl lead to foul spark plugs and other engine parts, a nearly homogeneous mixture in the cylinder, and less valve burning.

Flammability limits of methane in air are 5.3-14% by volume with the ideal stoichiometric mixture of about 9.4%. Neyeloff et al. (1975) reports that the ignition limits of $CH_4$ in air are affected by carbon dioxide dilutions and that under atmospheric conditions the mixture will not combust if the amount of $CO_2$ is greater than three times the amount of methane. The ignition limits should be affected very little by the temperature and pressures developed in an engine. Yet it was found experimentally that ignition occurred at fuel to air ratios of 0.065 to 0.185, both limits higher than those reported at atmospheric conditions.

Neyeloff et al. (1975) reported that an ignition advance of 30 degrees before top dead center (BTDC) gave the highest efficiency for their CFR engine operated at 900 rpm. They did state that at various times during the test runs that ignition timing was slightly varied and little, if any, changes in output were found. They reported that specific power output peaked at a 15:1 compression ratio but that at that compression ratio and above, audible knocking occurred. The specific power output was the highest at a fuel to air ratio of 0.10. A comparison was made using two compression ratios and different methane dilutions. If a compression ratio of 15:1 and percent dilution of 0% were given a score of 100 for specific power output, then a 15:1 compression ratio using a 50% dilution (67% methane and 33% carbon dioxide) would have a score of 79. Compression ratios of 7.5:1 with 0% and 50% dilutions would get scores of 73 and 60 respectively for specific power output.

Clark, Koelsch, Whitmar, and Walawender (1978) conducted an economic analysis on the use of digesters to power irrigation engines. Advantages of the digester system were almost 100% recovery of residue nutrients, production of relatively high energy gas, and negligible environmental impact. Disadvantages included 1.) a need to distribute about 90% of the original solids remaining after treatment, 2.) digesters are not suitable for fluctuating load demands without storage, and 3.) digesters are not suitable for rapid starts or shut-downs. Additionally, digestion is subject to a variety of instabilities as a result of temperature change, pH variations, carbon-nitrogen ratio, feed rate changes, and the presence of toxic materials.

The study was based on the operation of a digester for a three-month irrigation season. The economics improved slightly if the digester was utilized in crop drying in addition to irrigation. Clark et al. (1978) concluded that due to the high cost of gas storage facilities, a larger digester would be less expensive and that any excess gas produced could be flared.

Tests conducted by Stahl, Fischer, and Harris (1982) used a heat recovery system on an engine to heat water for the digester and for alcohol production. It was found that the thermal output of the engine is strongly dependent upon engine power output.

## Engine Performance

### Low Compression Spark Ignition

Stahl, Harris and Fischer (1982) operating a 3.6 liter engine at 1260 rpm produced 21 kW using digester gas with a 55% methane content. Minimum brake specific fuel consumption occurred at a manifold vacuum of 5 cm Hg and an equivalence ratio in the range of 0.6 to 0.8 while peak power was attained at an equivalence ratio of 0.825 and a manifold vacuum of 5 cm Hg. As the throttle was closed, maximum power for the throttle setting was achieved at equivalence ratios approaching 1.0. As the throttle was closed, the spark was advanced for optimum engine performance. The researchers also found that as the fuel-air mixture became richer, the spark must be retarded from a maximum of $50^{o}$ before top dead center(BTDC) to $45^{o}$ or even $40^{o}$ BTDC.

In two direct comparison tests performed by Stahl, Harris, and Fischer (1982), the engine had slightly higher efficiency when burning biogas coming from compressed storage tanks than biogas directly from the digester due to the lower water content of the compressed gas.

Koelsch (1982) reported that for an engine supplying power to a generator with an equivalence ratio greater than 1.2 that misfiring became a problem while with an equivalence ratio below 0.78 that the desired electrical output could not be maintained. Electrical efficiency peaked between equivalence ratios of 0.78 to 1.0. An equivalence ratio of approximately 0.85 provided the most efficient generation of electricity.

The test engine used by Koelsch et al. (1982) was run for 1220 hours on unscrubbed biogas with an average $H_2S$ content of 0.4%. The high sulfur content caused rapid deterioration of the buffering capacity of the oil resulting in short oil change intervals. The capacity of the oil to neutralize the effects of sulfur in fuel is indicated by the Total Base Number (TBN). Initially an oil with a TBN of 6 was selected, but after 94 hours of operation the TBN was 0.46, which was far below the condemnation level of 2.0. The use of oils with a TBN of 10 was not sufficient for unscrubbed biogas, slipping below 2.0 at 55 hours of operation and 0.46 TBN at 208 hours.

After 1220 hours of operation, all four rod bearing sets displayed pitting while one of the bearings exhibited severe pitting of the bearing surface and flaking out of the bearing surface material. The pitting and flaking was attributed to corrosive attack on the bearing surfaces by excessive acidity in the oil. An oil change interval of 250 hours was used once, while the other oil changes were done at 150 hours or less.

Another problem was a slightly excessive wear rate in some of the valve guides, which was attributed to the lack of lubrication in the fuel and the lack of oil reaching the area between the valves and guides. The other engine components were reported to be in excellent condition and were reinstalled in the engine.

The original spark plugs, with a nickel alloy electrode, decayed rapidly, increasing the plug gap from 0.030 in. to 0.045 in. in 100 hours of operation. Plugs with an inconel electrode base were installed and after 400 hours of engine operation, the gap had increased a maximum of 0.005 in. Five hundred hour plug change intervals were used with no apparent problems. Spark plug selection had a minor effect on engine operation with hotter plugs to be somewhat preferred. Standard plug gaps of 0.020 to 0.030 inch give satisfactory performance.

All spark ignition engines exhibit dramatic reductions in fuel economy under rich fuel-air mixtures or part load conditions. At full load, an additional 0.21 $m^3$ (7.5 $ft^3$) of biogas was needed to produce one kilowatt hour of electricity at an equivalence ratio of 1.1 as compared to 0.85. Koelsch et al. (1982) reported that their unit consumes between 0.74 and 0.77 $m^3$ (26 and 27 $ft^3$) of biogas per kW-hr of electricity produced.

Recommendations for lubrication oils for digester gas engines include oils with a high TBN rating. There are, however, some trade-offs to consider since high TBN oils also have high ash contents which can cause another set of engine problems. It was also recommended that engine oil and coolant temperatures be maintained above $210^{o}$ F to prevent moisture condensation in the oil or on the cylinder walls.

According to Koelsch et al. (1982) spark timing for an 1800 rpm generator engine should be $40^{o}$ BTDC or greater, while at part loads spark advance may be retarded. It should not, however, be retarded beyond $30^{o}$ BTDC. Koelsch (1984) also indicated that the spark advance values given were the minimum advances to achieve maximum power. It was indicated that advances of perhaps an additional $15^{o}$ showed little change in power output. These values do seem to be the reverse of the manner that most engines with vacuum advance are set up, indicating that special distributors may be needed for digester gas engines. A vacuum advance was not used in this test.

## High Compression Spark Ignition

Persson and Bartlett (1981) state that Penn State had built a high-compression spark ignition engine for biogas by replacing the injectors of a diesel engine with spark plugs and modifying the pistons. Preliminary tests on natural gas indicated higher thermal efficiency for this engine than for lower compression and dual-fueled engines. The efficiency was close to that of the diesel fueled engine while the maximum available power increased considerably over the diesel version.

Dual Fuel Compression Ignition

Persson et al. (1981) report that diesel engines can be used for biogas with modifications if some diesel fuel is used for ignition purposes. The original injection system can be maintained as well as the diesel's high compression ratio if the engine is dual fueled.

A constant minimum amount of liquid fuel must be injected for ignition, independent of the power output of the engine. If the injection timing was advanced from its normal $17^{\circ}$ advance to $24^{\circ}$ advance an increase in efficiency was obtained. The thermal efficiency for the dual-fuel engine was less than for diesel fuel.

The normal minimum amount of diesel fuel that could be used before irregular ignition occurred was constant for all tests and independent of load and injection timing. The amount of diesel fuel needed for proper ignition depends on the design of the fuel system and the engine design. At 100% load and maximum efficiency, the liquid fuel delivered less than 10% of the necessary fuel energy while at 10% load, the diesel supplied 40% of the energy. Since it is desirable to use mainly the biogas for power, the engine should operate as close to full load as possible. Persson et al.(1981) also reported that the dual-fuel engine operated quieter than did the 100% diesel engine.

Persson and Bartlett (1981) report that some of the water vapor in the gas should be removed to prevent condensation prior to use as fuel, as this will aid in preventing fouling of the gas lines and valves. It may also be desirable to maintain a fairly high engine temperature, even when not in operation to prevent water vapor condensation. Reductions of $H_2S$ concentrations to approximately 1 mg $H_2S$ per liter was also recommended. It was noted that high-strength steel seems to be more susceptible to hydrogen sulfide attack than cast iron or lower grade steels.

Ortiz-Cañavate, Hills, and Chancellor (1981) used a conventional diesel engine as a dual-fuel engine using diesel and synthetic biogas (natural gas and carbon dioxide mixtures). Twenty-three degree injector advance instead of the the normal $19^{o}$ advance seemed more favorable to avoid knocking and misfiring at low torque levels or high engine speeds.

The air ratio, the actual air used to stoichiometric ratio, was 1.2:1 for dual fuel as compared to 1.5:1 for diesel. Efficiency of the dual fuel engine when operating at high torque levels and low engine speeds was equivalent to that of the engine operating solely on diesel fuel. At higher speeds, the air-fuel mixture was less than stoichiometric and efficiency dropped while at low torques, the efficiency of the dual- fuel was much lower than the diesel and the engine did not run smoothly.

## Digester Gas Clean-up

Digester gas cleanup is important from several aspects. It may be necessary to treat the gas to meet pipeline specifications if the gas is to be used commercially. Pipeline specifications are important from the standpoints of uniform heating value, corrosiveness, and efficiency of transportation and heating. Digester gas usually will have four constituents; methane, carbon dioxide, hydrogen sulfide, and water vapor. Of these the hydrogen sulfide and the water vapor cause serious corrosion problems for pipelines, valves, furnaces, and engines. These constituents should be removed as soon as is feasible. The carbon dioxide is generally non-corrosive except when absorbed in water.

Dynatech (1978) conducted a study of different systems for the removal of water vapor, $H_2O$ and $CO_2$. The study was conducted for three raw gas flowrates and for two end delivery pressures. The flowrates were 100,000 standard cubic feet per day (SCFD), 1 million (MM) SCFD, and 3MMSCFD; the 0.1MMSCFD corresponds to the expected output from a 2000 head beef feedlot. The end pressure of the sweet (clean) gas was 125 psi for intrastate pipeline transmission and 1000 psi for interstate pipelines. No credit was assumed for sale of carbon dioxide or hydrogen sulfide, although it was estimated that as much as 70 tons per day of carbon dioxide could be recovered from the 3MMSCFD flow. The recovery of the hydrogen sulfide was felt to be economically unfeasible with production of 4.5, 45, and 134-lbs. per day from the three flow rates respectively.

Pipeline specifications require that the gas be at least 97% methane, less than 3 mole percent carbon dioxide, less than 0.25 grains per 100 SCF (0.00036%) $H_2S$, and water vapor shall not exceed seven pounds per MMSCF. The heating value of the gas shall not be less than 950 BTU per SCF.

After examining the expected costs of different systems now available, Dynatech decided that the water scrubbing process for carbon dioxide removal, hydrogen sulfide removal with a process developed by Eickmeyer & Associates, and water vapor removal with triethylene glycol (TEG) would be the most economical. Many of the systems now used by the commercial gas industry were developed for treatment of much larger quantities of gas and were therefore not economical to be scaled down for use of biogas cleaning.

For the Dynatech system, the costs for the three flow rates, 0.1, 1, and 3MMSCFD, were \$1.943, \$1.169, and \$.780 per million BTU at 130 psi delivery pressure and \$1.982, \$1.178, and \$.790 respectively, per million BTU at 1020 psi delivery.

The water scrubbing process has been used for centuries, and uses water to absorb carbon dioxide. The process works best at high pressures (above 300 psia is recommended) and at low temperatures. It is quite dependent on the partial pressure of the

$CO_2$. Dynatech suggested the use of a lagoon to desorb the carbon dioxide at atmospheric pressures while other schemes have used a series of flash desorbers so that the $CO_2$ may be recovered. If carbon dioxide recovery is desired, there are other commercially available methods that are more efficient.

The Eickmeyer process for hydrogen sulfide removal uses an undisclosed proprietary chemical solvent. The sour gas is bubbled through the solvent in a shallow tank at atmospheric pressure with almost complete removal of $H_2S$. The solvent is highly selective for $H_2S$ The amount of solution is on the order of 1000 gal/MMSCFD of gas flow. The process is non-regenerative.

Water absorption uses either diethylene glycol (DEG) or triethylene glycol (TEG). Advantages of this process are unusual hygroscopicity of the solution, good stability to heat and chemical decomposition, low vapor pressures, and ready availability at moderate cost. In a typical unit, water vapor is continuously absorbed from the process gas stream by countercurrent contact with a highly concentrated glycol stream (95 to 99%) in a packed or bubbly tray column. Dynatech reported that that they based their review on a system that had a TEG concentration of 99.8 weight percent for low pressures and for high pressure, a TEG concentration of 98.7 weight percent. Regeneration of the rich glycol solution is usually accomplished by inert gas stripping with the application of heat. When maximum dehydration is required in large installations, vacuum regeneration can also be used.

# INVESTIGATION

## Objectives

The objectives of the investigation were:

1. To compare the performance of the engine operation on two different sources of anaerobic digester gas with natural gas.

2. To determine the effect of the fuels on the engine components.

3. To determine the changes in spark advance needed to optimize operation.

## Theory

Patterson et al. (1972) report that the combustion process occurs in two steps, the ignition delay and the pressure-rise period. The ignition delay period is the time required for a fuel and air mixture to auto-ignite when subjected to a high temperature at a given pressure. The length of the ignition delay period depends upon charge pressure, fuel-air ratio, and octane number of the fuel. Ignition delay is independent of engine speed; thus, at high engine speeds the ignition delay occupies a larger number of crank angle degrees than at lower speeds.

The pressure-rise period depends on the type of fuel and the turbulence, among other factors. Since the flame velocity is proportional to the turbulence which in turn is proportional to engine speed, the pressure-rise period occupies about the same number of crank angle degrees at different speeds.

Methane is a slow burning fuel, therefore, the timing advance of engines fueled with methane needs to be increased to maximize the engine output power. If the spark advance is not advanced enough, the fuel will not have burned completely before being exhausted through the exhaust valves. This exposes the exhaust valves and manifold to higher temperatures and thus the expected life of these components is shortened. If the timing is too advanced, the peak pressure rise will occur before top dead center. Knocking may occur and performance suffer as a result. Dilution of methane with inert gasses such as carbon dioxide reduces the flame velocity of the methane mixtures. This dilution also causes the need for greater spark advance.

## Materials and Experimental Equipment

### Test Fuels

The fuels tested were natural gas, anaerobic digester gas derived from beef manure, and anaerobic digester gas derived from swine manure. The natural gas was supplied by Kansas Power and Light Company at Kansas State University in the Agricultural Engineering's Engine Test Laboratory. The beef-based biogas was produced at the U.S.D.A. Roman L. Hruska Meat Animal Research

Center at Clay Center, Nebraska while the swine-based gas was made available at the Del Valle Hog Farm at Austin, Texas.

The biogas at both locations was characterized by a 50 to 65% methane content, with carbon dioxide concentrations of 35 to 50%; the remainder of the constituents being hydrogen sulfide and water vapor.

Test Equipment

Test Engine The test engine was a Ford industrial engine, Model KSG-411. The engine was equipped with an Impco CA50-510 carburetor designed for operation on natural and low energy gas. The engine was originally equipped with a larger carburetor to allow for a greater amount of low energy biogas to enter the engine. Engine performance was not up to expectations and analysis of the exhaust gasses revealed high oxygen and low carbon monoxide levels indicating a very lean fuel-air mixture. The original carburetor was not properly matched to the engine. It was designed for a larger displacement engine. The carburetor fuel jets had the capability of providing adequate gas for the small engine but a large air flow was needed to provide the pressure drop across the air venturi to obtain the respective gas flow. This resulted in poor performance from a lean mixture. A solution to this problem proved to be a smaller carburetor with increased gas supply pressure to supply the needed quantity of hydrocarbons per given volume. The engine specifications are shown in Table 1.

Table 1. Ford KSG-411 Test Engine Specifications

| Specification | Value | Units |
|---|---|---|
| Cylinders | 4 | - |
| Bore | 80.98 | mm |
| Stroke | 53.29 | mm |
| Displacement | 1.1 | liters |
| Compression Ratio | 8:1 | - |
| Rated Speed | 1500-3600 | rpm |
| Power(continuous on gasoline) | 18 | kW |

The engine is equipped with a fan and radiator, alternator, and a muffler. A Rockwell PTO clutch and an instrument panel were installed. The panel featured a starter key switch, a coolant temperature gauge, an oil pressure gauge, a voltmeter, and a vernier type throttle control.

Dynamometer and Instrumentation. A model 70523 Cessna pump equipped with a Parker-Hannifin R10PH-11-BL pressure control valve was used as a dynamometer load for all the tests. A 90.7 kg (200-lb.) Lebow 3167 load cell installed on an 279.4 mm lever arm was used to determine torque. The load cell signal was received by a Daytronic 3270 strain gage conditioner and indicator. Engine speed was determined by a 60-tooth gear and an Electro 3010AN magnetic pickup, with the signal received by a Daytronic 3240 frequency conditioner and indicator.

The digester gas was filtered through two Winslow gas filters (Model 981136-B) arranged in parallel to avoid overloading the filters. A Rockwell 415 gas meter and indicator was used in the first series of tests to provide a visual indication of gas consumption. A magnetic reed switch was used to provide a pulse signal to the data acquisition system to determine the rate of gas use.

Copper-constantan thermocouples were used to measure the temperatures of the oil, the fuel, the inlet and outlet coolant, and ambient intake air. The exhaust gas temperature was measured with an iron-constantan thermocouple.

For the second series of tests, absolute fuel and atmospheric air pressures were measured by Setra Systems Inc. Model 204 0-345 kPa (0-50 psi) variable capacitance absolute pressure transducers.

Air flow measurement used a 31.75 mm (1.25 in.) diameter ASME nozzle mounted in a 114 liter (30 gal.) plenum chamber. The large plenum chamber was needed to smooth the pulses caused by the cyclic intake of air. The pressure drop across the nozzle was measured with a Setra Systems Inc. Model 239 0-1.4 kPa (0-0.2 psi) variable capacitance differential-pressure transducer. The fuel flow measurement involved measuring the pressure drop across a 12.7 mm (0.5 in.) diameter ASME nozzle. This pressure drop was also measured with a Setra Systems Inc. Model 239 differential-pressure transducer. The differential transducers were calibrated against a Meriam Model 34FB2 micromanometer. The discharge coefficient for the flow nozzles was determined by using a relationship proposed by Benedict (1966), relating the most probable discharge coefficient to the Reynold's number. The fuel and air supply system is illustrated in Figure 1.

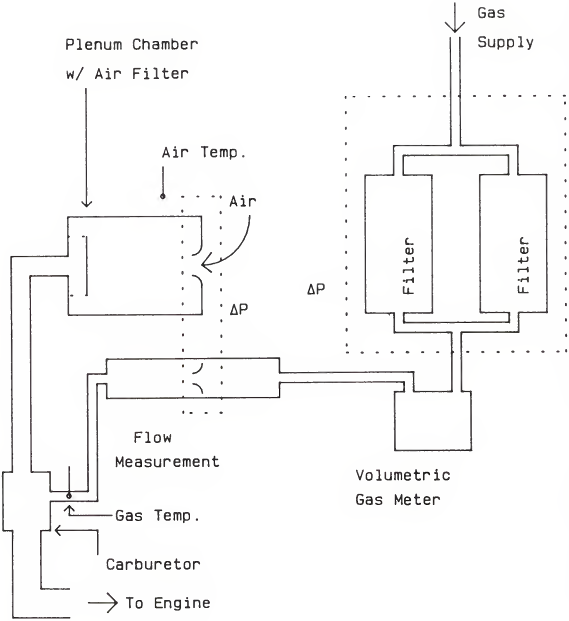


Figure 1. Fuel and Air Supply Instumentation

The gas and air measurement system used performed well at low speeds, but at engine speeds of approximately 2000 rpm a resonance frequency apparently appears and the indicated gas and air use is much lower than is realistic. Specific fuel consumption and thermal efficiencies were unreasonably low and high, respectively, and will not be included in the results.

The data acquisition system used a Synertek Systems Corp. SYM 1 computer to collect data. The test data was stored on cassette tape until the data could be processed on a DEC PDP 11-34 computer.

The SYM 1 featured a 16-channel, analog-to-digital board with two pulse counting channels to facilitate data collection. Data was taken at 15-second intervals and stored in three minute blocks which could be averaged using Fortran software. The Fortran program is included in Appendix B.

## Test Procedures

An engine break-in period of 50 hours was used prior to operating the engine on biogas. A 90-minute break-in cycle proposed by the Engine Manufacturer's Association (EMA) to the United States Department of Agriculture was used with commercial natural gas to fuel the engine. The break-in cycle was as follows:

1. 10 minutes at low idle
   (1200 rpm)

2. 10 minutes at 1/2 rated speed and no power

(1440 rpm)

3. 15 minutes at 3/4 rated speed and 1/2 rated power
(2160 rpm and 17.6 N-m)

4. 55 minutes at rated speed and rated power
(2880 rpm and 35 N-m)

The actual engine tests consisted of a 200-hour cycle also recommended by EMA. The test cycle was as follows:

1. 30 minutes at low idle
(1400 rpm)

2. 60 minutes at rated speed and rated power
(2880 rpm and 35 N-m)

3. 60 minutes at peak torque
(2400 rpm and 36.9 N-m)

4. 30 minutes at 80% rated speed and 25% rated power
(2300 rpm and 11.6 N-m)

For the first series of tests, due to the availability of the biogas, two three-hour cycles were run per day with an average of 25 hours of engine operation per week. Torque curves were run every 25 hours and performance was mapped at the beginning, middle and end of the 200 hour test. Data was taken at 200 rpm intervals and values for the performance map at intervals of one-tenth full torque. The engine was run twenty-five hours on natural gas prior to the second test to purge the engine of the first test's residuals. During the second test series, the engine was run for five three-hour loading cycles for a total of fifteen hours per day. The two-hundred hour test was completed in approximately two weeks. Collection of data was conducted the same as for the first series of tests.

Oil samples were collected at 25-hour intervals and sent to Farmland Industries to check for wear metal levels and oil deterioration. The oil and filters were changed at 100-hour intervals.

## Results

### Engine Performance on Natural Gas

The engine power peaked at 16.0 bkW, 89.5% of the rated power of the engine on gasoline. Peak power occurred at 3600 rpm. The peak torque, 54.2 N-m, occurred at 2400 rpm. The torque and power curves are illustrated for reference to the biogas tests in Figures 2 through 5.

### Test Series 1: Engine Performance on Beef Manure Biogas

Eight torque curves were conducted while operating on the beef-based biogas. The curves varied depending on the biogas properties at that time. The methane content was usually about 53 percent although fluctuations of up to 7 percent were seen in successive weeks. The gas pressure also was subject to small variations. Average torque and power curves were calculated from the eight torque curves. The results are plotted in Figures 2 and 4, respectively, as are the 95% confidence limits for the expected limits of day to day operation. The peak power of the average curve was 12.3 bkW, 77% of the peak natural gas power.

However, the highest observed power was 13.3 bkW, 83% of natural gas power. The highest value on the average torque curve was 38.7 N-m which occurred at 2000 rpm while the highest observed torque, 41.45 N-m, was at 2200 rpm.

Test Series 2: Engine Performance on Swine Manure Biogas

Eight torque curves were run at 25-hour intervals while the engine was fueled by swine-derived digester gas. The methane content of the gas was consistently 64 percent with only occasional slight variation of a maximum two percent change. Average torque and power curves with 95% confidence limits were calculated from the eight torque curves. The results are plotted in Figures 3 and 5. The peak power of the average curve was 15.0 bkW, 94% of the peak natural gas power and 122% of the peak average beef-derived gas. The highest power observed was 15.2 bkW, 95% of the peak on natural gas. The highest torque of the average curve was 41.6 N-m which occurred at 2000 rpm while the highest observed torque, 43.3 N-m was also at 2000 rpm. The peak on the average torque curve was 77% of the peak on natural gas and 107% of the peak of the average curve of the first test series.

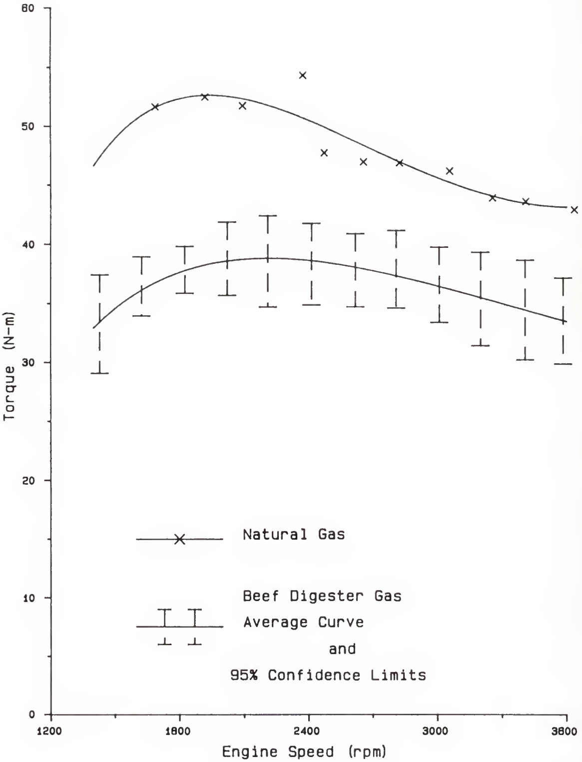


Figure 2. Torque Curve: Test Series 1

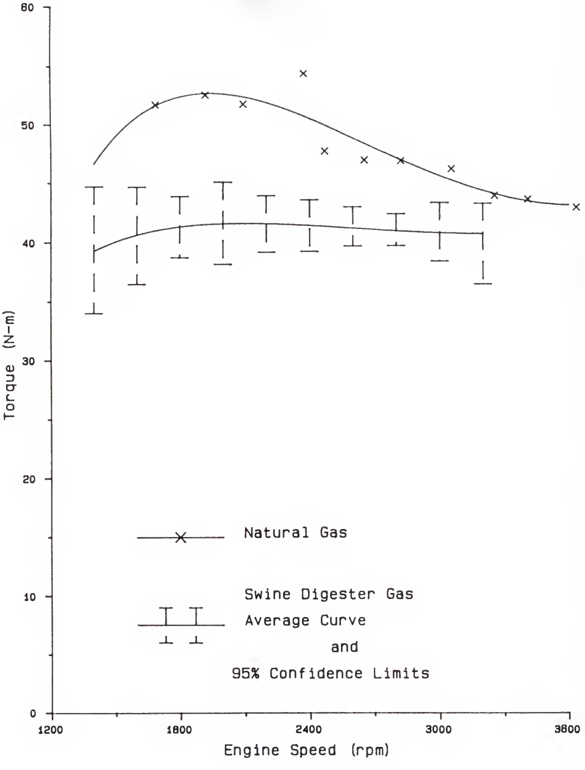


Figure 3. Torque Curve: Test Series 2

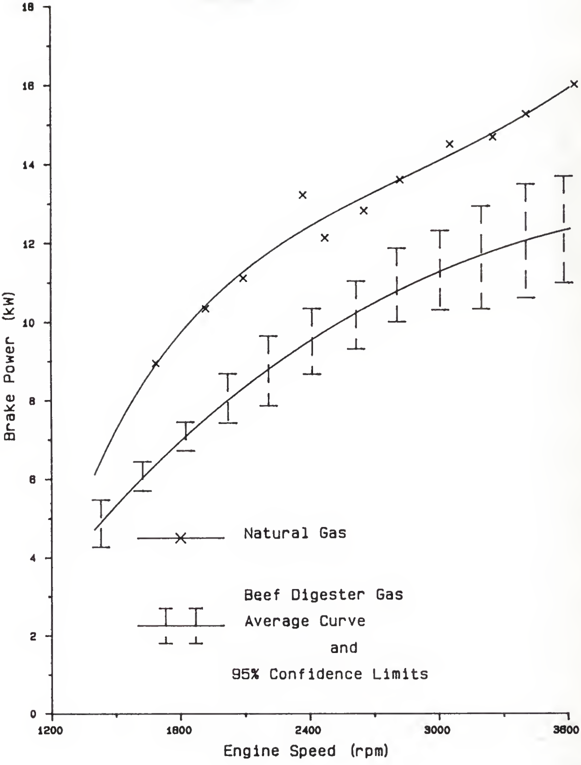


Figure 3. Power Curve: Test Series 1

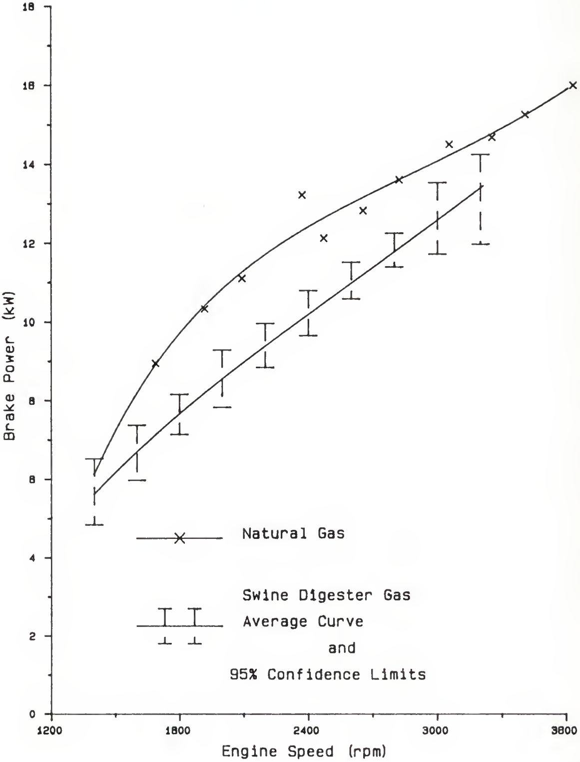


Figure 5. Power Curve: Test Series 2

# DISCUSSION

## Engine Performance

The hydrocarbon content of the fuel, the amount of water in the gas, and the fuel pressure appear to be the controlling factors in the engine performance. The biogas supply pressure was quite important. If the pressure was less than 750 mm $H_2O$, the performance was sluggish. At times in the first test series, during the 3 hour engine cycle, water vapor in the gas would reduce the engine output enough that the dynamometer load would kill the engine. At those times the engine could operate at 1800 rpm with a maximum torque setting of approximately 10 N-m. There were no problems with water vapor at the Texas facility. This can probably be attributed to the difference in the storage tank size between the two facilities. The Nebraska facility used two 1000 gallon propane tanks to hold the gas production while the Texas facility used a 60,000 gallon butane tank for storage. The larger tank would have less turbulence allowing the water vapor to condense and settle out of the gas. The tank could then be drained of the condensed water periodically. The methane content of the gas changed engine performance very little. A 5% drop of methane content gave only slightly noticeable performance deterioration.

## Spark Advance

Shown in Figure 6 is the spark advance of the engine operating on digester gas at full torque. The spark advance shown is

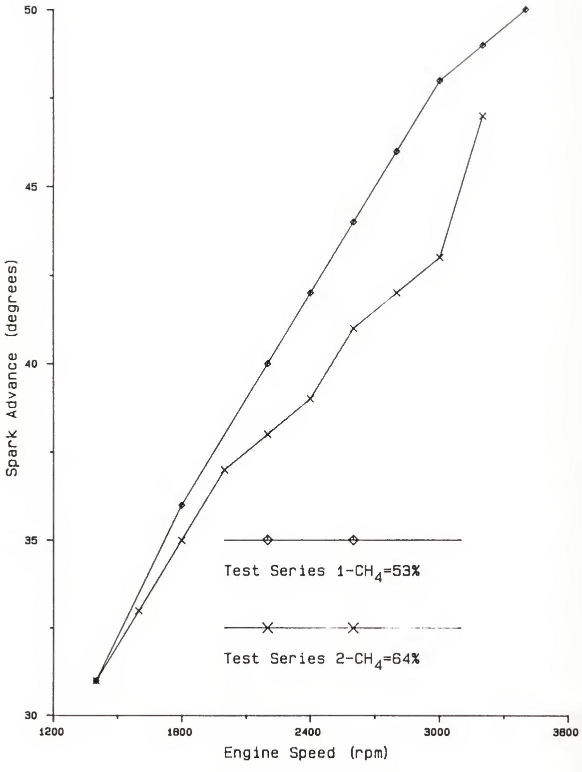


Figure 6. Spark Advance: Test Series 1 and 2

the minimum advance for best torque. The difference in the two curves is due to the difference in methane content of the two biogas supplies. The richer gas from the swine has less carbon dioxide dilution, resulting in less slowing of the flame velocity in the combustion chamber. If the advance is increased beyond these values, little change in power was detected.

## Engine Oil Analysis

Oil samples were collected at 25-hour intervals and were analyzed by Farmland Industries for wear metal concentrations. Metal concentrations (ppm) that are detected are iron, chromium, aluminum, copper, lead, tin, silver, nickel, silicon, sodium, boron, magnesium, calcium, barium, phosphorus, and zinc. The oil viscosity at 100 C, the presence of water and antifreeze in the oil, total solids, and oxidation were also monitored. Iron concentrations indicate cylinder wall wear. Chromium indicates ring wear. Aluminum points to piston wear. Copper, lead, and tin indicate bearing wear, while silicon indicates that dirt or sand is being induced into the engine usually from a leaking air cleaner. Total Base Number (TBN) was monitored during the second test series. TBN is an indicator of the remaining buffering qualities of the oil. Table 2 illustrates some of the wear metal concentrations (ppm) in the oil at 25 hour intervals.

Condemnation limits are not given as variations between engines are great; abrupt changes of metal concentrations between samples should be the primary indicator of problems and of excessive wear.

Table 2. Ford KSG-411 Engine Oil Wear Metal Analysis

| Hours | Iron | Chromium | Aluminum | Copper | Lead | Tin | Silicon | TBN |
|---|---|---|---|---|---|---|---|---|
| Test Series 1: Beef Manure Derived Fuel | | | | | | | | |
| 24 | 11 | 0 | 0 | 7 | 3 | 4 | 10 | - |
| 50 | 12 | 0 | 0 | 8 | 3 | 7 | 9 | - |
| 75 | 17 | 0 | 0 | 9 | 3 | 8 | 10 | - |
| 100 | 16 | 0 | 0 | 9 | 2 | 5 | 7 | - |
| 125 | 7 | 0 | 8 | 3 | 1 | 0 | 6 | - |
| 150 | 10 | 0 | 0 | 4 | 0 | 0 | 5 | - |
| 175 | 19 | 0 | 0 | 7 | 5 | 3 | 7 | - |
| 200 | 11 | 0 | 5 | 5 | 0 | 0 | 4 | - |
| Test Series 2: Swine Manure Derived Fuel | | | | | | | | |
| 25 | 18 | 0 | 7 | 10 | 3 | 0 | 6 | 10.1 |
| 50 | 23 | 0 | 8 | 15 | 3 | 0 | 5 | 10.7 |
| 75 | 34 | 0 | 0 | 26 | 0 | 3 | 7 | 10.0 |
| 100 | 34 | 0 | 2 | 23 | 0 | 9 | 7 | 9.9 |
| 125 | 11 | 0 | 2 | 7 | 0 | 5 | 9 | 10.3 |
| 150 | 14 | 0 | 9 | 12 | 2 | 0 | 6 | 10.4 |
| 175 | 20 | 0 | 7 | 18 | 1 | 0 | 7 | 10.2 |
| 200 | 24 | 0 | 0 | 17 | 0 | 0 | 7 | 9.9 |

The wear metal concentrations present in the oil indicate that no significant wear has occurred in 400 hours of engine operation on digester gas. The Total Base Number after 100 hours of use is still quite high suggesting that the oil change interval could be significantly extended. A greater period between oil changes will decrease maintenance time and cost, improving the economics of operating the engine as a power source.

## Engine Deposits and Engine Wear

The head was removed after the first 200 hour test series. The small amount of deposits visible on the head and on the upper cylinder walls was soft and easily removed. The engine was dismantled after the second test series and significant deposits were found on the head, the pistons, and the valves. The deposits on the head were nearly to the point of interfering with the valve seatings and looked as though valve problems would have soon developed. Photographs of the deposits on the components are located in Appendix A. An elemental analysis and an electron microscope scan were conducted on the deposits. The results show that carbon, hydrogen, nitrogen, and sulfur are present with trace amounts of zinc, calcium, iron,

The journal bearings and other engine components were examined after the second test series. No excessive wear was evident. No pitting or warnings of impending failure of engine components could be seen, unlike results found by researchers at Cornell. The main bearings were plastigaged to determine wear. The clearances were well within new bearing tolerances.

## Digester Gas Analysis

An attempt to determine the effectiveness of the biogas filters led to gas chromatography analysis of the swine biogas. A 50 meter column was used with a mass spectrometer to determine the compounds present in the gas. Compounds with mass number larger than 30 were being detected by the analysis with only carbon dioxide and hydrogen sulfide appearing. No mercaptans, that is compounds with an S-H radical, appeared in the analysis contrary to literature written by the filter manufacturer reporting the presence of the mercaptans. Gas samples were taken at three points in the system. The first was taken from the top of the digester before any filtering occurred. The second sample had passed through an iron sponge and had been compressed into the storage reservoir. The third gas sample had also passed though the iron sponge and the compresser as well as filtering though the Nelson filters. The gas at each sampling point was approximately 36 percent $CO_2$. The results indicate that no significant reduction of the hydrogen sulfide was taking place in either of the filters. The tests were set up to determine the quantity of $H_2S$ as a fraction of the quantity of $CO_2$. The results are shown in Table 3.

Table 3. Fraction of $H_2S$ Present in 36% $CO_2$ Biogas

| Filtering | %Total-$CO_2$ | %Total-$H_2S$ |
|---|---|---|
| none | 98.69 | 1.31 |
| none | 98.60 | 1.40 |
| none | 98.54 | 1.46 |
| iron sponge | 98.96 | 1.04 |
| iron sponge | 98.64 | 1.36 |
| iron sponge | 98.84 | 1.16 |
| iron sponge and Nelson filter | 99.18 | 0.82 |
| iron sponge and Nelson filter | 98.39 | 1.61 |
| iron sponge and Nelson filter | 98.63 | 1.37 |

## CONCLUSIONS

It is desirable to remove the hydrogen sulfide from the biogas as close to the digester as possible to minimize the damage caused by corrosion of metal surfaces. The filters that were used to remove the hydrogen sulfide from the digester gas proved to make insignificant, if any, reduction of the hydrogen sulfide content.

Larger storage vessels to allow water vapor to condense are desirable both for optimizing engine output and for minimizing the corrosion of engine components.

Proper sizing of the engine, carburetor, and accessories are essential for satisfactory performance.

Engine wear is not considered to be a problem, at least for lubrication oil change intervals of 100 hours. Further tests should be conducted to determine the maximum period of safe engine operation per oil change.

The head should be removed periodically for inspection to identify any problems that might be corrected by minor alterations, such as addition of a filter to remove oil leaking out of the compresser into the gas.

Optimum spark advance and fuel gas to air mixtures is dependent on the methane content of the gas. Small changes of timing or mixture affect peak torque and power only to a slight extent.

Operation of an engine on digester gas should yield torque and power outputs of 80 to 95 percent of operation on natural gas. This will depend on the energy content of the biogas gas which is dependent upon the type of animal, the feed ration, the operation of the digester, and the digester design.

## SUMMARY

This study shows that with minor engine adjustment, operation and performance of a spark ignition engine with an 8 to 1 compression ratio on digester gas is satisfactory. Adjustments include advancement of the timing, enrichment of the gas to air mixture, and an increased fuel supply pressure.

The results of the study show that biogas supplies 80 to 95 percent of natural gas power. Lubricating oil did not exhibit serious signs of deterioration with 100-hour oil change intervals.

Problems encountered include water vapor and hydrogen sulfide concentrations in the gas. In long term engine tests, severe corrosion of engine components may occur. Engine deposits were quite heavy after 200 hours of operation on swine digester gas.

These results can be projected only for short term use of anaerobic digester gas as an engine fuel. Long term tests are needed to verify any assumptions derived by extrapolation of the results.

## SUGGESTIONS FOR FURTHER RESEARCH

Further studies should be made in the development and effectiveness testing of filters for removing hydrogen sulfide from anaerobic digester gas. The filter needs to be inexpensive and have low maintenance requirements.

Research should be performed to better utilize the high octane content of the biogas by testing high compression, spark ignition engines to increase the efficiency of the engine cycle. If these engines would also have increased turbulence in the combustion chamber, the flame speed would probably increase. This would allow the spark advance to be retarded so that a sharper pressure rise could occur near top dead center to optimize the power output of the engine.

A lubricating oil with a high buffering capacity should allow greater time intervals between oil changes. An alternative to this would be to develop acid-resistant engine components.

For an industry that has demands on time that are essentially fully extended, the entire system needs to operate automatically with a high reliability. This system would need a computer to monitor and control the entire process from loading the digester to running the engine generator during peak electric demand periods. An operator should have to do only periodic maintenance such as changing oil in the engine.

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## APPENDIX A. PHOTOGRAPHS OF ENGINE DEPOSITS

Figure 7. Piston Side View after 200 Hours of Engine Operation, Swine Digester Gas

Figure 8. Top of Piston after 200 Hours of Engine Operation, Swine Digester Gas

Figure 9. Face of Head after 200 Hours of Engine Operation, Swine Digester Gas

Figure 10. Intake and Exhaust Valves after 200 Hours of Engine Operation, Swine Digester Gas

## APPENDIX B. FORTRAN PROGRAMS

```
c  MAIN PROGRAM "ENGINE.F"
       include 'flog.h'


c  this program calculates torque, power, mean effective
c  pressure, fuel flow rate, specific fuel consumption,
c  thermal efficiency, correction factor for temperature
c  and pressure for values on the torque curve for a 67
c  cubic inch displacement Ford engine on a dynamometer
c  with a load cell on an 11 inch lever arm.
c  c or cf prefix indicates corrected for temp. and press.
c  m suffix indicates metric units

       real pgas, paatm, frac
       real tmps(8),gas,load,rpm,pt(5)
       real stmps(8),sumcnt,srpm,sload,spt(5)
       common/data/tmps,gas,load,rpm,pt
       common/gas1/pgdiff,d2,b2,pgabs,wh2,gcfm,hw2,densg,
     &    viscg,rd2,cd2
       common/gas2/ftemp,ch4,co2


       common/air/padiff,paabs,d,b,airtmp,wh,acfm,hw,densa,
     &    visca,rd,cd


c  read input file from device 1; write to output device 7
       character*40 fin,fout
       write(6,*)iargc()
       if (iargc() .lt. 2) stop 'usage engine input output'
       call getarg(1,fin)
       call getarg(2,fout)
       open(1,file=fin)
       open(7,file=fout)
       pi = 3.141592654
c      ratio of air nozzle diam. to plenum intake =infinity
       b=0.0
c      ratio of fuel nozzle diam. to fuel tube diam.
       b2=0.25
c      air nozzle diam. (in)
       d=1.25
c      fuel nozzle diam. (in)
       d2=0.5


c  enter data from terminal


       write(6,*)'enter the percent methane (0.xx)'
       read(5,*)frac
       write(6,*)'enter the initial voltages on
```

```
&               press. trans. 1 2 3 4 5'
      read(5,*) pterr1,pterr2,pterr3,pterr4,pterr5

      call fsetre(1)

c  initialize analog channels
 1    sumcnt=0.
      do 10 i=1,8
      stmps(i)=0
 10   continue
      srpm=0
      sload=0
      do 11 i=1,5
      spt(i)=0
 11   continue

100   call getdat
      if (eof .eq. 0 .and. count .eq. 15 .and.
&        pultim .eq. 15) then
      srpm=srpm+rpm
      sload=sload+load
      do 12 i=1,8
      stmps(i)=stmps(i)+tmps(i)
 12   continue
      do 13 i=1,5
      spt(i)=spt(i)+pt(i)
 13   continue
      sumcnt=sumcnt+1.
      endif




      if (last .eq. 0) go to 100

      if (sumcnt .ge. 2) then
      srpm=srpm/sumcnt
      sload=sload/sumcnt
      do 14 i=1,8
      stmps(i)=stmps(i)/sumcnt
   14 continue
      do 15 i=1,5
      spt(i)=spt(i)/sumcnt
   15 continue

c     psi
      padiff=.04*abs(spt(1)-pterr1)
c     psi
      paabs=10.*(spt(2)-pterr2)+14.7
```

```
c       psi
        paatm=paabs
c       psi
        pgdiff=.04*abs((spt(5)-pterr5))
c       psi
        pgabs=10.*(spt(4)-pterr4)+14.7
        ch4=frac
        co2=1.-frac
c       degree R
        ftemp=(stmps(5)+273.)*9./5.
c       degree C
        airtmp=stmps(4)



        call gasflo
        call airflo

c       dyno load (kg)
        loadm=sload*.45359
c       dyno torque (ft-lb)
        torque=sload*11./12.
c       dyno torque (N-m)
        torqm=torque*1.3825

c       brake horsepower
        bhp=2.*pi*torque*srpm/33000.
c       brake kW
        bkw=bhp*.7457

c       brake mean effective pressure (psi)
        bmep=11821.*bhp/srpm
c       brake mean effective pressure (Pa.)
        bmepm=bmep*6894.4

        pgas=pgdiff
        gasp=(paatm+pgas)*13.6

c       fuel gas mass(lb/hr)
        ffrgas=wh2
c       mass of methane(lb/hr)
        ffrch4=ffrgas*frac
c       mass of fuel gas(kg/hr)
        ffrgsm=ffrgas*.45359
c       mass of methane(kg/hr)
        ffrchm=ffrgsm*frac
c   mass of air (lbs/hr)
        airmas=wh
c   mass of air (kg/hr)
        airm=wh*.45359
c   fuel to air ratio (mass basis)
        ftoa=wh2/wh
```

```
c  methane to air ratio
      ch4toa=ffrch4/wh
c  methane to air ratio (volume basis)
      ftoav=gcfm*frac/acfm




c  calculate specific fuel consumption of gas
c  and methane (lbs-gas/bhp-hr)
      sfcgas=ffrgas/bhp
c     lbs-methane/bhp-hr
      sfcch4=sfcgas*frac
c     kg-gas/kW-hr
      sfcgsm=ffrgsm/bkw
c     kg-methane/kW-hr
      sfcchm=sfcgsm*frac

c     calculate thermal efficiency
      theff=bhp*2545./(ffrch4*21297.)*100.

c  calculate correction factor
      cf=1.18*29.3139/paatm*
 &      ((stmps(8)+273.)/298.)**.5-.18




c  correct power and specific fuel consumption
c  for values on the torque curve

      cfbhp=bhp*cf
      cfbkw=bkw*cf
      csfcg=sfcgas*cf*bhp/cfbhp
      csfcch=sfcch4*cf*bhp/cfbhp
      csfcgm=sfcgsm*cf*bkw/cfbkw
      csfccm=sfcchm*cf*bkw/cfbkw


      write(7,500)yr,mo,day,hr,min,sec
      write(7,501)stmps(1),stmps(2),stmps(3),stmps(4),
     &           stmps(5),stmps(6),stmps(8)
c
      write(7,502)loadm,sload,srpm,torqm,torque,bkw,
     &bhp,bmepm,bmep,ffrgsm,ffrgas,ffrchm,ffrch4,airm,
     &ch4toa,ftoav,sfcgsm,sfcgas,sfcchm,sfcch4,
     &theff,cf,cfbkw,cfbhp,csfcgm,csfcg,csfccm,csfcch
c
c
  500 format(6(i2,1x))
  501 format('inlet water temperature',10x,f6.2,/,
```

```
     &'outlet water temperature',9x,f6.2/,'engine
     &     oil temperature',11x,f6.2,/,'air temperature'
     &     ,18x,f6.2,/,'fuel temperature',17x,f6.2,/,
     &     'exhaust gas temperature',9x,f7.2,/,'ambient
     &     air temperature',10x,f6.2,/)
c
  502 format('load-kg (lb)',21x,f6.2,2x,'(',f6.2,')',/,
     &     'rpm',29x,f6.2,/,'torque-nm (ft-lb)',16x,f6.2,
     &     2x,'(',f6.2,')',/,'brake power-bkw (bhp)'12x,
     &     f6.2,2x,'(',f6.2,')',/,'brake mean effective
     &     pressure N/m2 (psi)',12x,f6.2,4x,'(',f4.2,')',/,
     &     'gas fuel flow rate kg/hr (lb/hr)',24x,f6.4,2x,
     &     '(',f6.4,')',/,'ch4 fuel flow rate kg/hr
     &     (lb/hr)',24x,f6.4,2x,'(',f6.4,')',/,
     &     'air mass per hour kg/hr',5x,f7.4,/,'fuel to air
     &     ratio mass basis (volume basis)',5x,f4.2,5x,'(',
     &     f4.2,')',/,'gas specific fuel consumption kg/bkw-hr
     &     (lb/bhp-hr)',5x,f6.4,2x,'(',f6.4,')',/,
     &     'ch4 specific fuel consumption kg/bkw-hr
     &     (lb/bhp-hr)',5x,f6.4,2x,'(',f6.4,')',/,
     &     'thermal efficiency',15x,f6.2,'%',/,
     &    'correction factor for torque curve values',13x,f6.2,
     &     /,'corrected brake power bkw (bhp)',2x,f6.2,
     &    consumption kg/bkw-hr (lb/bhp-hr)',1x,f6.4,2x,'(',
     &    f6.4,')',/,'corrected ch4 specific fuel consumption
     &     kg/bkw-hr (lb/bhp-hr)'
     &    ,1x,f6.4,2x,'(',f6.4,')',///)
      endif
      if (eof .eq. 0) go to 1


      stop
      end

c  ****************************************************


      subroutine getdat
      include 'flog.h'
      real tmps(8),gas,load,rpm,pt(5)
      common/data/tmps,gas,load,rpm,pt

  200 call fgetre
      if (eof .ne. 0) return

c  calculate copper-constantan temperatures
      do 210 i=1,5
  210 analog(i)=analog(i)-analog(16)
      analog(8)=analog(8)-analog(16)
      vref=fttov(analog(8)*20.)
      do 220 i=1,5
  220 tmps(i)=fvtot(analog(i)+vref)
      tmps(8)=fvtot(vref)
```

```
c       calculate iron-constantan temperature
        analog(6)=(analog(6)-analog(16))*1000.
        tmps(6)=tmps(8)+((analog(6)*32.72)-32.)
 &              *5./9.
        rpm=(analog(9)-analog(7))*1000.
        load=(analog(10)-analog(7))*100.
        gas=pulse(1)
        do 240 i=1,5
        pt(i)=analog(i+10)-analog(7)
 240    continue
        return
        end

c   ***********************************************************
c   subroutine gasflo

c       calculates gas fuel flow from the differential
c        pressure across an asme nozzle.


c   argument(s) required from the calling routine

c   (located in common statement)
c       pgdiff
c       d2
c       b2
c       pgabs


c   argument(s) supplied to the calling routine

c   (located in common statement)
c       wh2
c       gcfm
c       hw2
c       densg
c       viscg
c       rd2
c       cd2
c*********** gas flow calculations ***************

        subroutine gasflo
        include 'flog.h'
        real kf,kch4,kco2
        real lhvch4,lhvco2
        common/gas1/pgdiff,d2,b2,pgabs,wh2,gcfm,hw2,
 &       densg,viscg,rd2,cd2
        common/gas2/ftemp,ch4,co2

c    nozzle discharge coeff. initially assumed to be 0.99
        cd2 = .99
c    nozzle thermal expansion factor is assumed to be 1.0
```

```
      f = 1.0
c     velocity of approach factor is assumed to be 1.0
      f1 = 1.0
c     d2 is nozzle diameter in inches.
c           kf is gas specific heat ratio cp/cv of fuel
c           rf2 is fuel gas constant (ft-lbf/lbm-deg r)
c         b2 is ratio of nozzle dia. to intake tube dia.
c
      tfuelr = ftemp
      pi = 3.141592654

c  calculate the fuel molecular weight

      tmwt=ch4*16.+co2*44.

c     partial molecular wt of fuel due to ch4
      zch4=ch4*16./tmwt
c     partial molecular wt of fuel due to co2
      zco2=co2*44./tmwt

c     gas constant for ch4 (ft-lbf/lbm-deg r)
      rch4=96.33
c     gas constant for co2 (ft-lbf/lbm-deg r)
      rco2=35.11

c  calculate fuel gas constant

      rf2=rch4*zch4+rco2*zco2

c     specific heat ratio (cp/cv) of ch4
      kch4=1.31
c     specific heat ratio (cp/cv) of co2
      kco2=1.31

c  calculate specific heat ratio of the fuel

      kf=kch4*zch4+rco2*zco2

c     viscosity of ch4 (lbf/hr-ft at 77f)
      vch4=.0259
c     viscosity of co2 (lbf/hr-ft at 77f)
      vco2=.0348
c  calculate fuel viscosity(lbf/hr-ft at 77f)

      viscg=vch4*zch4+vco2*zco2

c  convert viscosity to (lbm/ft-s)
c  multiply by 32.2(lbm-ft/lbf-s-s) & divide
c  by 32.2(ft/s-s)& divide by 3600(s/hr)

c       (lbm/ft-s)
      viscg=viscg/3600.
```

```
c       lower heating value for ch4 (btu/lbm)
        lhvch4=21500.
c       lower heating value for co2 (btu/lbm)
        lhvco2=0.

c  calculate lower heating value of fuel (btu/lbm)

        lhv=zch4*lhvch4+zco2*lhvco2

c  calculate lower heating value of fuel at
c  std. temp. of 77 deg f & std. press. of
c  14.7 psi (btu/cu.ft.)

        lhvv=(pgabs*144/tfuelr)*(lhvch4*zch4/rch4
&       +lhvco2*zco2/rco2)*(537./tfuelr)

c************** calculation of gas flow **************

c       pressure differential across nozzle(in.h2o)
        hw2 = pgdiff*1728.0/62.4
c       pressure ratio at nozzle
        rf1 = (pgabs-pgdiff)/pgabs

c  expansion factor for nozzle

        y=(kf/(kf-1.)*(1.-rf1**((kf-1.)/kf))/
 &       (1.-rf1))**.5

c       density of fuel (lbm/cu.ft.)
        densg = pgabs*144.0/(rf2*tfuelr)

c  mass flow rate of fuel (lbs/hr) across nozzle
c  calculated by iteration starting with an assumed
c  coefficient of discharge
 100    wh2 = 359.0*cd2*f*(d2*d2)*y*f1*(hw2*densg)**.5

c  convert mass flow rate of fuel to linear flow rate
c  (ft/s)

        vbar = ((wh2/3600.0)/densg)*(576.0/(pi*d2*d2))
c
c       reynold's number
        rd2 = (vbar*(d2/12.0)*densg)/viscg

c  recalculate coefficient of discharge from
c  reynold's number

        ccd = .19436+(.152884*(alog(rd2)))-(9.7785e-03
&       *(alog(rd2))**2)+(2.0903e-04*(alog(rd2))**3)

c  compare calculated cd2 with assumed cd2
        x1 = abs(cd2-ccd)
        if(x1 .le. 1.0e-4) go to 200
```

```
      cd2 = ccd
      go to 100
 200  cd2 = ccd
      wh2 = 359.0*cd2*f*(d2*d2)*y*f1*
 &    (hw2*densg)**.5
c     gas flow rate(cfm)
      gcfm = wh2/(60.0*densg)
      return
      end

c  ************************************************
c  subroutine airflo


c  located in the common statement
c  (d b paabs padiff airtmp)
c  argument(s) supplied to the calling routine
c     wh
c     acfm
c     hw
c     densa
c     visca
c     rd
c     cd


      subroutine airflo
      include 'flog.h'
      common/air/padiff,paabs,d,b,airtmp,wh,
 &    acfm,hw,densa,visca,rd,cd
      real k

c nozzle discharge coeff. initially assumed to be 0.99
      cd = .99
c  nozzle thermal expansion factor is assumed to be 1.0
      f = 1.0
c     velocity of approach factor is assumed to be 1.0
      f1 = 1.0
c     d is nozzle diameter in inches.
c     k is air specific heat ratio cp/cv
      k = 1.4
c     r2 is gas constant (ft-lbf/lbm-deg r)
      r2 = 53.34
c    b is ratio of nozzle diameter to intake tube diameter
c     if intake diameter equals infinity, then b=0.0
c     ambient temp. in deg. rankine
      taambr = (airtmp*(9.0/5.0)+32.0)+459.69
      pi = 3.141592654
      paatm=paabs

c************* calculation of air flow *************
```

```
c  abs. viscosity of air (lbm/ft-s) from air temp.
c  between 450r and 576r

      visca = (-4.991211+(2.852927*taambr)
&     *(.1011469e-02*taambr**2))/1.0e+8

c     pressure differential across nozzle(in.h2o)
      hw = padiff*1728.0/62.4

c     pressure ratio at nozzle
      r1 = (paatm-padiff)/paatm

c  expansion factor for nozzle

      y=(k/(k-1.)*(1.-r1**((k-1.)/k))/(1.-r1))**.5

c     density of air (lbm/cu.ft.)
      densa = paatm*144.0/(r2*taambr)

c  mass flow rate of air (lbs/hr) across nozzle
c  calculated by iteration starting with an assumed
c  coefficient of discharge

 100  wh = 359.0*cd*f*(d*2)*y*f1*(hw*densa)**.5

c  convert mass flow rate of air to
c   linear flow rate (ft/s)
      vbar = ((wh/3600.0)/densa)*(576.0/(pi*d*d))

c     reynold's number
      rd = (vbar*(d/12.0)*densa)/visca

c  recalculate coefficient of discharge
c  from reynold's number

      ccd = .19436+(.152884*(alog(rd)))-(9.7785e-03
&     *(alog(rd))**2)+(2.0903e-04*(alog(rd))**3)

c  compare calculated cd with assumed cd
      x1 = abs(cd-ccd)
      if(x1 .le. 1.0e-4) go to 200
      cd = ccd
      go to 100
 200  cd = ccd
          wh = 359.0*cd*f*(d*d)*y*f1*(hw*densa)**.5
c     density of air(lb/cu.ft.)
      densa = paatm*144.0/(r2*taambr)
c     air flow rate(cfm)
      acfm = wh/(60.0*densa)
      return
      end
c
c  FLOG.H
```

```
o  Header file for sym data logger files
c
c  Miohael D. Sohwarz
c  May 5, 1982
c

      character*8 file
      integer unit, errflg, eof, lineno, havlin
      integer id, switoh, nalog, ndig, npul, tsmp,
      integer tavg, tsav
      integer nyr, nmo, nday, nhr, nmin, nsec, noount

      common /head/ file, unit, errflg, eof, lineno
      common /head/ havlin,id, switoh, nalog, ndig
      common /head/ npul, tsmp, tavg, tsav,nyr, nmo
      oommon /head/ nday, nhr, nmin, nsec, noount

      integer first, last, yr, mo, day, hr, min, sec
      integer count
      real    analog(32), pulse(2)
      integer dig(8), pultim

      common /rec/ first, last, yr, mo, day, hr, min
      common /rec/ sec, count,analog, dig, pultim, pulse
c
c  FGETREC.F
c  getrec  get a record from the file and place in the
c  rec sturct.
c
o  Michael Sohwarz
c  May 5, 1982
c


      subroutine fsetre(nunit)
      inolude 'flog.h'

      integer nunit
      lineno = 0
      havlin = 0
      unit = nunit
      eof = 0
      errflg = 0
      end

      subroutine fgetre
      inolude 'flog.h'

      integer i
      character*1 errchr
      character*80 linein

      if (eof .ne. 0) then
```

```
      stop 'Getrec reentered after EOF'
      endif

 1    if (havlin .eq. 0) then
      errflg = 0
      first = 1
      last = 0
      lineno = lineno + 1
      read (unit, 500, end=99, err=80) file,
     & id, switch, nalog, ndig, npul, tsmp, tavg, tsav
c  write(6,500) file, id,
c&       switch, nalog, ndig, npul, tsmp, tavg, tsav
      errflg = 0
  if (file(7:7) .eq. 'E' .or. file(7:7) .eq. 'e') errflg = 1
   if (file(4:4) .ne. 'L' .and. file(4:4) .ne. 'l') then
      goto 80
      endif
      lineno = lineno + 1
      read (unit, 510, end=90, err=1) nyr, errchr, nmo,
     &   nday, nhr, nmin, nsec, ncount
c  (6,510) nyr, errchr, nmo,
c&       nday, nhr, nmin, nsec, ncount
      if (errchr.ne.'/'.and.errchr.ne.' ') goto 1
      else
      first = 0
      endif

      yr = nyr
      mo = nmo
      day = nday
      hr = nhr
      min = nmin
      sec = nsec
      count = ncount

      if (nalog .ne. 0) then
      do 11 j=1,nalog,6
      read(unit, 550) linein
      do 12 k=8,80,12
          if (linein(k:k).eq.'E') linein(k:k)='e'
 12   continue
      read (linein, 520, end=90, err=80)
     &   (analog(i), i=j,min0(nalog, j+5))
 11   continue
c  (6,520) (analog(i), i=1,nalog)
      lineno = lineno + (nalog + 5)/6
      endif
      if (ndig .ne. 0) then
      read (unit, 530, end=90, err=80) (dig(i), i=1,ndig)
c  (6,530) (dig(i), i=1,ndig)
      lineno = lineno + (ndig + 9)/10
      endif
      if (npul .ne. 0) then
```

```
      read (unit, 540, end=90, err=80) pultim,
     &   (pulse(i), i=1,npul)
c  (6,540) pultim, (pulse(i), i=1,npul)
      lineno = lineno + 1
      endif
      lineno = lineno + 1
      read (unit, 510, end=90, err=85) nyr, errchr, nmo,
     &  nday, nhr, nmin, nsec, ncount
c  (6,510) nyr, errchr, nmo,
c&       nday, nhr, nmin, nsec, ncount
      if (errchr.ne.'/'.and.errchr.ne.' ') goto 85
      havlin = 1
      eof = 0
      return

c   read error handling code.
c unexpected error

 80   write (6,600) lineno
      havlin = 0
      goto 1

c this occurs normally at the end of a cassette file

 85   last = 1
      havlin = 0
c  (6,620) lineno
      return

c   read eof handlng code.
c unexpected end of file

 90   write (6,610) lineno

c normal end of file

 99   eof = 1
      last = 1
c  (6,630) lineno
      return

c  FORMAT STATEMENTS

 500  format(a7,8i4)
 510  format(i2,a1,5(i2,1x),i5)
 520  format(6(e11.4,1x))
 530  format(8(i5,1x))
 540  format(i5,2f10.0)
 550  format(a80)

 600  format('GETREC> unexpected error   lineno:',i4)
 610  format('GETREC> unexpected end of file
&       lineno:',i4)
```

```
c  620 format('GETREC> Hit end in data file
&        lineno:',i4)
c  630 format('GETREC> Hit end of file in data file
&        lineno:',i4)
       end
c
c
c  FTHERMO.F
       function fmtot(mv)
       real mv, fmtot

       fmtot = (( (-0.77) * mv) + 26.0 ) * mv
       return
       end

       function fttom(t)
       real t, fttom, sqrt

c  /* mv = (26 - sqrt(676 - 3.08 * t))/1.54  */
       fttom = 16.88311688 - sqrt(285.0396356889
&      - 1.2987012987 * t)
       return
       end

       function fvtot(v)
       real v, fvtot

       fvtot = (( (-0.77e+6) * v) + 26.0e+3 ) * v
       end

       function fttov(t)
       real t, fttov, sqrt

c  /* mv = (26 - sqrt(676 - 3.08 * t))/1.54  */
       fttov = 16.88311688e-3 - sqrt(285.0396357e-6
&      - 1.298701299e-6 * t)
       return
       end
c
c
c  FTRANS.F
       subroutine ftrans
include 'flog.h'

       do 10 i=1,nalog
       if (i .ne. 7) then
       analog(i) = analog(i) - analog(7)
       endif
 10    continue

       analog(8) = analog(8) * 20
       compen = fttov(analog(8))
d      write(6,*) compen, analog
```

```
      do 20 i=1,nalog
      if(i.ne.8.and.i.ne.7.and.i.ne.11.and.i.ne.16)then
      analog(i) = fvtot(analog(i) + compen)
      endif
 20   continue
      return
      end
c
c
c  LOGVAR.F
c  calculate the variance of the analog data channels

include 'flog.h'
      real sum(32), sumsq(32)
      character*80 fname

      call getarg(1, fname, 80)
      write(6,*) fname
      call setfil(1, fname)
      call fsetre (1)

 100  do 101 i=1,32
      sum(i) = 0
      sumsq(i) = 0
 101  continue
      n = 0

 1    call fgetre
      if (eof.ne.0) then
      stop 'end of data'
      endif

      call ftrans
      do 10 i=1,nalog
      sum(i) = sum(i) + analog(i)
      sumsq(i) = sumsq(i) + analog(i)*analog(i)
 10   continue
      n = n+1
      if (last.eq.0) goto 1
      if (n.eq.1) then
      div = 1
      else
      div = n-1
      endif

      write(6,*) file, n, id
      write(6,600) yr,mo,day,hr,min,sec
 600  format(i2,'/',i2,'/',i2,1x,i2,':',i2,':',i2)
      do 20 i=1,nalog
      sum(i) = sum(i)/n
      xsq = sumsq(i) - sum(i)*sum(i)*n
      if (xsq.lt.0.0) then
      write(6,*) 'sqrt of negative',xsq,sum(i),sumsq(i)
```

```
      xsq = -xsq
      endif
      sumsq(i) = sqrt(xsq/div)
      if (sumsq(i).ne.0.0) then
      varian = 100.0*sumsq(i)/sum(i)
      else
      varian = 0
      endif
      write(6,*) i, sum(i), sumsq(i), varian
 20   continue
      write(6,*)
      goto 100
      end
c
c
c  TEST.F
#include "flog.h"

      call setrec(1)

 1    call getrec()
      write(6,*) file, unit, errflg, eof, lineno
      write(6,*) havlin,id, switch, nalog, ndig, npul
      write(6,*) tsmp, tavg, tsav,nyr, nmo, nday, nhr
      write(6,*) nmin, nsec, ncount

      write(6,*) first, last, yr, mo, day, hr, min, sec
      write(6,*) count,analog, dig, pultim, pulse

      if (eof .eq. 0) goto 1
      write(6,*) 'got EOF'
      call getrec()
      write(6,*) 'call 2'
      call getrec()

      stop 'from main'
      end
```

BEEF AND SWINE DIGESTER GASSES-

EVALUATION AS FUELS FOR

SPARK IGNITION ENGINES

by

JERRY DWIGHT MARR

B. S., Kansas State University, 1983

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AN ABSTRACT OF A MASTER'S THESIS

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ABSTRACT

Anaerobic digester gas from two facilities were individually evaluated for performance in an internal combustion engine. The first test series used digester gas produced from beef manure while the second series of tests measured the performance on a swine derived gas.

The beef based gas, produced at a research digester, had an average methane content of approximately 53 percent although week to week variations in the methane content were sometimes as great as ten percent. The swine based gas, from a more steady state commercial operation, had a 64 percent methane content with variation of less than two percent.

Gas chromotography tests on the swine based gas revealed that in addition to the primary methane and carbon dioxide gas constituents, that approximately one-half percent of the gas was hydrogen sulfide which may be a cause of premature engine failure. The gas chromotography also revealed that the filters used on the system to reduce the hydrogen sulfide concentration made insignificant, if any, reduction of the hydrogen sulfide content.

Engine tests were conducted using a four cylinder, spark ignition engine with an 8 to 1 compression ratio. The 200- hour engine test cycle was the alternative fuel screening test recommended by the Engine Manufacturer's Association. Engine performance, lubricating oil deterioration, and engine wear and

deposits were measured or monitored to determine problems encountered with the use of the fuel.

Engine performance indicated that the torque and power output of the engine fueled with digester gas was from 80 to 95 percent of the outputs of the engine fueled with natural gas. The output depended on the methane concentration in the fuel, the amount of water vapor present, and the gas supply pressure.

Modifications required for satisfactory engine operation included a higher fuel supply pressure than that needed for natural gas. The engine performance was the best with gas pressures of 7 to 12 kPa (35 to 50 inches of water). An increase in the fuel to air mixture and increased spark advance were also required.

Wear metal concentration levels in the lubricating oil revealed no excessive engine wear. Total Base Number of the oil remained high with oil change intervals of 100 hours. Main bearing clearances after 400 hours of operation on the digester gas were well within the tolerances of new bearings. Visual inspection of other engine components revealed no excessive wear.

A problem found in the first test was excessive amounts of water vapor in the gas. Deposits on the engine oomponents were minimal after the tests on the beef digester gas, but after operation at the swine facility, the deposits were significant, engine failure could have ocourred within a short time period.