DUAL FUELING AGRICULTURAL SIZE DIESEL ENGINES

by

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INTRODUCTION

Dr. Rudolf Diesel patented the compression ignition engine principle in 1892. Working from the theoretical side, he stated in his patent that higher efficiency could be expected because of the larger expansion ratio of the gases following combustion.

According to Gray (13), the first diesel powered agricultural tractor was offered commercially in 1931 when the Caterpillar Tractor Company produced the "Diesel 65". The delay in the adaptation of the compression ignition engine to tractors was the result of the diesel engine's requirement of precision-made parts for straining and metering the fuel and special designing of parts to withstand extremely high pressures. These requirements resulted in a higher first cost than equivalent spark ignition engine powered tractors. Thus, to realize any economical advantage of the diesel tractor with its lower cost fuel and higher efficiency, it was necessary to have a high annual use--at least 1000 hours.

Technological developments in manufacture, materials, and engine design have made the diesel powered tractor more competitive with the spark ignition powered tractor. Implement and Tractor Magazine reports figures which show that approximately 58 per cent of the wheel tractors produced in the United States in the first half of 1966 were diesel powered. Ninety-four per cent of the tracklayers were also diesel powered. In 1960 about 41 per cent of the wheeled tractors produced were diesel powered. Diesels are invading the smaller power ranges. Of the 58 per cent of wheel tractors produced which were diesels, 22 per cent were under 50 horsepower, and even 10 per cent were 34 horsepower or under. These smaller diesel powered tractors will be assigned a wide variety of jobs; and the power unit's flexibility to meet this variety will be far different from that of their predecessors of 20 or 30 years ago.

The United States has been able to develop its industrial might because the American farmer has never been quite satisfied with the way he found things. He has usually done something about it. Thus in 1963 one farmworker supplied 30.74 persons all of their farm products compared to 10.69 persons in 1940 and 4.12 in 1820. The American farmer has always been an improviser.

A number of Kansas farmers have found that if they add a little liquefied petroleum gas (LPG) to the intake air of their diesel tractors, they get an increase of power. This thesis is an investigation of the merits of this practice.

The author has chosen to use the following terminology:

Dual fueling--the practice of replacing part of the diesel fuel oil required to produce rated power with a gaseous fuel introduced with the engine intake air.

Power fueling--the practice of adding gaseous fuel to the intake air of a diesel engine without reducing the conventional diesel fueling rate in order to increase the maximum power above the rated power.

REVIEW OF THE LITERATURE

In 1901, the U. S. Patent Office issued Dr. Diesel a patent covering the principle of compressing a mixture of gaseous fuel and air to a pressure and temperature below the ignition point of the mixture. The mixture is ignited by injecting a liquid fuel that has an ignition point below the conditions attained in the compression of the mixture. This patent covered the essential features of what are now called the dual fuel and gas diesel engines.

The Diesel Engine Manufacturers Association (35, 7) defines a gas diesel engine as one that uses a gas as the primary fuel so that the ignition of the fuel-air mixture is affected, or aided, by injecting pilot fuel after the fuel-air mixture's compression has almost been completed. A dual fuel engine is defined as an engine that can be operated as (1) an oil diesel, that is, only air is compressed, and the liquid fuel is injected into the compressed air, and (2) as an engine using a combination of gaseous and liquid fuels. The gaseous fuel is mixed with the intake air, the mixture is compressed, and then the liquid fuel is injected. The proportion of the gaseous fuel can be increased until the liquid fuel is at a minimum to permit it to ignite the gaseous fuel-air mixture. Then the engine operates as a gas diesel.

Armstrong and Hartman (2) state that dual fueling has been used with large stationary engines since about 1940 when natural gas suppliers made it available at an attractive price on an interruptible basis.

Thus a diesel engine is not restricted to the use of diesel oil as a fuel. In fact, according to Dr. Rosen (33), one of the diesel engine's merits is its ability to utilize a wide range of fuels. In areas of the world with no petroleum but an abundant supply of vegetable oils, these oils could be utilized to advantage in diesel engines.

Armstrong and Hartman (2) state that the modern oil diesel consumes 25 to 100 per cent excess air at rated power. From this it appears that an engine's capacity could be increased by adding more fuel to utilize this air.

McLaughlin (22) reports that diesel truck operators in California have, in many cases, increased the fuel injection rates above those recommended by the engine's manufacturer to increase the performance for their mountain routes. They report that the engines produce more smoke--probably because this extra fuel is not mixed with oxygen-rich air during combustion. They contend that LP gas can be added to the air via the air intake; and because this fuel is well mixed with the air before combustion takes place, an increase in power can be obtained without any increase in the smoke production rate.

In many ways the introduction of a volatile fuel into the intake air of a diesel engine results in a hybrid between an Otto and a Diesel engine. But there are basic departures from both of them. These will be discussed later, but a brief review of the general theory of combustion as it specifically applies to the Otto cycle and Diesel cycle is probably in order.

The chain theory (25, 36) is now generally accepted for the combustion of a fuel-air mixture. This theory states that one of the products of the reaction of combustion, a chain carrier, is able to cause, or may even be necessary to cause, a reaction which in turn produces more chain carriers. The chain of events continues until combustion is complete or something breaks the chain.

One method of breaking the chain is to cool the constituents below the ignition temperature. Gross and Gronomski (14) report that flame cooling is one of the major factors which cause diesels to smoke.

The ignition temperature (25) is that temperature at which spontaneous ignition of a fuel-air mixture will occur when it is heated gradually. Three things vary the ignition temperature: (1) the kind of fuel, (2) the fuel-air ratio, and (3) the pressure. The influence of these is readily explained on the basis of the chain reaction theory. But according to Maleev (21), the ignition temperature is not influenced so much by the pressure as by the heat transfer, which itself is influenced by the density of the fuel-air mixture.

Mullins (25) says that Semenov calculated that the condition for explosion could be represented by the equations

$$\tau p^n = A\ e^{E/RT}$$

or

$$\tau (P/T)^n = A\ e^{E/RT}$$

where τ = ignition delay

P = pressure

n = number of energy terms representable by "square terms"

A = a constant

e = exponential number

E = activated energy

R = molar gas constant

T = ignition temperature.

The first equation is correct if the length of the chain and the reaction rate are determined by the pressure, and the second is correct when they are determined by the density. These equations are based on the assumption that the reactions accelerate isothermally to the point of ignition. This is not true, though, because as the reaction rate increases, there is a rise in temperature.

Maleev (21) states that the ignition temperature is given by the empirical relation

$$T = C/W^m$$

where T = ignition temperature in degrees Rankine

C = constant and is 820 for aliphatic hydrocarbons, C_nH_{2n+2} and C_nH_{2n} and varies from 1030 to 1100 for aromatic fuels, C_nH_{2n-6}

W = specific weight of the charge when it ignites

m = constant and is 0.16 for the aliphatics and varies from 0.193 to 0.23 for the aromatics.

Lichty (20), in a quotation from Wolfer (40), states that the ignition delay can be given by the expression

$$\text{Ignition delay} = f\ (e^{a/T})/P^{b}$$

where T = temperature, degrees Rankine

P = pressure, atmospheres

a = constant and is 8200 for liquid paraffins

b = constant and is 1.2 for liquid paraffins.

From work with all types of chemical reactions, but especially from work with fuel-air combustion and with explosives, it is known that it takes at least a minimum amount of energy to cause a reaction to start (25, 26, 34, 41). This minimum energy is known as the activation energy.

Combustion in the Spark Ignition Engine

The spark ignition engine utilizes a near stoichiometric mixture of fuel and air. The fuel is gaseous and well mixed with the air by the time combustion is to occur. The charge also contains approximately 10 per cent or less residual gases from the previous cycle, since, as Taylor and Taylor (36) point out, the Otto engine has a compression ratio of 10:1 or less and is of 4-stroke design.

This gaseous mixture is compressed, and because of the increased pressure and temperature, there are reactions which take place before actual ignition. Retaillian (30) has shown by the use of a motored nonfired engine that 53 per cent of the fuel had undergone chemical reaction and alteration. Paraffinic hydrocarbons react most readily, with time and temperature seeming to be the predominate governing factors.

In the modern engine a high-voltage electric spark ignites the charge. Because of the near stoichiometric fuel-air mixture, the electric spark can supply the activation energy. The charge does not burn simultaneously, but a definite reaction zone, known as a flame front because of its luminosity, moves out from the ignition point. Taylor and Taylor (36) point out that because of this nonsimultaneous burning, each portion of the charge goes through a different pressure and temperature history.

They also show that because the flame sweeps across the combustion chamber the major portion of the pressure rise occurs during the latter part of the flame's travel across the chamber. It is also this phenomenon which produces the phenomenon of detonation or knock. A small portion of the charge, the end gas, which has not yet been burned, is compressed both by the piston and the expanding of the burned part of the charge till it reaches a temperature and pressure sufficient for it to auto-ignite. If a sufficient amount of charge is involved and the reaction occurs rapidly enough, it sends out a pressure wave which results in an audible knock.

Some fuels are more resistant to knocking because they have a longer delay period, that is, there is a greater lapse of time between when they reach their ignition temperature and when they react. Retailliau (30) has demonstrated that there is a definite relation between prereactions and knock in a spark ignition engine. Anything that tends to increase the end gas temperature or shorten its delay time aggravates detonation.

Combustion in the Diesel Engine

Most authorities agree that the chemical reactions of combustion in the diesel cycle engine and the Otto cycle engine are similar but that the physical processes are quite different. Air only is compressed in the diesel. Because of the high compression ratios used, usually over 12:1 and less than 20:1, the residual gas content will be quite low. Liquid fuel is sprayed into the compressed air, and, as Taylor and Taylor (36) point out, there is undoubtedly some chemical reaction immediately as soon as the first fuel contacts the air because of the high pressures and temperatures. These reactions are so slow that there is no visible flame or measurable pressure rise. After the <u>delay period</u> or <u>ignition delay</u>, flame does appear and the pressure starts to rise.

Ricardo (31) divided diesel combustion into three stages: (1) a delay period, which allows an appreciable amount of fuel to accumulate before ignition occurs, (2) a period of rapid combustion because the fuel has vaporized and become well mixed with the air, and (3) a period during which the remaining fuel burns at a rate controlled by the speed at which it contacts the necessary oxygen.

There appears to be disagreement on the events of the delay period. Taylor and Taylor (36) state that it can be inferred that the time lag in establishing temperature equilibrium between the injected fuel and the air is negligible. This is based on constant volume bomb tests. But Wood (42), Wieber (39), Priem (28), and El Wakil (11), have calculated the histories of a fuel drop in high temperature air based upon the air's velocity, pressure, and temperature, and the fuel drop's vapor pressure, diffusion coefficient, thermal conductivity, temperature, and size. They have also used experimentation to confirm their calculations. Wieber has calculated that it is possible for a fuel drop to reach its critical temperature before there is any appreciable vaporization if there is sufficiently high combustion pressure. He states that this phenomenon could lead to high pressure combustion instability.

Combustion in the Dual Fuel Engine

In the past dual fueling has been restricted to large stationary engines. If spark ignition had been used on these large engines, a near perfect mixture would have been necessary to enable the spark to deliver the activation energy to insure ignition. But as was pointed out earlier, detonation occurs when the end gas is auto-ignited by the rising pressure before the flame has swept completely across the cylinder. With these large, slow-speed engines, a considerable amount of time is required for the flame to travel the entire distance across the cylinder--especially if only one ignition point is used. Thus it would take a fuel with a considerable ignition delay to have knock-free operation.

The gaseous fuels can be utilized in large engines, though, by using dual fueling. Unlike the spark ignition engine, the gaseous fuel is carbureted into the intake air to form a lean mixture instead of a stoichiometric mixture, and thus the ignition temperature of the fuel air charge is raised and its delay period is lengthened. The longer delay period allows the charge to burn in an orderly manner rather than the last portion autoigniting. Also, the higher ignition temperature and the longer delay period of the lean mixture allow higher compression ratios to be used than are possible with stoichiometric mixtures resulting in some gain in thermal efficiency. But the lean mixture raises the required activation energy above that which can be reliably supplied by an electric spark. A pilot injection of diesel oil, though, injected into the compressed charge can supply the required activation energy level.

Detonation is further controlled in the dual fuel engine with its premixed charge, as Armstrong and Hartman point out, by the widely separated, simultaneous ignition points which result from the wide dispersion of the fuel droplets. They point out also that, like the oil diesel engine, the injection of the pilot oil charge is followed by an ignition delay governed by the same factor as that in the oil diesel engine. If the pilot charge is less than that which would accumulate during the delay period of the oil diesel engine, the pressure rise will be less.

The combustion of the oil droplet starts in the vapor film around it. At the outer edge of this film, the mixture has the composition of the primary fuel-air mixture. The air-fuel ratio varies from that of the primary fuel-air mixture down to zero next to the droplet. Wood (42), Priem (28), and El Wakil (10) state that combustion will start when the temperature, as the result of heat transfer into the vapor film, and the fuel-air ratio are such that the ignition point has been reached. The heat thus released will further vaporize the fuel drop if it is not already in the vaporous state. The results found by Wood, Priem, and El Wakil seem to indicate that combustion can occur even though part of the drop is still in the liquid state.

Armstrong and Hartman (2) state that there is another ignition delay between the combustion of the oil and the combustion of the gas charge. Following this delay, combustion rapidly accelerates till chemical equilibrium is reached. They point out that the several successive steps in the combustion process have an effect of producing a lower maximum cylinder pressure than would be anticipated. If the delay period causes part of the gas charge's combustion to take place after top dead center, the indicator card will have a rounded combustion portion which resembles the combustion portion of the mechanically injected oil diesel.

THE INVESTIGATION

Objective

The objective of this investigation was to find the effect that power fueling an agricultural size diesel engine has on its economy and its maximum power. Since economy is determined by both thermal efficiency and engine life, it is necessary to measure what effect power fueling could be expected to have on these factors. Expected engine life can only be determined by long field testing; but Ricardo (31) pointed out that durability and reliability were influenced to a great extent by the efficiency of an engine because low efficiency means higher exhaust temperatures, which in turn cause more rapid deterioration of parts and more difficult lubrication. Thus as a measure of the effect that power fueling can be expected to have on engine life, an inference can be made from exhaust temperature, oil temperature, peak cylinder pressure, the mean effective pressure, and thermal efficiency.

The effect upon efficiency and maximum power can be measured directly.

Thus the objectives of this investigation and the necessary measurements to ascertain them are:

1. The effect of power fueling upon maximum power
    - A. Determine engine speed
    - B. Determine torque

2. The effect of power fueling upon economy

   A. Efficiency

      a. Determine power

      b. Determine LP gas consumption rate

      c. Determine diesel oil consumption rate

   B. Engine life

      a. Exhaust temperature

      b. Coolant temperature

      c. Oil temperature

      d. Mean effective pressure

      e. Detonation limit.

Equipment

The engine used in this study was a Continental Motors Corporation model GD157, specification 319, engine number 2459. This is a four-cylinder engine with a 3 3/8-inch bore and 4 3/8-inch stroke, a displacement of 157 cubic inches, and a compression ratio of 15.54:1. The type of combustion chamber is a Lanova Cell. The manufacturer's rating is 39 brake horsepower at 2000 rpm and a maximum torque of 113.5 lb.-ft. at 1250 rpm.

The engine was factory equipped with American Bosch Corporation fuel injection equipment, model PSB4A-700-3485A2. This is a constant-stroke, single-distributing plunger, sleeve-control type pump. The plunger is actuated by a cam and tappet arrangement which also carries the governor and a gear type fuel supply pump.

The generator, fan, radiator, and air cleaner were left on the engine. It was necessary, though, to remove the air cleaner during the series of tests to determine optimum injection timing in order to facilitate the setting of the timing.

The engine was connected to the Agricultural Engineering Department's hydraulic dynamometer for power absorption and measurement.

Fuels

The fuels used in this series of tests were commercial fuels of the type supplied by local suppliers. The diesel fuel was a No. 2 with an API gravity at 60°F of 34.8. The LP gas used was commercial propane. Lower heating values used for thermal efficiency calculations and for determining the proportion of energy supplied by the primary and the secondary fuels were:

Diesel fuel	18,380 BTU/lb
LP gas	21,800 BTU/lb

The diesel fuel was handled by the engine's conventional fuel system. The diesel fuel was considered the primary fuel.

The LP gas was withdrawn from the pressure container as a vapor. A pressure regulator was used to reduce the pressure to 10 psi. To facilitate setting the flow rate a tappered tube flow meter was used to estimate the flow rate. The flow rate was controlled by a needle metering valve. A solenoid-operated valve was used to turn the LP gas on and off to the engine.

The LP gas was added to the intake air immediately upstream of the air cleaner through a fitting placed in the air stack. When the air cleaner was removed for the timing tests, the gas was added to the air in the air horn of the intake manifold.

Instrumentation

Both fuels were stored in containers on platform balances which weighed 0.01 pound, and the consumption rate was determined by timing with a stop watch the length of time for a given quantity of fuel to be consumed. Several readings were taken during each run and averaged to remove any timing error.

Air consumption was determined by a calibrated 2-inch diameter square-edged orifice before a surge tank. All tests, except those used to determine the optimum injection timing, were run with the air cleaner in place. The engine's crankcase ventilation system fed into the air cleaner so that the indicated air consumption was slightly less than the actual air consumed.

The lubricating oil temperature was taken by replacing the oil level dip stick with a copper-constantan thermocouple probe in the oil sump. A copper-constantan couple was also placed in the top radiator tank to obtain the coolant temperature. These thermocouples were connected to a calibrated potentiometer. The exhaust temperature was obtained by an iron-constantan thermocouple placed in the exhaust stream approximately 25 inches up the exhaust stack from the exhaust valves. The temperature was indicated on a calibrated millivolt pyrometer. The temperatures thus obtained are not absolute values to which the engine parts are subjected, but these values do indicate relative values and are valid for comparison with one another.

A water manometer was used to obtain intake vacuum.

The author built several models of smoke meters in an effort to obtain some measure of smoke produced by the engine. These smoke meters were based upon the principle of passing a light beam through the exhaust stream and measuring the amount of light passing through the stream by means of a photovoltaic cell connected to a millivoltmeter.

According to The Engineer magazine, if one can assume that the mean darkening effect due to the smoke particles can be expressed in terms of equivalent, equally distributed, spherical particles of uniform size, then

$$I/I_o = e^{-KmL}$$

where I_o = strength of light prior to entering the smoke

I = strength of light after passing through the smoke

L = distance of travel of light beam through the smoke

K = a constant depending on the size of particles

m = a measure of the quantities or density of the particles.

Diesel engines are usually limited in capacity due to the amount of smoke in the exhaust (1). It was therefore desirable to obtain some measure of smoke production, but all models constructed were unsuccessful because of carbon deposits over the light source and reading element. Attempts to use air pressure to keep these elements clean failed.

Plate I is a picture of the equipment used in this study of power fueling. The air measuring orifice with the surge tank may be seen at the right, the diesel fuel storage container on its balance is in the center behind the engine, and the propane container is at the left of the picture.

Procedure

In starting a series of tests, the engine was started and then brought up to maximum power at the desired speed and fuel setting. Then the coolant and oil temperature were allowed to reach equilibrium after which fuel consumption, exhaust temperature, coolant temperature, oil temperature, air consumption, torque, speed, air intake vacuum, and ambient conditions were determined.

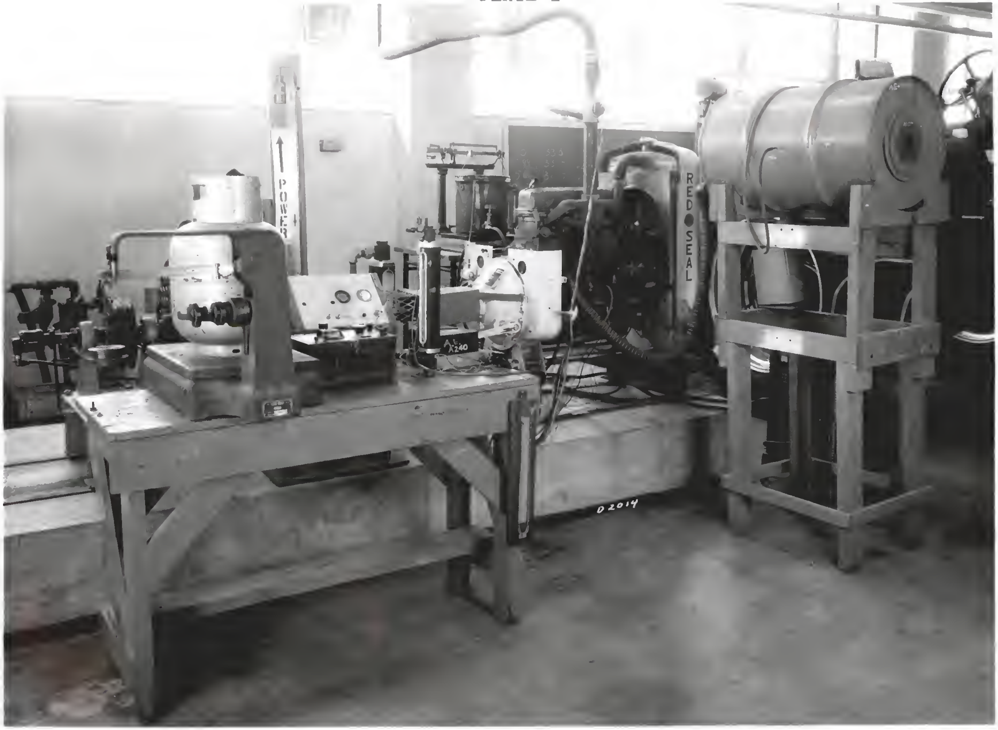
Results

Plate II shows the power curve and the specific fuel consumption curve obtained for the engine used in this study. It is interesting to note that the specific fuel consumption is independent of load for the power range tested. This is one of the principal claims for the Lanova type combustion chamber (2).

EXPLANATION OF PLATE I

Power fueling test equipment.

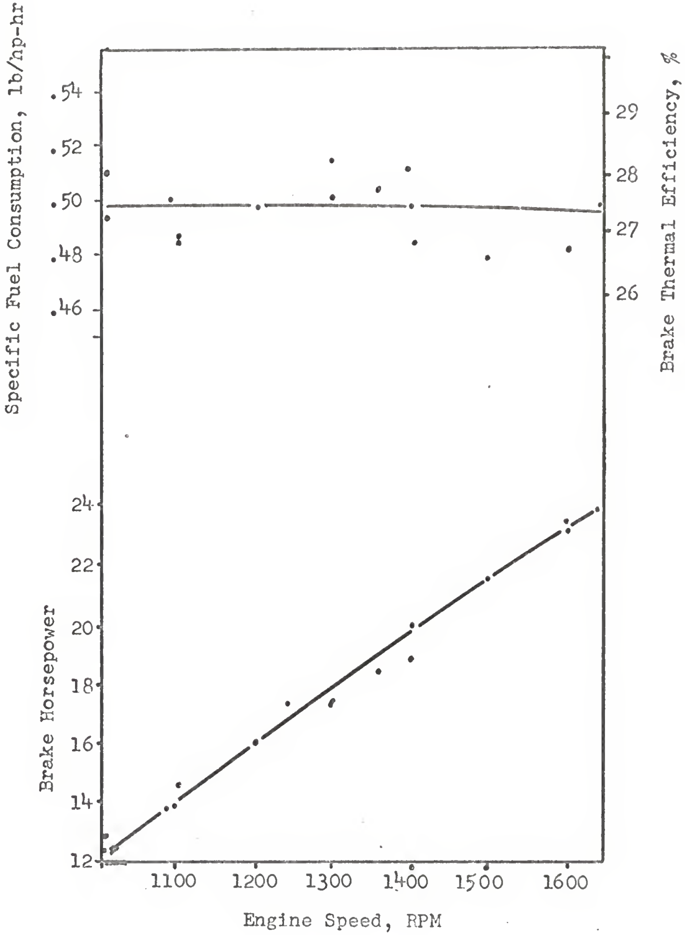
PLATE I



EXPLANATION OF PLATE II

Engine's characteristics as a conventional diesel.

PLATE II


Specific Fuel Consumption, lb/hp-hr
.54
.52
.50
.48
.46
Brake Thermal Efficiency, %
29
28
27
26
Brake Horsepower
24
22
20
18
16
14
12
1100
1200
1300
1400
1500
1600
Engine Speed, RPM

The engine's injection timing could be varied. The manufacturer's recommended timing for conventional operation is 24° and 22° as specified in the engine's instruction manual when the maximum governed speed is 1600 rpm and 1400 rpm, respectively. In order to determine if performance could be improved during power fueling by changing the injection timing, a series of tests were run at 1500 rpm with the injection timing at various settings and two different rates of power fueling. Plate III shows the results. Curve A is for the conventional diesel without any LP gas; curve B is for 1 lb/hr of LP gas consumption; and curve C is for 2 lb/hr. The diesel fueling rate was approximately 10.75 lb/hr for all cases since it is determined by engine speed and governor setting which was set at its maximum.

It appears that the optimum timing for best power is the same for all three fueling rates.

Plates IV and V show the relationship between fueling rate and brake horsepower. The fueling rate has been shown as BTU/hr in order to put both the diesel oil and LP gas on a common basis. It should be noted that, at both speeds, the rate at which the power increases for a given increase in the fueling rate decreases after the maximum power is reached for the conventional diesel operation. This indicates that the thermal efficiency starts to drop off with an increasing level of power fueling.

Plate VI shows this as a plot of brake thermal efficiency against per cent power where the maximum power for the conventional diesel operation is the basis. The results at both speed levels fall along the same curve.

EXPLANATION OF PLATE III

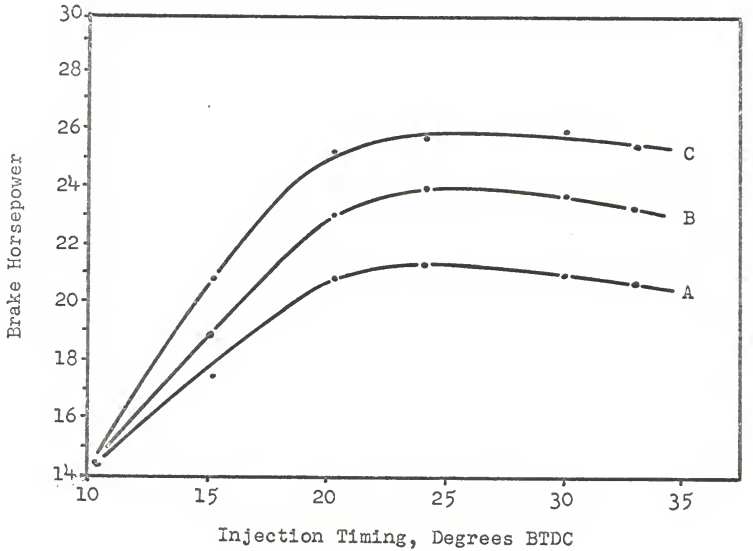
Effect of injection timing on power.

A. Conventional diesel operation--10.75 lb/hr diesel oil

B. Power fueling--10.75 lb/hr diesel oil, 1 lb/hr LP gas

C. Power fueling--10.75 lb/hr diesel oil, 2 lb/hr LP gas.

PLATE III

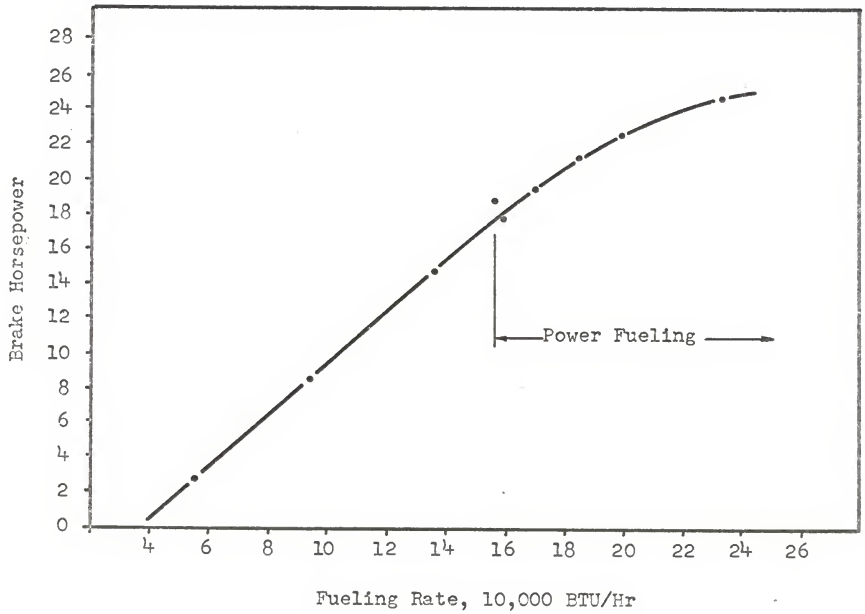

Brake Horsepower
30
28
26
24
22
20
18
16
14
10
15
20
25
30
35
Injection Timing, Degrees BTDC
C
B
A

EXPLANATION OF PLATE IV

Effect of fueling rate on power

engine speed--1275 rpm.

PLATE IV

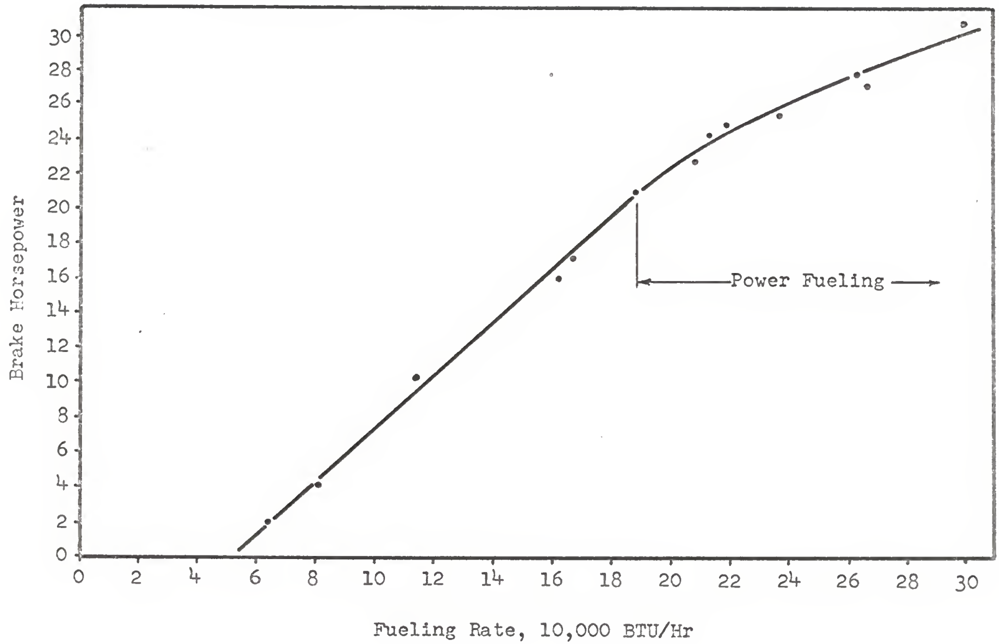

Brake Horsepower
28
26
24
22
20
18
16
14
12
10
8
6
4
2
0
4
6
8
10
12
14
16
18
20
22
24
26
Power Fueling
Fueling Rate, 10,000 BTU/Hr

EXPLANATION OF PLATE V

Effect of fueling rate on power

engine speed--1500 rpm.

PLATE V

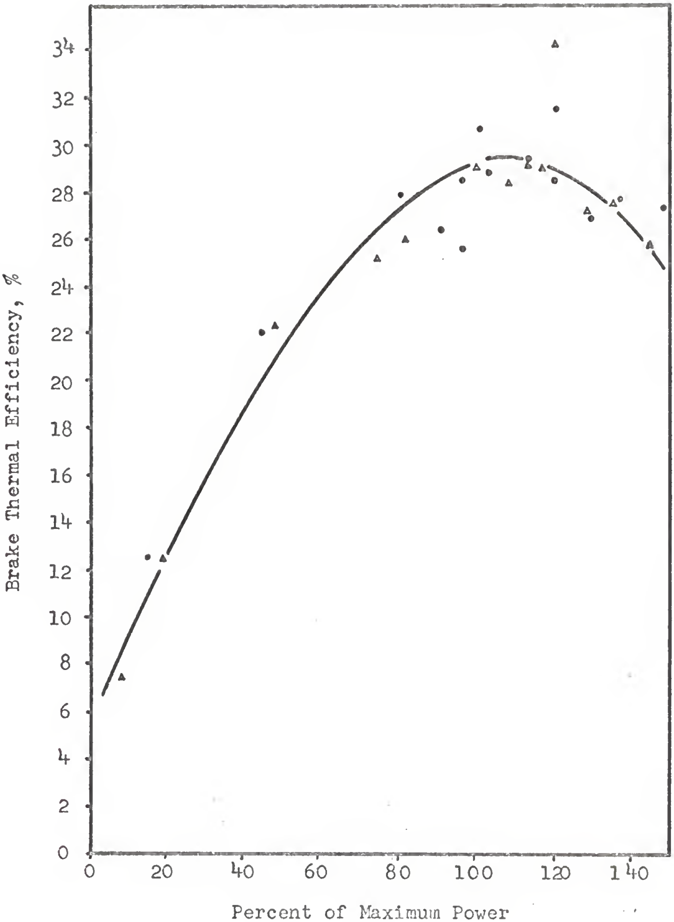


Brake Horsepower
0
2
4
6
8
10
12
14
16
18
20
22
24
26
28
30
Power Fueling
Fueling Rate, 10,000 BTU/Hr

EXPLANATION OF PLATE VI

Effect of power on efficiency.

PLATE VI



Brake Thermal Efficiency, %
0
2
4
6
8
10
12
14
16
18
20
22
24
26
28
30
32
34
0
20
40
60
80
100
120
140
Percent of Maximum Power

It was noted that during tests where the engine speed was below 1300 rpm and the power fueling rates were below about 1 lb/hr, there was a steady cracking sound in the exhaust. It was particularly noticeable when the power fueling rate had been higher and the engine warmer than was characteristic for the present setting. The engine speed fluctuated more than usual during this operation, and several times the measured power was below that obtained during conventional diesel operation. It is possible that this was the result of pre-ignition due to hot spots in the combustion chamber or to detonation due to the early injection of the diesel fuel, since the timing of injection was set for an engine speed of 1500 rpm. But neither of these explanations is entirely satisfactory because the operation occurred with extremely lean LP gas-air mixtures and did not occur with richer mixtures when higher power fueling rates were used.

What may be a related phenomenon was noted when LP gas was added immediately after the engine started. It is characteristic for a diesel to give off white smoke and run rough, indicating incomplete combustion, after it is started cold. It was thought that the addition of LP gas might improve the combustion during this warm-up period, but rates of over about 1/3 lb/hr caused the engine to stall.

However, when LP gas was added during idling and the engine was completely warm, it was noted that the intensity of the audible diesel knock was reduced considerably, which indicated a possible reduction in the ignition delay period and thus the amount of accumulated fuel oil at ignition.

Plate VII shows that power fueling and dual fueling do not cause an increase in exhaust temperature except as they cause an increase in the power output rate. Data with the injection timing varied are also plotted (circled). Retarding the timing causes a rapid rise in exhaust temperature.

An inspection of the air consumption data reveals that the effect of power fueling on air consumption is negligible. This is as might be expected because at the stoichiometric ratio propane occupies only 3.7 per cent of the air volume. At power fueling levels the fuel would make up less than 1 per cent of the volume of the air.

In an effort to determine the possibility of using dual fueling for stationary engines, such as irrigation or feed grinder power plants where natural gas or LP gas is available economically, several runs were made with LP gas replacing some of the diesel fuel.

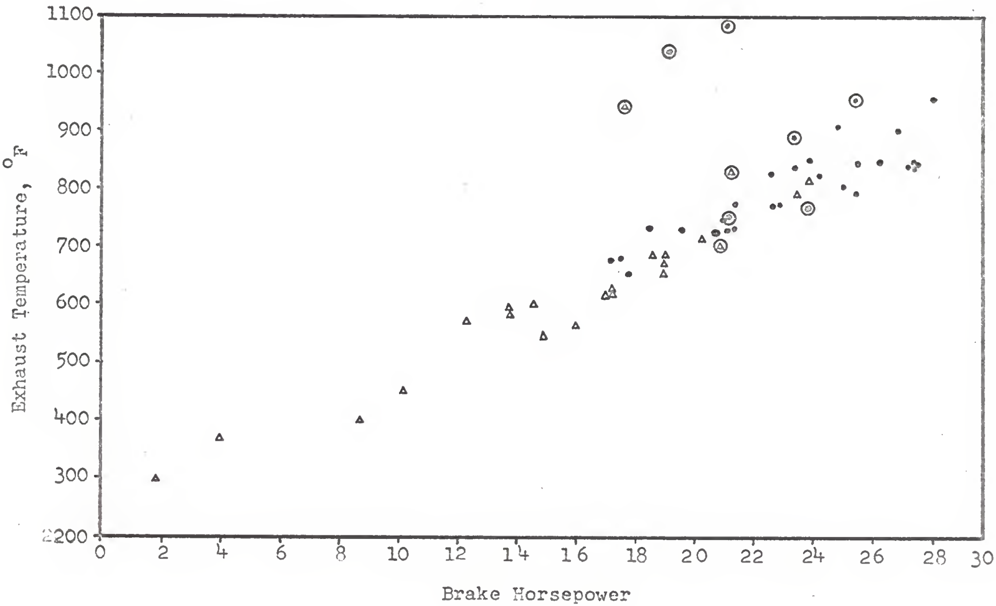
The maximum power at 1500 rpm was 21.5 hp as a conventional diesel. Fuel consumption was 10.3 lb/hr with a brake thermal efficiency of 29 per cent. With a fueling rate of 8.82 lb/hr of LP gas and 1.57 lb/hr of diesel fuel, the thermal efficiency dropped to about 26 per cent, but the other factors, such as air consumption, exhaust, oil, and coolant temperatures, remained about the same. The LP gas air ratio was .035, or 60 per cent of stoichiometric.

This trial was run immediately after one with a higher power output. Thus the engine was hotter than characteristic

EXPLANATION OF PLATE VII

Effect of power on exhaust temperature. Circled points indicate data taken with various injection timings.

PLATE VII


1100
1000
900
800
700
600
500
400
300
200
Exhaust Temperature, °F
0
2
4
6
8
10
12
14
16
18
20
22
24
26
28
30
Brake Horsepower

for the new power setting. A knock developed shortly after the beginning of the new fueling rate and the LP gas was turned off and then back on, and no knock developed.

This phenomenon was experienced on several other trials where the engine was hotter than characteristic for the fuel setting and the LP gas comprised about 40 per cent of the fuel or more. Whether the knock was due to pre-ignition or detonation is not known.

Conclusions

1. The addition of LP gas (commercial propane) to the intake air of a diesel engine increases the maximum power obtainable. A 13 per cent increase in fueling rate, on a BTU basis through power fueling, yielded a 9 per cent increase in maximum power, and a 26 per cent increase in fueling rate increased the maximum power 18 per cent.

2. There is a decrease in brake thermal efficiency with the introduction of a gaseous fuel into the intake air to increase the maximum power.

3. The exhaust temperature appears to be independent of the method of introducing the fuel to the combustion chamber, that is, whether a gaseous fuel is introduced into the intake air or fuel oil is injected by the diesel's conventional system.

4. The optimum injection timing is the same during power fueling operation as during conventional diesel operation. For

this engine the optimum injection timing is 24° before top dead center at 1500 rpm. There are indications, however, that at small power fueling rates and when the engine is lugged down, that is, when the timing is advanced beyond optimum, and the engine is hot, that the power fueling may reduce the power a very small amount.

5. Power and dual fueling have a negligible effect upon air consumption.

6. Dual fueling is possible, although it might be somewhat precarious, with as high as a 60 per cent of stoichiometric mixture of LP gas and air without knocking in an engine with a compression ratio of 15.5:1 and at rated power.

SUGGESTIONS FOR FURTHER RESEARCH

To obtain more basic data on the combustion during power fueling, and to determine the difference in peak cylinder pressures between conventional diesel operation and power fueling, it would be desirable to install a pressure transducer in the combustion chamber of an engine.

An investigation of the effect that the various types of combustion chambers, such as open chamber or precombustion chamber, have on power fueling should be considered.

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APPENDIX

Table 1. Engine Test Data.

		Fuel and air consumption, lbs/hr			Temperatures					
RPM	BHP	LPG	Diesel	Air	Cool-ant	Lub oil	Ex-haust	Dry bulb	Wet bulb	Baro-graph
1640	25	.97	11.70	251	183	198	783	90	78	28.7
1460	23	.97	9.93	223	184	197	765	84	67	28.9
1330	21	.98	8.92	206	183	197	720	84	67	29.0
1230	19	.84	8.06	190	185	193	725	85	74	28.7
1190	18	.88	7.77	185	184	192	675	89	74	28.7
1160	17	.58	7.70	180	184	194	670	89	74	28.7
1475	23	1.60	9.71	224	186	194	830	86	73	28.9
1600	27	1.43	11.57	242	190	197	900	89	73	28.9
1600	30	2.81	11.69	238	190	200	1000	89	73	28.8
1280	25	3.64	8.34	208	208	202	900	90	79	28.6
1275	22	2.18	8.24	190	202	202	820	95	79	28.6
1275	21	1.51	8.28	195	198	201	770	96	79	28.6
1275	20	.91	8.24	195	190	202	720	96	79	28.6
1275	18	.33	8.26	196	180	201	650	97	79	28.5
1500	28	3.32	10.25	220	200	208	950	92	77	28.7
1500	26	2.09	10.33	224	195	202	860	91	77	28.7
1500	24	1.08	10.19	229	192	197	820	90	77	28.7
1500	25	1.35	10.24	227	191	186	800	90	75	28.7
1500	23	.81	10.24	230	187	192	770	90	75	28.8
1500	21	2.00	7.84	235	184	192	700	91	75	28.8
1500	21	3.84	6.10	230	185	193	725	91	75	28.8
1500	21	2.93	7.06	230	186	194	735	91	75	28.8
1500	21	1.66	8.50	232	188	195	725	97	78	28.7
1500	21	.93	9.14	233	188	195	725	97	78	28.7
1100	18	1.08	7.34	204	185	190	700	89	74	28.8
1400	23	1.15	9.71	247	190	202	750	92	74	28.8
1500	27	3.09	9.58	256	197	209	840	92	74	28.8
1500	27	4.05	8.42	255	199	211	840	92	74	28.8
1500	27	4.35	7.40	254	200	213	835	92	75	28.8
1500	27	5.20	6.64	254	200	213	830	92	75	28.8
1500	27	7.00	2.63	251	203	216	840	92	75	28.8
1500	31	5.11	10.60	247	203	200	980	82	69	28.8
1500	29	3.32	10.62	246	196	200	940	82	69	28.8
1500	26	6.24	6.08	246	190	200	830	82	69	28.8
1275	26	4.36	8.01	226	210	204	910	92	71	28.7

Table 1 (cont.)

		Fuel and air consumption, lbs/hr			Temperatures					
RPM	BHP	LPG	Diesel	Air	Cool-ant	Lub oil	Ex-haust	Dry bulb	Wet bulb	Baro-graph
1275	24	3.27	8.09	229	204	204	875	92	71	28.7
1275	21	2.27	8.19	232	194	204	815	92	71	28.7
1275	17	1.22	8.21	233	181	198	820	91	71	28.7
1275	17	.52	8.57	235	180	193	740	91	71	28.7
1000	12		6.07	160	175	190	560	87	66	29.1
1600	13		11.71	278	182	199	820	83	69	28.9
1650	24		11.98	254	180	200	810	81	66	29.1
1300	17		8.46	199	179	194	620	85	66	29.1
1400	19		9.51	218	178	196	665	85	66	29.1
1100	14		7.18	177	176	192	580	87	66	29.1
1600	24		12.70	245	183	198	783	90	78	28.7
1400	20		9.91	217	180	195	715	84	67	28.9
1245	17		8.24	194	177	198	625	84	67	29.0
1160	14		7.44	182			600	89	74	28.7
1090	14		6.91	175	178	194	600	89	74	28.7
1360	18		9.18	211	180	190	685	85	73	28.9
1275	19		8.51	198	180	180	640	91	77	28.6
1275	15		7.37	200	176	189	535	91	77	28.6
1275	8		5.05	204	166	182	390	91	77	28.7
1275	3		3.05	211	148	170	270	91	77	28.7
1500	4		4.43	249	160	175	360	88	75	28.8
1500	2		3.50	251	153	170	295	90	77	28.8
1500	17		9.02	235	190	178	620	89	77	28.8
1500	10		6.23	242	167	184	450	90	77	28.7
1500	22		10.20	233	180	191	720	91	75	28.8
1500	16		8.81	237	172	188	565	91	75	28.8
1400	20		9.51	249	180	193	720	83	69	28.9
1275	16		8.40	233	175	195	700	91	71	28.7
1000	13		6.28	181	174	185	600	83	69	28.9
1100	14		7.03	200	176	185	620	83	69	28.9
1200	16		7.97	214	178	189	650	83	69	28.9
1300	17		8.72	233	180	190	655	83	69	28.9
10 Degrees BTDC injection timing										
1500	14		10.50	Misfired						
1500	14	1.00	10.50	Misfired						
1500	14	2.00	10.50	Misfired						

Table 1 (concl.)

RPM	BHP	Fuel and air consumption, lbs/hr: LPG	Diesel	Air	Temperatures: Coolant	Lub oil	Exhaust	Dry bulb	Wet bulb	Barograph
			15 Degrees BTDC injection timing							
1500	18		10.84	268	89	77	178	198	935	28.78
1500	19	.97	10.86	265	89	77	181	198	1035	28.78
1500	21	1.94	10.62	262	89	77	183	198	1080	28.78
			20 Degrees BTDC injection timing							
1500	21		10.72	267	89	76	178	200	820	28.78
1500	23	.97	11.01	263	89	76	183	201	880	28.78
1500	25	2.05	10.56	260	86	76	186	202	940	28.78
			24 Degrees BTDC injection timing							
1500	22		10.20	233	91	75	180	191	720	28.81
1500	24	1.08	10.19	229	90	77	192	197	820	28.77
1500	26	2.09	10.33	224	91	77	195	202	860	28.77
			30 Degrees BTDC injection timing							
1500	21		10.80	265	95	79	184	208	700	28.74
1500	24	1.02	11.01	260	95	79	189	210	765	28.74
1500	26	2.04	10.87	256	95	79	195	210	845	28.74
			33 Degrees BTDC injection timing							
1500	21		11.21	262	100	79	190	210	750	28.83
1500	24	1.00	11.08	262	100	79	192	213	830	28.84
1500	26	2.08	11.03	259	101	78	196	211	845	28.84

DUAL FUELING AGRICULTURAL SIZE DIESEL ENGINES

by

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Diesel powered agricultural tractors are making up a larger proportion of the tractors produced each year. With a diesel engine it is possible to increase the maximum power obtainable by carbureting LP gas into the intake air. This investigation was made to determine the effect power fueling an agricultural size diesel engine has on its economy and its maximum power.

It was found that the increase in maximum power was obtained at the expense of overall brake thermal efficiency. It can be inferred from the higher operating temperatures that engine life would be shortened. Increasing the fueling rate 13 per cent, by power fueling, increased the maximum power 9 per cent. A 26 per cent increase in the fueling rate increased the maximum power 18 per cent. The optimum injection timing proved to be the same during power fueling operation as during conventional diesel operation.

In investigating the possibility of dual fueling agricultural size diesel engines, it was found that LP gas could replace part of the diesel fuel in a conventional diesel engine with a compression ratio of 15.5:1, where knock-free operation was obtained with LP gas mixtures as high as 60 per cent stoichiometric.