THE APPLICATION OF HYDROGEN TO AN AGRICULTURAL INTERNAL COMBUSTION ENGINE

by

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NOMENCLATURE

$\emptyset$	=	Equivalence Ratio
η_t	=	Thermal Efficiency
r_v	=	Compression Ratio
k	=	Specific Heat
R.P.M.	=	Revolutions Per Minute
LP-gas	=	Liquid Petroleum Gas
^{o}BTDC	=	Degrees Before Top Dead Center
^{o}ABDC	=	Degrees After Bottom Dead Center
^{o}BBDC	=	Degrees Before Bottom Dead Center
^{o}ATDC	=	Degrees After Top Dead Center

INTRODUCTION

Recent experiences with oil embargoes and natural gas shortages have created an increasing concern with the future availability of energy. As a result of these happenings, many people realize that our fossil fuels are limited in supply and eventually new sources of energy will need to be developed. For the agricultural economy, the availability of sufficient quantities of fuel at the appropriate times is a necessity for maintaining high levels of food production. To replace agriculture's heavy reliance on natural gas and petroleum products, new, more reliable energy sources need to be developed.

Presently, the internal combustion engine represents the major instrument for converting chemical energy to mechanical energy on the farm. The pumping of irrigation water, powering of field operations, and transporting of produce to market all draw on the energy provided by the internal combustion engine. One possible source of chemical energy to replace hydrocarbon fuels in the internal combustion engine may be hydrogen.

Much work has been done in the field of hydrogen fueled engines. However, these studies have concentrated on its application to the automobile and other light duty vehicles. This thesis will focus on the application of hydrogen to powering heavier duty engines such as those used in irrigation and field operations. The ability of a hydrogen engine to meet the specialized requirements of these applications will be a major consideration in the selection of an engine performance test program.

LITERATURE SURVEY

The use of hydrogen in an internal combustion engine has mystified researchers as early as the 1820's. Reverend W. Cecil first proposed the use of hydrogen in an internal combustion device that harnessed the decrease in molar density resulting from the combustion of hydrogen and oxygen. Serious research into this topic began during the early part of the 20th century with the work of H. R. Ricardo, A. F. Burstall, and Rudolph Erren among others. All of these individuals confronted the problem of backfiring at fuel-air mixtures that approached stoichiometric conditions. Erren achieved control of this problem by injecting hydrogen into the cylinder during the compression stroke. However, this involved extensive modification and some complex auxiliary equipment (Billings and Lynch, 1973).

Work by R. O. King during the 1950's provided successful operation on hydrogen at high compression ratios without injection equipment (King, et al. 1958). Operation at a compression ratio of 20:1 was possible only at lean fuel-air mixture operations. However, the highest compression ratio at which maximum power could be attained was 14:1. King noted several possible causes of backfire as being hot spark plugs, exhaust valves, and oil accumulation in the cylinder. King speculated that the accumulation of oil may result in small high temperature particles in the residual gases which would ignite the incoming mixture (King, et al. 1948 and 1955). This hypothesis will be further discussed later in this thesis.

Energy and environmental interests have sparked a renewed interest in hydrogen combustion. Recent research efforts have involved three classes of engines: (1) automotive multi-cylinder engines, (2) small general utility engines, and (3) laboratory research engines. The general trend of research is to adapt hydrogen to automotive or other light duty internal combustion engines. Engine performance, exhaust emissions, and induction manifold backfire control have been the major areas of study.

The most extensive work with hydrogen engines has been performed by Billings Research Corporation (Escher, 1975). Feasibility demonstrations of the automotive applications of a hydrogen engine utilizing a wide variety of backfire control methods have been the major reported emphasis by this corporation. Besides more common backfire control methods such as water induction and exhaust gas recirculation, an increase in the combustion chambers surface area to volume ratio has been utilized (Lynch, 1975). A change of this nature will allow quicker cooling of the residual gasses which will hopefully eliminate backfire from hot particles in the residual gases. Only limited performance data has been released by this corporation. An 86% increase in efficiency of a Monte Carlo converted to hydrogen when compared to gasoline operation was reported by Billings for urban driving conditions.

A research team at UCLA has been heavily involved with the demonstration of hydrogen in automotive and laboratory test engines (Finegold, et al. 1973, Finegold and VanVorst, 1974). For one particular automotive test engine, brake thermal efficiency increases of 25 to 100% and 90% reductions in oxides of nitrogen emissions were reported for the engine operated on hydrogen (quality governed) as compared to gasoline operation. Finegold and VanVorst (1974) also noted that "Some form of charge dilution is essential to permit operation with hydrogen at high power output."

Without charge dilution, maximum horsepower attained with hydrogen was 40% less than attained on gasoline (Finegold and VanVorst, 1974).

A rather unique approach to quality governing of a hydrogen automotive engine has been demonstrated by Swain (1973) of the University of Miami. Hydrogen was fed to the cylinder separately from the air by a tube that opened into the intake valve seat. Thus, the intake valve controlled the separate openings for both the air and hydrogen. This separation of air and hydrogen was an attempt to reduce the chance of severe backfiring in the intake manifold. However, low-level backfiring continued to occur frequently if oil deposits were allowed to build up on the exhaust valve. Swain attributed the initiation of backfiring to preignition beginning at the sodium filled exhaust valve. Reported operation of this hydrogen engine never exceeded 50% of stoichiometric conditions. Efficiency of this system averaged about 50% greater than gasoline operation.

Variations of the hydrogen engine have also been experimented with by a number of institutions. General Moters Laboratory and the Jet Propulsion Laboratory of Pasadena, California are investigating hydrogen addition to gasoline in an attempt to improve fuel economy and reduce emissions (Escher, 1975). Perris Smogless Automobile Association is pursuing the development of a hydrogen-oxygen power system in order to completely eliminate harmful emissions (Underwood and Derges, 1971). A number of other institutions are also developing hydrogen power units. To avoid repetition, their findings will not be discussed.

One further area of work worth mentioning involved the application of hydrogen to compression ignition engines. Karim, Rashidi, and Taylor

(1974) of the University of Calgory in Canada have made extensive theoretical studies of the compression ignition characteristics of hydrogen-air mixtures inducted through the intake of a reciprocating engine. For this situation it is critical for the ignition delay period during the compression cycle to allow autoignition to occur at a time when the pressure rise will create peak performance. Karim noted that only a relatively narrow range of intake fuel-air mixtures and temperatures will allow acceptable timing of the pressure rise. He states that "acceptable operation with air appears possible only within a relatively narrow equivalence ratio range which is even more restrictive than with similar conditions involving spark induced flame propagation."

Compression ignition usually involves timing of the pressure rise by means of controlling the time the fuel is injected directly into the cylinder. Hydrogen injection has been used by several research groups recently. Compression ignition under these circumstances could not be achieved by a research group at Cornell University despite the use of compression ratios up to 29 to 1. Their conclusion was that "ignition lag time apparently is too long compared with the time available." R. G. Murray of Oklahoma State University has reported successful compression ignition with direct cylinder injection of hydrogen. However, no details have been released. Billings has reported compression ignition to be possible by mixed diesel/hydrogen injection but very little information is available (Escher, 1975). Karim and Klat (1976) have used hydrogen induction and diesel injection in a dual fuel engine to achieve satisfactory compression ignition operation. Their findings indicate that stable operation lies in a narrow range of diesel and hydrogen mixtures. The controlling factors are excessive pressure rises resulting in knock and erratic ignition. Presently available information indicates that compression ignition of hydrogen alone is impractical, but the use of a diesel pilot fuel with inducted hydrogen holds some potential.

The use of hydrogen injection also offers certain advantages in spark ignition engines. The primary purpose of hydrogen injection at Oklahoma State and Cornell is to prevent backfiring by timing injection to occur after the closing of the intake valve. This practice also eliminates the losses in volumetric efficiency and power experienced by a naturally aspirated hydrogen engine. Hydrogen's low volumetric energy density results in 29.6% of the cylinder volume being occupied by hydrogen at stoichiometric conditions when it is inducted through the intake manifold. Twenty to twenty-five percent less power can be expected from the same engine when operated on hydrogen in comparison to gasoline. Hydrogen injection will not only recover this loss of power, but can also have a supercharging effect. An increase in power of 10 to 20% above a gasoline baseline, can be expected with hydrogen injection (Escher, 1975).

There are certain problems to be expected with hydrogen injection. This practice requires relatively high-pressure hydrogen supply and sophisticated timing and flow control hardware. Relatively low thermal efficiencies have been reported with hydrogen injection engines due to the energy requirements of the injection process (Murray, et al. 1972). Space for location of an injector on many present spark ignition engines may also be a problem.

Much activity in the development of hydrogen power units has occured in recent times. Most of this work has studied the application of hydrogen to light duty automotive engines. Increases in efficiency have generally been noted. Backfiring and low volumetric energy densities have caused reductions in power levels achieved as compared to hydrocarbon fuels. A number of backfire control methods have been used in an effort to eliminate this problem. Hydrogen injection seems to hold some promise in eliminating backfire and power losses. However, compression ignition of hydrogen seems to be impractical.

INVESTIGATION

Objectives

The major objectives of this study are as follows:

1) Determine the applicability of hydrogen to an agricultural internal combustion engine.

2) Document the performance of an internal combustion engine while operating on hydrogen and LP-gas separately.

3) Examine the effectiveness of water induction for controlling backfiring.

4) Investigate the origin of backfiring in a hydrogen engine and other noteworthy combustion characteristics.

The primary purpose of this research is to determine the applicability of hydrogen to agricultural internal combustion engines. In particular, hydrogen use in a tractor or irrigation engine will be considered. For hydrogen to be accepted initially, presently used farm engines will need to be converted to hydrogen. The ease with which this can be accomplished will be of major importance. Consideration will be given on all modifications as to the services locally available to a farmer to make these modifications. Normally a machine shop or equipment dealer with some machine shop capabilities is available.

Major consideration will also be given to the performance of an engine operating on hydrogen. Specific requirements of an agricultural engine must be considered. Of course, for any farm engine available, power and fuel economy are important factors to consider. The reduction of power that can be expected in the conversion from a hydrocarbon fuel to hydrogen will also be a concern to the farmer. When considering the suitability of hydrogen to a tractor's power unit, the ability of the engine to react to momentary overloads is very important. Operation of an irrigation engine will be normally under constant speed and load conditions.

A third primary objective involves the application of water induction for controlling backfiring. This method of charge dilution seems to be the simplest potential method of promoting smooth engine operation with a minimum effect upon performance. The necessary rate of water induction to prevent backfiring and detrimental effects upon performance will be investigated.

Finally, an attempt will be made to determine cylinder pressure characteristics and temperatures at various critical points in the cylinder. Hopefully, this information will provide some insight into the cause of backfiring and other peculiar combustion traits of a hydrogen fueled engine. The effect of equivalance ratio and water injection on these parameters will be assessed.

Theory

The characteristics of hydrogen are quite distinctive from hydrocarbon fuels. Careful consideration of these characteristics is necessary before one can explain some of the peculiarities of a hydrogen engine performance. This section will include an explanation of these features of hydrogen and their effects upon engine power, efficiency, preignition, and backfiring.

Control of Backfiring

One of the first problems that most researchers encounter with hydrogen combustion in an internal combustion engine is backfiring. This nagging and possibly destructive phenomena restricts fuel-air mixtures to less than about $\emptyset$=0.5. Control of this problem is necessary before one can achieve the maximum potential power from a hydrogen engine.

Three basic characteristics of hydrogen have been identified as contributors to the problem of backfiring (Table 1). The low ignition energy required to begin combustion of hydrogen-air mixtures makes it susceptible to small heat sources. Once combustion starts, it is very likely that it will continue due to the high flame speed and minimal quenching distance of hydrogen in air. Table 1 indicates that these properties vary greatly from common hydrocarbon fuels.

Several sources of sufficient thermal energy to initiate a back-lashing have been indicated by past research. "Hot spots" in the combustion chamber, such as the spark plug electrode, exhaust valve, casting projections in the head, and carbon deposits in the cylinder may initiate preignition which could lead to backfiring. Elimination of these hot spots includes replacement of present engine components with

Table 1. Properties of Several Fuels*

	Hydrogen	Methane	Propane	Gasoline
Auto Ignition Temperature (^oC)	585	540	510	440
Minimum Ignition Energy (mJ)	0.02	0.28	0.25	0.25
Maximum Flame Velocity Laminar (cm/sec)	270	38	40	30
Quenching Distance (cm)	0.06	0.22	0.19	----
Lower Heat of Combustion (Joules/gram)	119,900	50,020	46,360	44,200
Joules/cm^3 at 20^oC and 76.00cm of Hg	10.05	33.35	84.98	----
Stoichiometric Mixture Volume % in Air	29.6	9.5	4	1.7
Flammability Limits Volume % in Air	4-75	5-15	2.2-9.5	1.3-7.1

*Most of this information is taken from VanVorst and Finegold.

sodium filled exhaust valves and cooler operating spark plugs. Also, removal of any casting projections or carbon deposits in the cylinder may be helpful. However, these efforts are often not sufficient for control of backfiring despite the fact that they are no longer a source of preignition except at high compression ratios and near stoichiometric conditions (King, 1955). It appears that backfiring is not always initiated by preignition.

The hypothesis promoted by King (1948) and later, Lynch (1975), explains that particulate matter in the exhaust gas may cause backfire. The higher heat capacity and greater mass of particulate matter causes this source to remain at higher temperatures longer than the surrounding residual gases. The thermal energy of these sources could ignite the fresh fuel-air mixture entering the cylinder. In an internal combustion engine, oil leakage by the valves and piston represents the most likely culprit. Carbon particles often remain unburned and suspended in the residual gases thus providing the particulate matter for initiating backfire. It has also been shown that inert particles as well as combustible carbon particles in the intake charge could induce backfire (King, 1948). Thus, backfire due to particulate matter in the residual gases seems highly possible.

If an explosion due to a heat source is to occur, the heat source and the chemical reaction around the heat source must release more energy than is conducted to the surroundings. The addition of an inert substance which increases the heat capacity of the mixture surrounding the heat source will tend to slow the thermal reaction. The mixture of fuel, air, and inert substance is able to absorb greater quantities of heat before and during a chemical reaction about a heat source, thus reducing the possibility of an uncontrolled explosion or backfire.

Water represents one such substance that should act in this manner to resist backfire caused by particulate matter in the residual gases and hot spots in the combustion chamber. In prior work with hydrocarbon fuels, reduced flame velocities and lower peak combustion temperatures were experienced with water mixed into the fuel-air charge (Nicholls, et al. 1969 and Quader, 1971). It was also noted that the reduction in combustion temperatures from evaporation cooling of the water was very minor compared to the charge dilution affect due to the increased heat capacity of steam. This indicates that water can enter the cylinder as steam or liquid with very little effect on its degree of backfire control. Thus, an inert substance such as water, which has the ability to slow the chemical reaction, offers a potential means of backfire control.

One final cause of backfire has been related to induced sparking (Billings, et al. 1974). Parallel or crossed ignition cables can experience induced sparks. Normally, in a gasoline engine this would not represent a problem. However, these induced sparks can cause difficulties due to the smaller ignition energy of hydrogen. Properly grounded and shielded cables can be effective in reducing the magnitude and frequency of the induced sparks and eliminating backfiring due to this cause.

Quality Governing

Another distinct characteristic of hydrogen is its wide flammability limits (Table 1). Because of this peculiarity, engine horsepower output can be controlled over a wide range by varying the richness of fuel-air mixture. The only potential area of difficulty is due to hydrogen's inability to burn lean enough to allow the engine to idle.

This concept is quite different from a conventional spark ignition engine which relies on variation in the charge density as a means of power control. A throttle plate in the air intake controls the density of the relatively constant fuel-air mixture that enters the cylinder. The throttle plate has the disadvantage of creating a vacuum in the intake system during part loads. The energy necessary to maintain the vacuum will need to be provided by the fuel. At part load, this loss of energy can have considerable effect upon the engine's thermal efficiency. However, at loads near the engine's maximum power, the inefficiency due to the throttle is much less. This last situation represents more closely an agricultural power unit in a tractor or on an irrigation well. The control of an agricultural engine burning hydrogen by varying fuel-air mixture may hold only limited advantage oyer a throttle controlled engine.

Performance

Some differences in a hydrogen engine's performance compared to operation on other common hydrocarbon fuels can be expected. As already mentioned, the fuel consumption efficiency can be improved by quality governing due to the elimination of the throttle. Examination of theoretical thermal efficiency of the Otto cycle reveals another major factor affecting the efficiency of hydrogen operation.

$$\eta_t = 1 - \frac{1}{r_v^{k-1}}$$

Thermal efficiency of the Otto cycle is dependent upon compression ratio, r_v, and the ratios of the specific heats, k, of the gases involved in the combustion process (Obert, 1973). For a specific engine with a fixed compression ratio, only the ratio of specific heat of the working fluid will influence the theoretical thermal efficiency. This factor is dependent upon the fuel, air, combustion gases produced during the expansion process, and the temperature of all of these fluids. As indicated by Figure 1, the ratio of specific heats for hydrogen and its products of combustion are higher than those of propane which should result in a higher efficiency for hydrogen combustion. Also, lean hydrogen operation should reduce the temperature of combustion and increase the ratio of specific heat of the products of combustion compared

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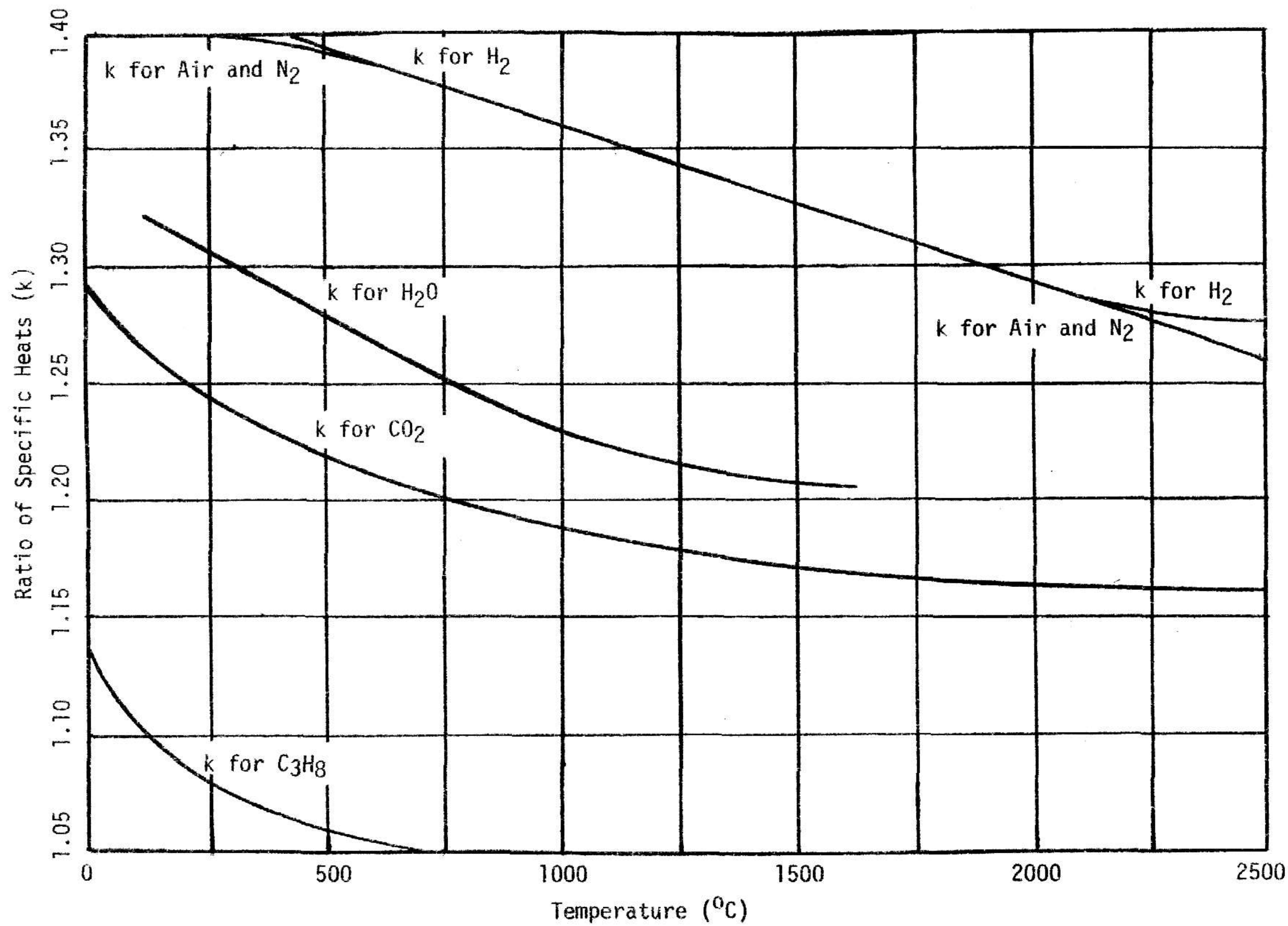


Figure 1: Specific Heats Ratio vs. Temperature

to stoichiometric operation of hydrocarbon fuels. Again, some increase in thermal efficiency should be expected.

Maximum power output of a naturally aspirated hydrogen engine should be less than a similar engine using hydrocarbon fuels. Hydrogen has a much lower volumetric energy density upon entering the cylinder than most other hydrocarbon fuels. As a result the volume of cylinder occupied by hydrogen at stoichiometric conditions is almost 30% of the total cylinder volume. By comparison, propane occupies only 4% of the cylinder.

The higher energy density of hydrocarbon fuels allows greater quantities of energy and oxygen necessary for releasing the energy to be present in the cylinder. The result is higher power output for hydrocarbon fuels such as LP-gas or gasoline. Finegold and VanVorst (1974) predicted that the power output of hydrogen should approach 75% of that obtainable with gasoline. A comparison of LP-gas and hydrogen based purely upon the heating value per volume of stoichiometric fuel-air mixture would lead one to expect hydrogen to achieve 88% of the power as achieved in the same engine on LP-gas. Of course, this does not consider the differences in factors affecting combustion pressure experienced during expansion which also affects maximum power levels.

Water induction into the intake manifold can have some major effects upon performance. Billings and Lynch (1973) found that rates of water induction up to five kilograms of water per kilogram of hydrogen were not detrimental to power or efficiency. Rates less than a five to one ratio have resulted in some increase in efficiency and power. However, performance deteriorated at rates greater than a five to one ratio. Water induction during operation on hydrocarbon fuels has produced similar results, but at different rates (Nicholls, et al. 1969).

An evaluation of the characteristics of hydrogen has provided several insights into its performance in an internal combustion engine. Ignition at inappropriate times may be a major problem due to the low ignition energy, high flame speed, and minimal quench distance of hydrogen. The addition of water should impede any chemical reaction in the intake mixture, thus reducing the possibility of untimely ignition. The performance and means of controlling hydrogen operation will also vary from conventional fuels due to a number of rather unique characteristics.

Modifications to the Engine

A number of modifications to LP-gas engines were necessary to allow operation on hydrogen. The engine used in this study is a 172 cubic inch Ford industrial engine. It was donated to Kansas State University by Ford Motor Company. This engine is similar to industrial LP-gas or natural gas engines used for irrigation pumping. This particular model engine was also used in Ford tractor model 881-L of the late 1950's and early 1960's. Further descriptive information on the engine is available in Appendix A.

The majority of the modifications made were for backfire control. A number of changes were made within the engine cylinder to eliminate potential "hot spots". The unmachined casting areas in the head were smoothed down. The Champion H-11 spark plugs used for LP-gas operation were substituted with a set of Champion H-8 spark plugs. The spark gap was also narrowed from 0.025 inch to 0.015 inch. The stock exhaust valves were also removed and replaced with sodium filled exhaust valves, VE-1251X, previously used in 272 and 292 in^3 Ford truck engines between 1956 and 1964. The stem diameter of the sodium filled exhaust valve was 0.4346 inches as opposed to 0.341 inch stem diameter for the original exhaust valve. A local machinist drilled and reamed the valve guide to accept the new exhaust valves. The face angle of the engine head and seat angle of the valve were both 45^o angles. This allowed for maximum heat transfer from the valve face into the engine head across the area of contact when it was closed. All other dimensions of the sodium exhaust valve were similar enough to the original exhaust valve that no other changes were made.

After some initial runs, excessive carbon deposits were noted in the cylinder under the valves. It was found that the rubber umbrellas used for controlling oil seepage between the valve and guides were ineffective. With some modification to the top of the valve guides, perfect circle seals were installed. This change diminished but did not eliminate the oil deposits in the cylinder. No further alterations were made to control oil seepage into the cylinder.

Despite these changes, backfiring was not eliminated. Water injection into the intake manifold just above the updraft carburetor was used. Initially, an attempt was made to inject the water in the form of steam by utilizing the heat of the exhaust gases to produce the steam. The simple heat exchanger consisting of copper tubing wrapped around the exhaust pipe and insulated with aluminum foil and an asbestos paper layer approximately 3/4 inch thick was constructed. The ease in obtaining a more uniform mixture of steam with the fuel-air was the major reason behind this decision. It was later found that the heat exchanger could not meet the needs for steam. Due to this and other minor problems, water instead of steam was used in all tests with hydrogen. The water induction flow rate was controlled manually.

During operation on hydrogen, it was discovered that the LP-gas carburetor did not allow large enough quantities of hydrogen to reach the cylinder. A jet controlled by a needle valve was determined to be the limiting factor. A second fixed size opening was drilled which allowed operation at maximum power levels.

With the substitution of spark plugs and the closing of the second jet of the carburetor, the engine would again run on LP-gas. The modifications that could not be reversed should not cause any measurable difference in engine performance. All modifications made could be done by a machinist. The needed parts were available through a local auto supply store. Most farmers should have these services available in a nearby community.

Other modifications were made to allow operation under laboratory conditions. The radiator and fan were not installed. Cooling was accomplished by tap water that passed through a heat exchanger to cool the engine water. The system thermostat was set by the factory at approximately 170^{o}F. The alternator was connected so that it was constantly charging the battery. The battery was always kept fully charged by a battery charger prior to any series of tests. The air cleaner also was removed. Some increased restriction was added to the air intake in the form of an air flow measurement nozzle and surge tank. However, the pressure drop due to the air flow measurement system was always less than three inches of water.

Measurement Systems

Figure 2 provides a layout of the test engine and test equipment used in this experiment. A listing of all test equipment, its purpose, and manufacturer can be found in Table 2.

The engine was loaded by a water brake dynamometer. The water was replaced by hydraulic oil to provide the necessary loading capacity for this engine. The dynamometer did not provide precise control of the load and speed of the engine. Therefore, all tests were run at any speed within a ± 10 R.P.M. of the designated speed.

The thermocouples were all located in the number one cylinder. The spark plug thermocouple was located in the insulator of the central electrode of the spark plug. The wall temperature measurement was made at a point in the cylinder head approximately one inch below and to the side of the spark plug. A thermocouple for a measurement of exhaust gas of cylinder number one was located at a point three inches from the exhaust manifold. An attempt was also made to measure temperature of the gases in the cylinder and the exhaust valve face temperature. However, both of these measurements proved unsuccessful.

The quartz pressure transducer for cylinder pressure measurement was made accessible to the cylinder by a passage into the spark plug that emerged near the central electrode. Since a special thermocouple plug was located in cylinder number one, the pressure transducer was placed in cylinder number four. In a four cylinder engine, cylinders one and four should receive approximately the same fuel-air mixture because of their similar proximity from the carburetor. This prompted the decision to place the transducer in cylinder number four.

The fuel consumption measurements presented some peculiar problems due to the light weights of hydrogen. A natural gas meter provided the means for measurement of both hydrogen and LP-gas. Because this was a volumetric measurement, fuel temperature and pressure were also recorded.

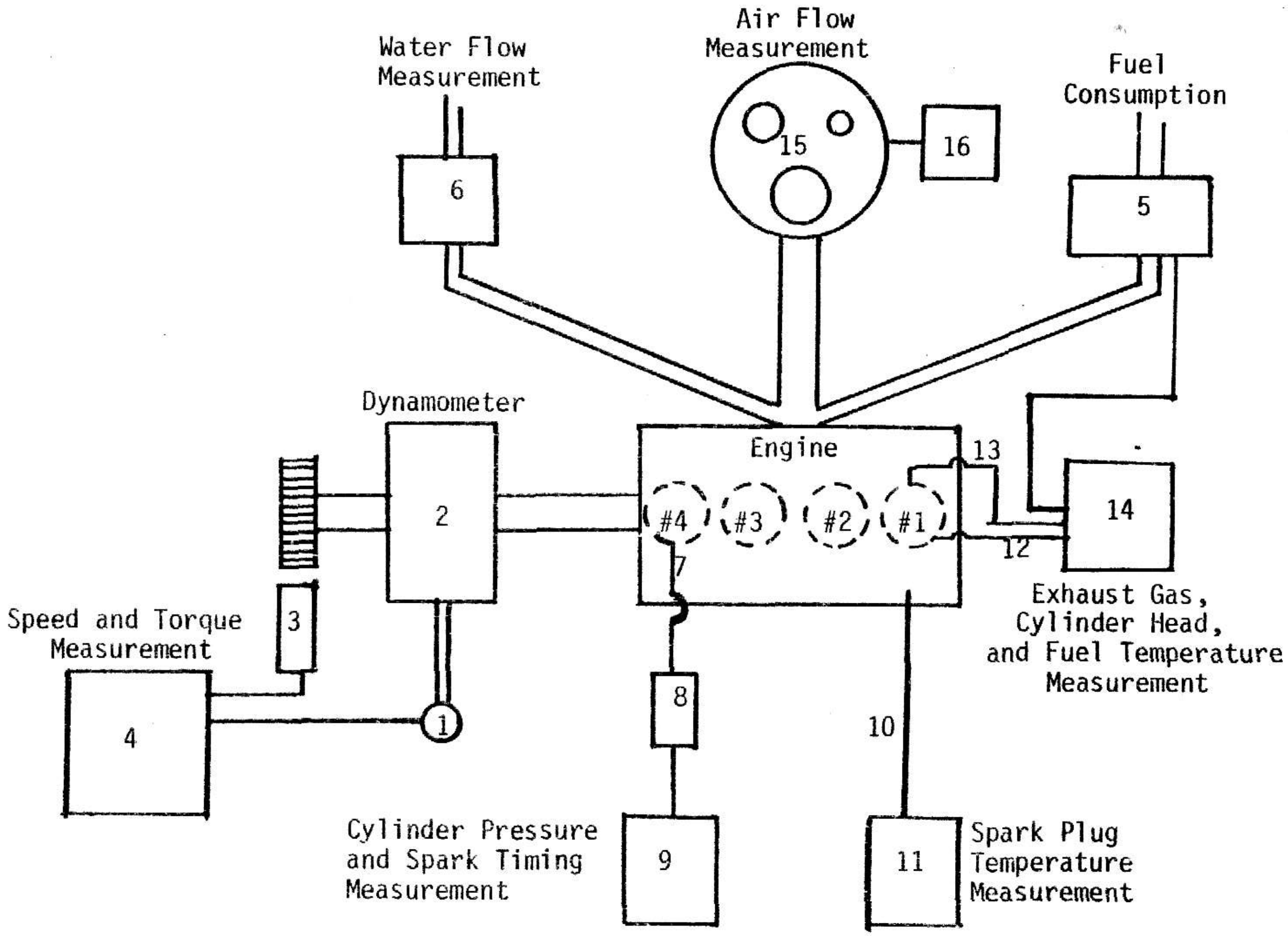


Figure 2: Schematic Diagram of Engine and Measurement Systems

Table 2. Listing of Test Equipment

	Purpose	Equipment Description	Manufacturer
1.	Torque measurement	Load cell	Transducers, Inc.
2.	Loading device	Water brake dynamometer	
3.	Speed of power take off shaft	Magnetic speed transducer	
4.	Readout of speed and torque	Digital indicator model DS-100-T4	Doric Scientific Corporation
5.	Fuel flow rate	Model 750 gas meter	Rockwell Manufacturing Co.
6.	Water induction rate	Flowrater	Fischer & Porter
7.	Cylinder pressure measurement	Model 601B quartz pressure transducer	Sundstrand Data Control, Inc.
8.	Charge amplification	Charge ampliphier model no. 566	Kistler Instrument Corporation
9.	Cylinder pressure trace and spark timing measurement	Dual trace oscilloscope with storage model #5103N	Teletronix, Inc
10.	Spark plug temperature	Platinum and platinum+ 10% rhodium thermocouple spark plugs, H-8 and H-11	Champion Spark Plug Company
11.	Spark plug temperature readout	Millivolt potentiometer model #8686	Leed and Northrup Company
12.	Cylinder wall temperature	Chromel constantan thermocouple probe	Medtherm Corporation
13.	Exhaust gas temperature	Iron constantan thermocouple probe	
14.	Cylinder wall & exhaust gas temperature readout	Digitec model 1268 data logger	United Systems Corporation
15.	Air flow measurement	AMCA air flow nozzles 1", 1.6" and 2"	Helander Spinning Company
16.	Pressure drop across nozzles	Micromanometer	Merriam Instrument Company

Test Procedures

The test program involved three distinctly different types of tests. The effects of variations in water induction rate, different loads at a constant speed, and maximum loads over a range of speeds were studied. All tests were conducted for both hydrogen and LP-gas operation except for the tests involving the effects of water induction. These were carried out for hydrogen operation alone.

The rate of water induction was studied initially to assess its affect upon the performance of the engine. This allowed us to determine the quantities of water that could be mixed with the fuel and air without detrimentally affecting the engines performance. At 1800 R.P.M., the engine was loaded to a point slightly prior to when backfiring began to occur without water induction. The rate of water induction was varied from zero to six kilograms of water for every kilogram of hydrogen burned. The necessary information for defining engine performance, equivalence ratio, cylinder temperature data, and cylinder pressure characteristics were collected.

A second series of tests on LP-gas and hydrogen were conducted at a constant speed of 1800 R.P.M. for a number of loads. Two methods of controlling engine power during hydrogen operation were used. One set of tests employed the use of a throttle to control power similar to conventional spark ignition engines. Fuel and air mixture was maintained at approximately $\emptyset$=1.0. Quality governing of the engine by changing the setting of a needle valve was also utilized with the throttle in the wide open position. The major objective of these studies was to determine differences in engine performance. Data on cylinder temperatures and pressure characteristics were also collected for quality governed hydrogen and LP-gas operation. A water induction rate of four kilograms of fuel was used for both methods of hydrogen operation. A set of tests using no water was also conducted for quality governed hydrogen operation.

The final series of tests were conducted at maximum power over a wide range of speeds. Again, engine performance characteristics of hydrogen and LP-gas operation were of primary interest. A water induction rate of four kilograms of water for every kilogram of hydrogen was used except when more water was needed to control backfiring.

A specific procedure was used in preparation for conducting a series of tests. Initially, the engine was warmed up for approximately twenty minutes on LP-gas at a moderate power level. After this period, the engine either remained on LP-gas or was quickly switched to hydrogen.

If throttled hydrogen or LP-gas operation was to be investigated, the fuel-air ratio was adjusted to the leanest possible mixture at which maximum power continued to be achieved. This adjustment was made at a fully loaded speed. If hydrogen operation was to be quality governed, the fuel-air richness was set at some desired level and the throttle was placed in the wide open position.

The engine was then set at a desired speed and power level. Spark timing was adjusted to the minimum spark advance at which maximum brake torque could be maintained. Speed and power were again checked to make sure they remained at the desired level. The engine was then allowed to run for approximately five minutes in order to attain equilibrium conditions. At this time, the necessary measurements were taken. For each of the following tests while the engine is continuously running, the procedure is repeated except for the warm up period.

Results

A complete summary of the data is presented in Appendix B. The more important results are plotted in Figures 3 through 12. The collected information on power and torque is corrected to atmospheric pressure of 74.63 cm of mercury, dry bulb temperature of 29.4^{o}C, and wet bulb temperature of 17.5^{o}C. These conditions represent SAE standards for test engines. The correction of power and torque to these conditions was not applicable to part throttle operation. Thus, a comparison of LP-gas at part throttle to quality governed hydrogen operation which required wide open throttle was not accurate. For these comparisons, quality governed hydrogen operation was also corrected to similar atmospheric conditions as experienced during throttled LP-gas operation. Under all situations, brake thermal efficiency was not corrected because the effects of pressure, temperature, and humidity on a fixed jet carburetor are not predictable (Obert, 1973).

It should be noted that the correction factor requires that one know the friction horsepower of the engine. Frictional horsepower was obtained from Ford Motor Company based on tests run on a similar engine. This may cause some error in corrected brake torque and power although it should be very small.

All calculations of thermal efficiency for LP-gas operation were based on the heat of combustion of propane. The specifications of the manufacturer for the LP-gas used in the tests stated that the fuel must contain a minimum of 90% propane and a maximum of 5% propalene, 0.1% methane, 6% ethane and 2.5% butane. Since the exact fuel content was not known for individual shipments, it was assumed that the properties of propane would closely approximate the properties of LP-gas.

DISCUSSION OF RESULTS

Effect of Water Induction of Engine Performance

The initial stage of this investigation provided some insights on the effect of water induction rate on hydrogen engine performance. The results of two separate tests are shown in Figures 3 and 4. This information was the basis for choice of water induction rates for later tests. Before we examine this information, one should note that we encountered some difficulty in maintaining a constant mixture richness. The water induction affected the engine speed which in turn changed the hydrogen pressure entering the carburetor. An attempt was made to correct the hydrogen pressure, but some variation in fuel-air mixture was still noted. This factor seemed to have some affect upon power output.

Water induction seemed to cause a small rise in power output for rates less than 4 kilograms of water per kilogram of hydrogen as displayed by Figure 3. A slight leaning of the fuel-air mixture after the initial run without water induction may explain a lack of power increase for the series of tests displayed in Figure 4. However, if any increase in power does occur, it appears to be very small. The important factor to note is that power output is not adversely affected until water induction rates greater than four to five kilograms of water per kilogram of hydrogen are used. Even at rates of six kilograms of water per kilogram of hydrogen the loss of power is only about 3%.

Adverse effects on brake thermal efficiency were not noted for water induction rates up to six kilograms of water per kilogram of hydrogen. However, it appears that fuel efficiency may experience some

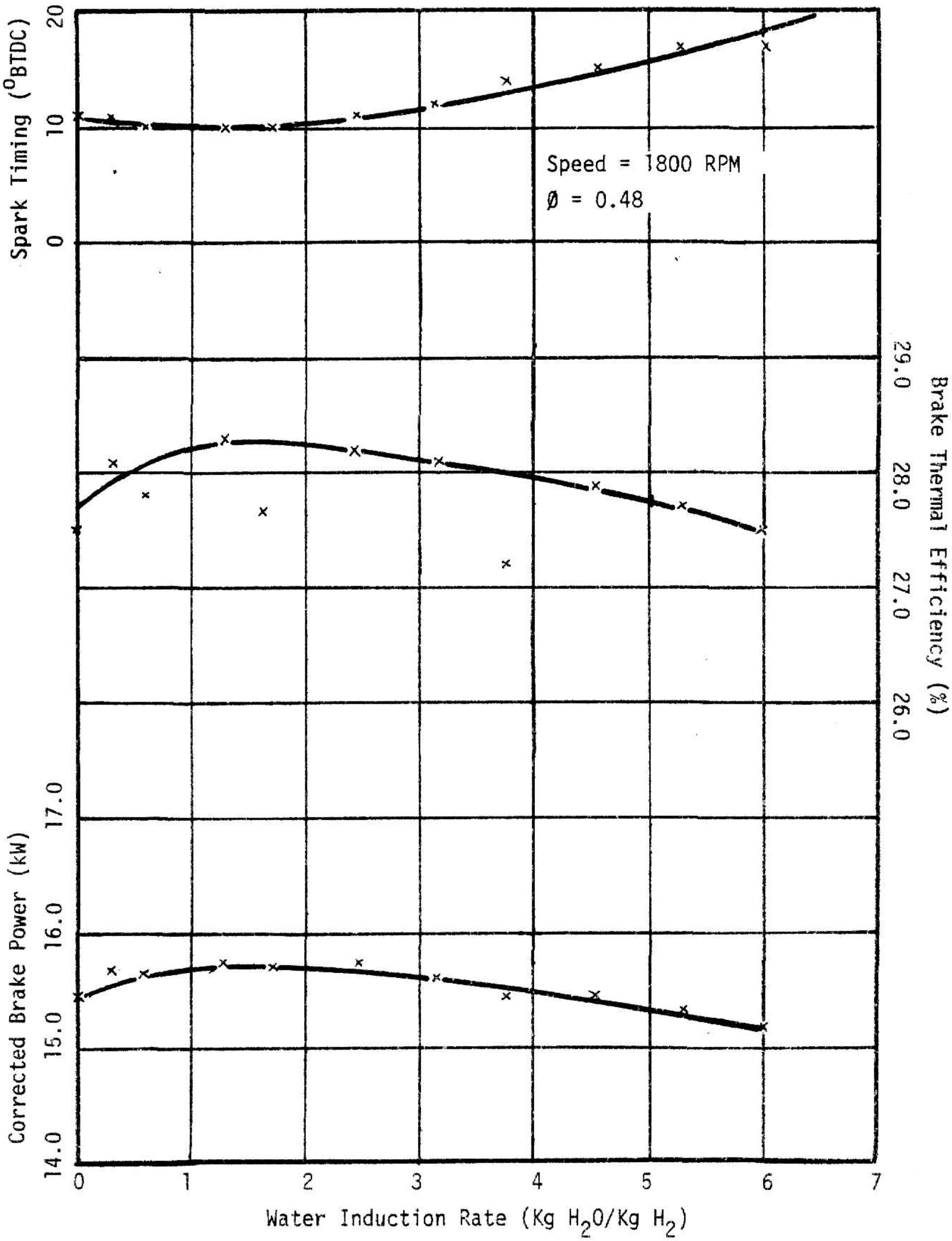


Figure 3: Variation in Engine Performance Due to Water Induction

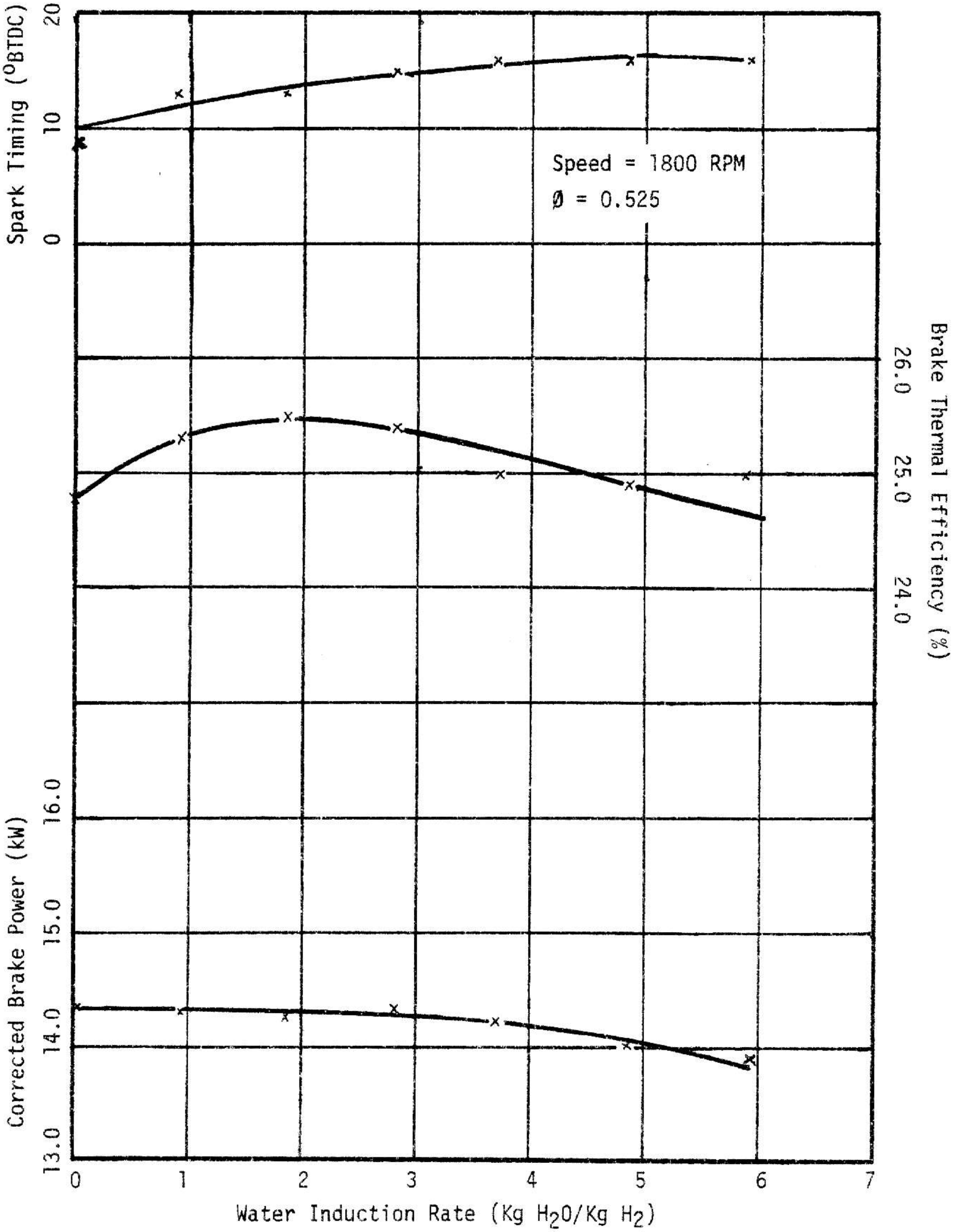


Figure 4: Variation in Engine Performance Due to Water Induction

decline for higher water induction rates due to the downward slope of the plot of efficiency occurring at the maximum tested water induction rate. Some improvement in fuel economy was noted at lower water induction rates. The greatest relative improvement in brake thermal efficiency was approximately 2 to 3% and was noted at an induction rate of two kilograms of water per kilogram of fuel.

Constant Speed Performance Tests

The next set of tests was designed to demonstrate the effect of hydrogen fuel-air mixture richness on power and fuel efficiency (Figure 5). Tests were conducted both for water induction at a rate of zero and approximately four kilograms of water per kilogram of hydrogen. A decrease in brake thermal efficiency and brake power was noted with water induction when compared to the runs without water induction for similar fuel-air mixtures. Based on our previous tests, it is difficult to attribute all of this change in performance to the water induction. However, no other reason was found to explain the differences in performance.

Backfiring during hydrogen operation with no water induction was noticed for all tests run at an equivalence ratio greater than 0.5. The graph also displays that backfiring prevented operation for mixtures richer than $\emptyset$=0.584. This series of tests were conducted shortly after the cylinder and head had been cleaned of all oil deposits. It was noted after the engine had run longer periods of time that backfiring began to occur at slightly leaner fuel-air mixtures than reported in this test. Without water induction, backfiring seems to limit hydrogen operation to mixtures less than 50% of stoichiometric conditions.

With water induction at a rate of approximately 4 kilograms of water per kilogram of fuel, backfiring never occurred at speeds of 1800 R.P.M. Under these conditions, a maximum brake power of 23.42 kilowatts was experienced. This peak occured at a fuel-air mixture slightly richer than stoichiometric conditions. It should be noted that power changes rapidly with mixture richness for lean operation but much slower for mixtures approaching stoichiometric conditions. Fifty and seventy-five percent of maximum power occurs at $\emptyset$=0.41 and $\emptyset$=0.60 respectively. The engine could be run at lower power levels than indicated by Figure 4 by quality governing. However, stable operation was difficult to maintain. It was impossible to idle the engine without using the throttle.

The peak brake thermal efficiency which occurred at $\emptyset$=0.65 was slightly greater than 28%. This is a slightly richer mixture for peak efficiency than what the literature indicates (Escher, 1975). The water induction may account for some of this difference. Without water induction, the maximum brake thermal efficiency appears to occur at an equivalence ratio between 0.5 and 0.6. At leaner fuel-air mixtures, efficiency drops due to the fact that friction represents a larger portion of the power generated. For mixtures richer than $\emptyset$=0.65, a drop in efficiency is also noted. This occurrence is a result of theoretical restrictions of the Otto cycle and additional heat losses due to higher combustion temperatures (Obert, 1973).

A comparison of performance of quality governed hydrogen, throttled hydrogen, and LP-gas operation at a constant speed of 1800 R.P.M. can be made by Figures 6 and 7. Throttle control hydrogen operation is more efficient than LP-gas operation for similar power levels. This advantage in fuel economy almost completely disappears at the maximum power level of hydrogen. An additional increase in thermal efficiency occurs by eliminating the throttle during hydrogen operation and relying upon quality governing. The gain in brake thermal efficiency of hydrogen over LP-gas combustion appears to be a result of the type of fuel being burned and the method of controlling power.

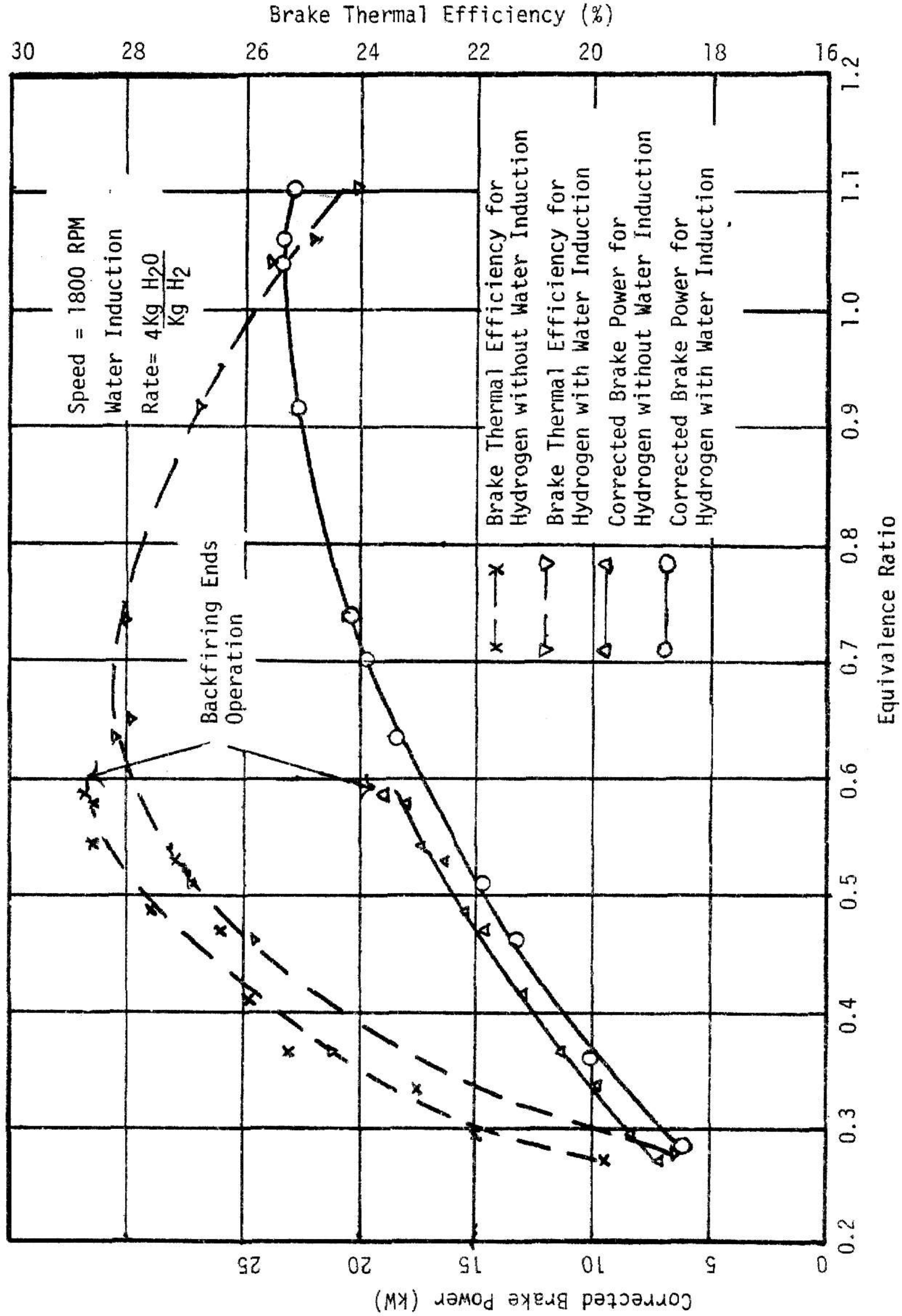


Figure 5: Performance vs. Equivalence Ratio at Constant Speed

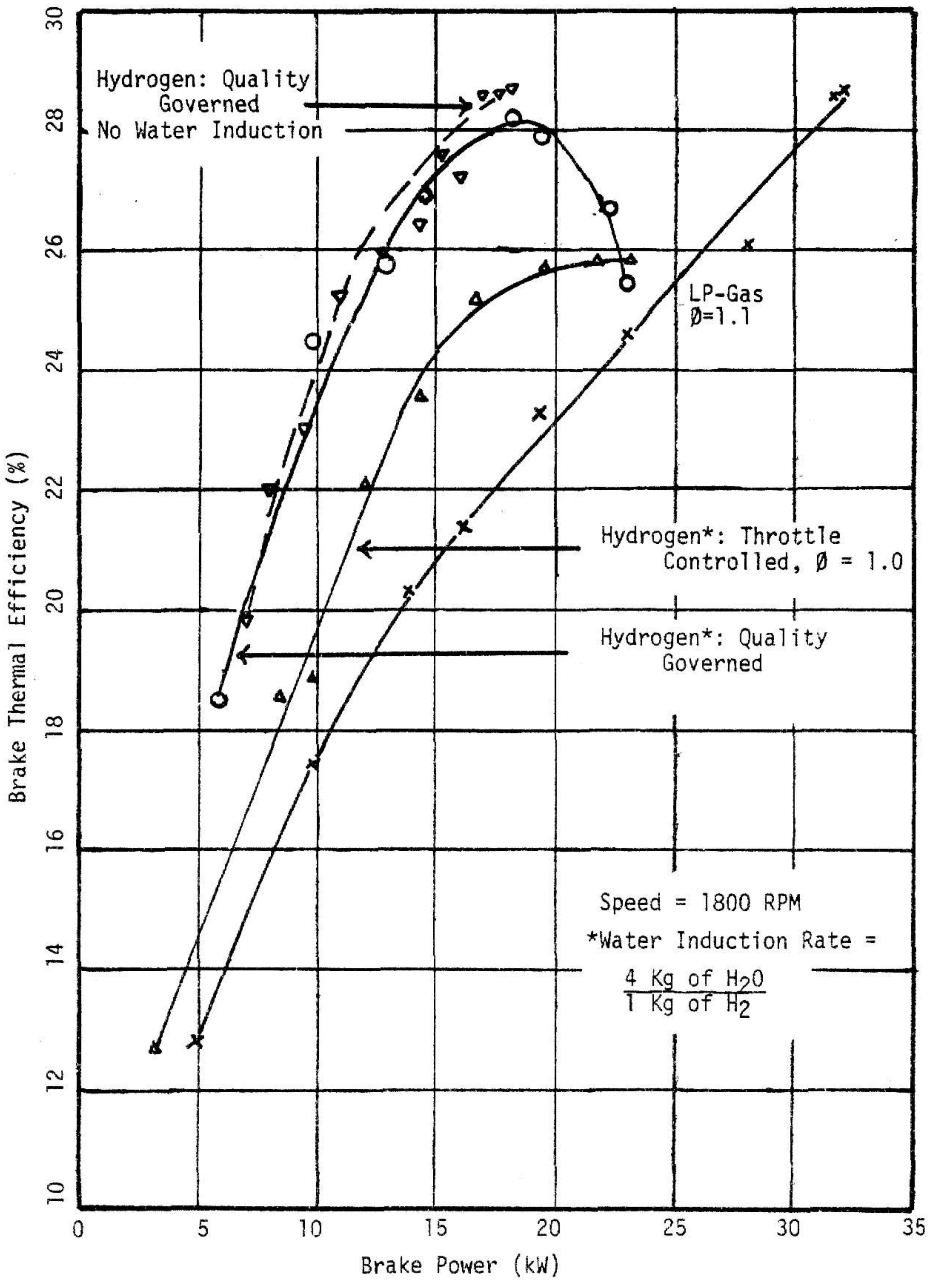


Figure 6: Power vs. Efficiency For Hydrogen and LP-Gas at a Constant Speed

The relative improvement in efficiency of quality governed hydrogen compared to throttled LP-gas amounts to a maximum of 35% at part loads. This advantage disappears as a hydrogen engine approaches its maximum power. At 23 kilowatts, maximum power for hydrogen at 1800 R.P.M., the advantage in efficiency of hydrogen over LP-gas operation has dwindled to approximately 5%.

The value of quality governing compared to throttled hydrogen operation is most evident at part loads (Figures 6 and 7). At maximum load operation for hydrogen, there is no advantage in fuel efficiency. But, for operation at any power levels less than 93% of maximum hydrogen power, quality governing will offer a relative brake thermal efficiency increase of 5% or more over throttled hydrogen operation. It appears that quality governing offers real fuel savings over a wide range of power levels.

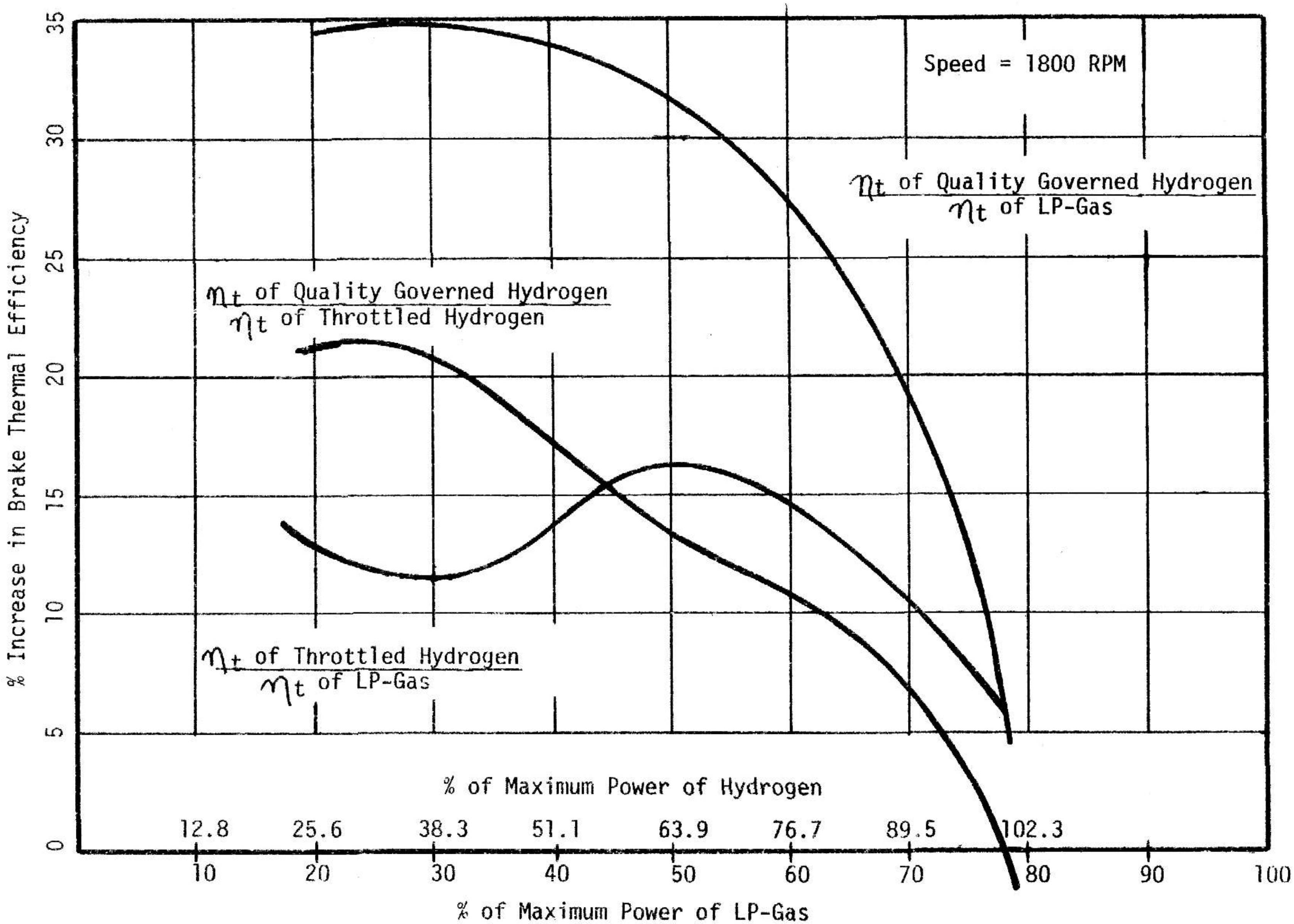


Figure 7: The Increase in Brake Thermal Efficiency of Hydrogen Compared to LP-Gas Operation at Various Power Levels

Maximum Power at Various Speeds

A comparison of maximum power available at various speeds for hydrogen and LP-gas reveals several deficiencies of hydrogen combustion in an internal combustion engine (Figure 8). One should notice that the hydrogen to air mixture dropped below stoichiometric conditions at the higher speeds (Table 9). Since the fuel-air mixtures were slightly leaner than the point at which maximum power occurs, some errors will be noted. Assuming that the effect of equivalence ratio variations on performance for all speeds experiencing the error is similar to the measured values at 1800 R.P.M., one would expect the maximum error for brake thermal efficiency and power to be of the relative magnitude of 3% high and 1% low respectively.

A water induction rate of four kilograms of water per kilogram of hydrogen provided adequate protection against backfiring for maximum power operation at speeds of 1800 R.P.M. and greater. Backfiring was a nuisance at 1600 R.P.M. until water induction was increased to six kilograms of water per kilogram of hydrogen. Operation at speeds less than 1600 R.P.M. at maximum power were impossible at any reasonable rate of water induction.

The maximum power available from hydrogen is approximately 85% of the maximum power generated on LP-gas. Maximum power for hydrogen occured at 2600 R.P.M. while LP-gas operation experienced maximum power at a speed of 2200 R.P.M. The collected data indicates that the gap between maximum power of hydrogen and LP-gas is much larger at slower speeds but narrows for higher speed operation.

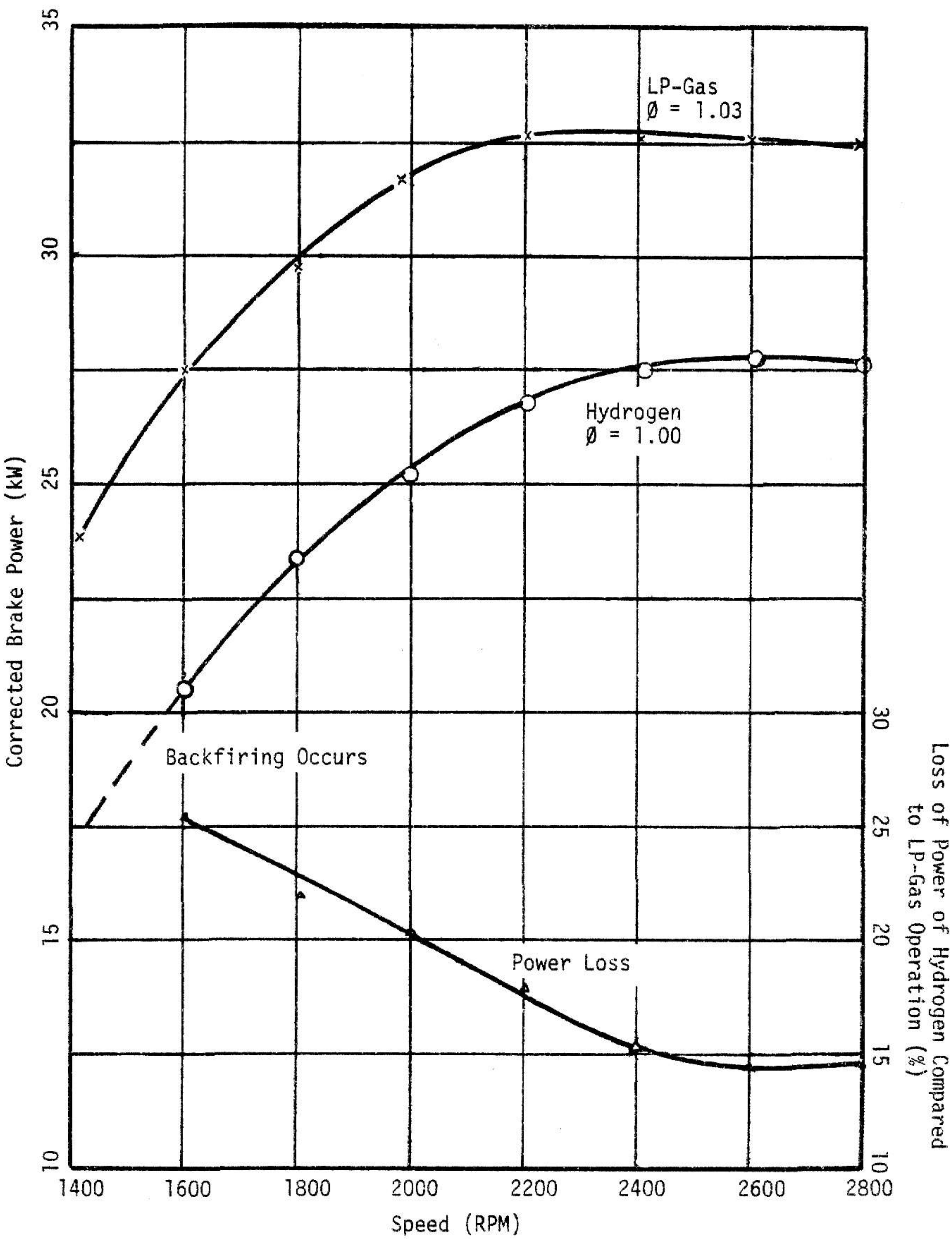


Figure 8: Maximum Power at Various Speeds for LP-Gas and Hydrogen

Torque observations for hydrogen (Figure 9) reveal little change in torque for speeds of 1600 to 2000 R.P.M. At speeds between 2400 and 2800 R.P.M., torque declines at a rate roughly equal to the increase in speed. Thus, very little change in power is observed in this range of speeds. A comparison of brake torque produced by hydrogen and LP-gas combustion reveals that the advantage of LP-gas operation dwindles at higher speeds. There appears to be some advantage in operating a hydrogen engine at a higher speed than LP-gas.

Brake thermal efficiency at maximum power of hydrogen and LP-gas reveals that LP-gas has the advantage. The friction horsepower which is constant for both fuels represents a small proportion of the total indicated horsepower generated by the engine when on LP-gas. This factor makes a higher thermal efficiency possible for the fuel that is able to produce the greatest power if all other factors affecting efficiency are fairly constant. One might also note that the gap between efficiency of hydrogen and LP-gas operation narrows at higher speeds. Due to the leaning of the hydrogen-air mixture that occured at higher speeds, this occurence may not be as great as indicated. As stated earlier, values of brake thermal efficiency for hydrogen operation between speeds of 2200 and 2800 R.P.M. should be reduced by a relative factor of two to three percent.

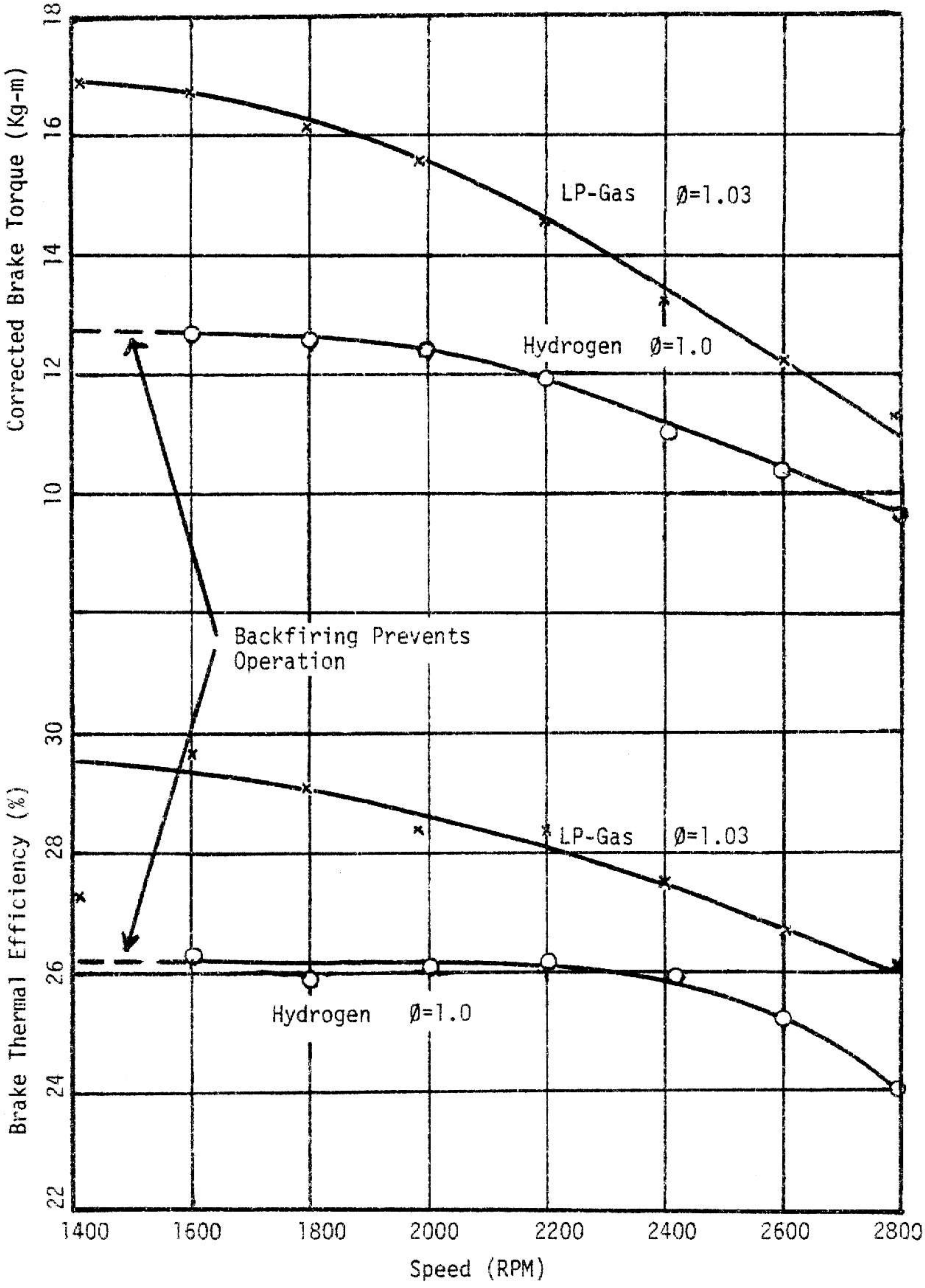


Figure 9: Torque and Efficiency vs. Speed at Maximum Power

Temperature Studies

A series of temperature studies of the center electrode of the spark plug, cylinder head, and exhaust gas were made for various water induction rates and for various power levels at a constant speed. An attempt to measure the exhaust valve face temperature proved to be unsuccessful.

The rate of water induction had a rather limited effect upon temperatures in the cylinder (Figure 10). The thermocouples in the spark plug and the wall of the cylinder head indicates that almost no change in temperatures occured at these poinst when the power level was not affected by increasing rates of water induction. The temperature in the central electrode insulator of the spark plug declined only at rates of water induction which cause a decline in power level. There was a rather noticeable drop in the temperature of the exhaust gas which amounted to about 30°C at a rate of water induction of four kilograms of water per kilogram of hydrogen as compared to when no water was added. This decline also suggests that some reduction in the temperature of the face of the exhaust valve may occur because of the exhaust gas influence upon the temperature of an exhaust valve (King, 1955).

The effect of fuel richness and power upon the measured temperatures at 1800 R.P.M. with water induction can be seen in Figure 11. The temperature of the spark plug approached the self ignition temperature of 585°C only at an equivalence ratio greater than one. However, this temperature does not insure that ignition will occur. Other factors such as mixture density, time lag, and composition

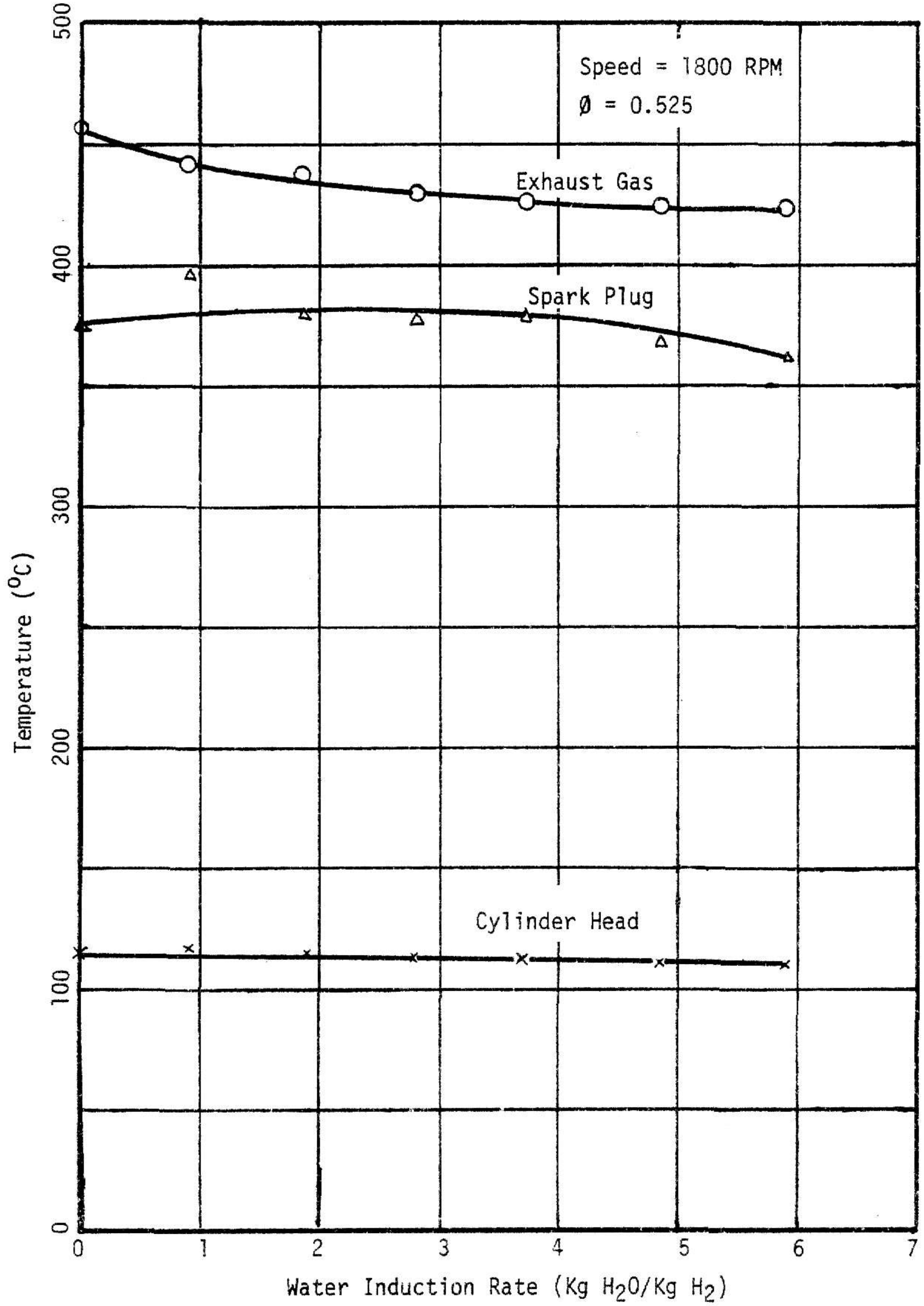


Figure 10: Variation in Cylinder Temperatures due to Water Induction

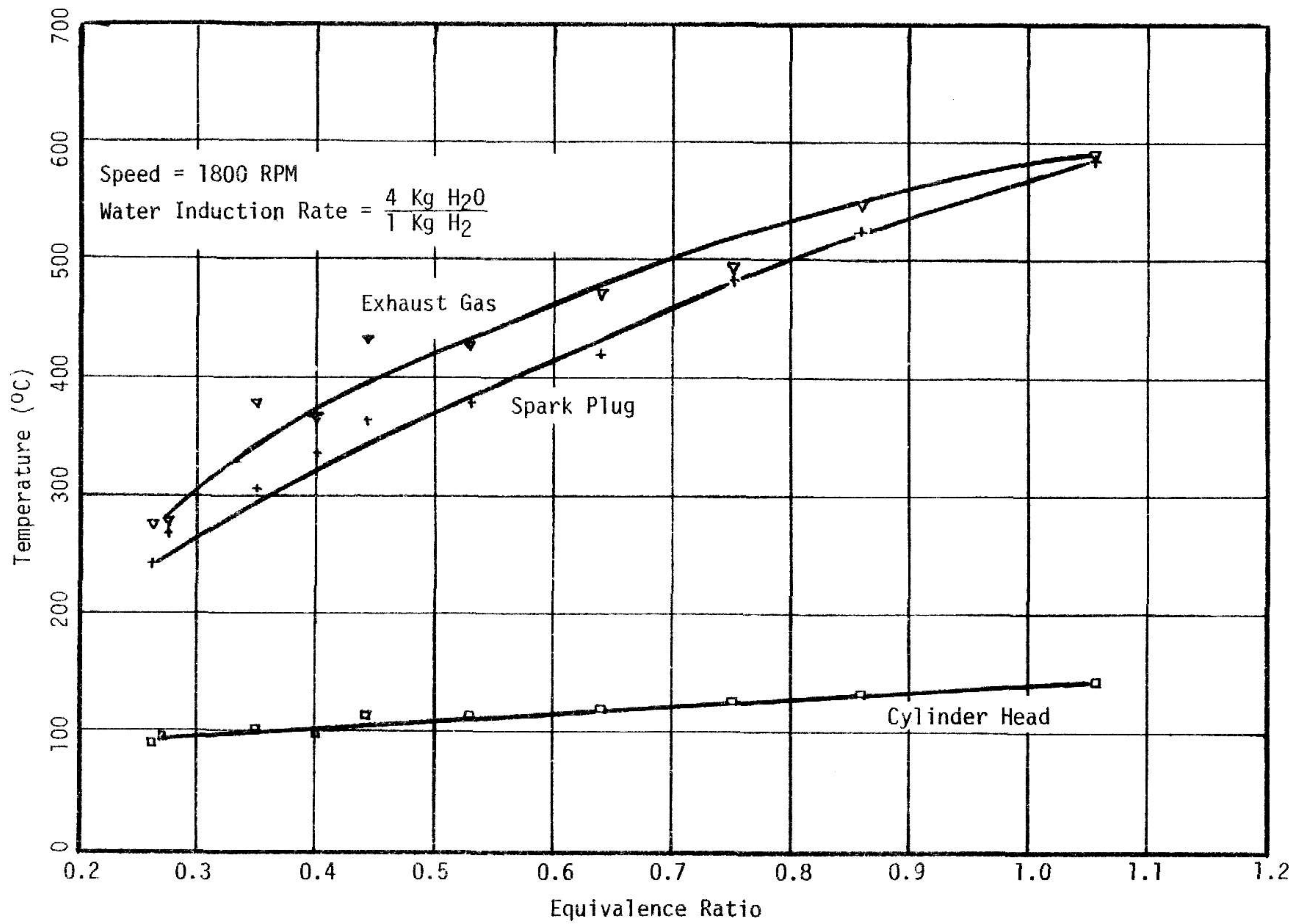


Figure 11: Temperature of Various Points in the Cylinder vs. Equivalence Ratio

of the fresh charge affect the occurence of self ignition. King and Rand (1955) did not experience backfiring caused by preignition until a temperature of 621^{o}C was attained for operation at 1800 R.P.M. The important fact that should be noted is that the temperature of the spark plug does not approach temperatures that could result in backfiring initiated by preignition until stoichiometric conditions are attained.

Backfiring which becomes a problem at Ø=0.5 without water induction could not be initiated by a hot spark plug electrode at this fuel-air mixture. Of course, this test was conducted with water induction but this addition causes little decline in temperature of the spark plug electrode as noted previously.

Another potential hot spot, the exhaust valve, does not appear to be a source of preignition at Ø=0.5. Our efforts to measure this point proved unsuccessful. However, it may be possible to correlate the temperature of the valve to that of the exhaust gas, which was measured. King (1955) reported that a standard uncooled exhaust valve experiences temperatures about 40^{o}C higher than the exhaust gas. The face of the sodium coolec valve in an air cooled engine also was reported to operate under a wide variety of conditions at temperatures always less than that of the exhaust gas (Sanders, 1943). The position of the exhaust gas thermocouple was similar in all cases. However, engine test conditions and type of engine varied between our testing and that of the reported authors. Also the design of the sodium cooled valve used by this author differed slightly from that used by Sanders.

It is reasonable to assume that the exhaust valve used in this study was operating at temperatures less than the exhaust gas. At the very worst one would not expect the valve's temperature to exceed the exhaust gas temperature by more than that reported by King for a standard uncooled exhaust valve. Based upon these assumptions, the temperature of the sodium cooled exhaust valve is far below the level which could cause preignition at $\emptyset$=0.5. Even after an allowance for the cooling of the exhaust gas and the exhaust valve due to water induction, the temperature of the exhaust valve is far from causing preignition at lean mixtures where backfiring is first noted.

The cylinder head temperature, as indicated by Figure 11 remains far below a level that might cause preignition of the fuel-air mixture. In fact, it is difficult to visualize that a casting projection on the wall would ever reach temperatures that could cause self ignition. There appears to be little need for smoothing down the rough casting of the head.

From this data and the assumptions, it appears that the spark plug, exhaust valve, and cylinder head are not the source of backfiring problems that are experienced for lean mixtures. Only as fuel-air mixutres near stoichiometric conditions, the spark plug or sodium cooled exhaust valve approach temperatures that could lead to preignition and backfiring. Therefore, some other source of the backfiring must exist. Since this backfiring occurred after thorough cleaning of cylinder, oil deposits in the cylinder were probably not the problem source. It appears that residual gasses or particles in the residual gases, as speculated by King (1948) and Lynch (1975), remain as possible sources of our backfiring problems at lean mixtures. The available information will not allow the author to speculate between these two sources.

A comparison of temperatures for LP-gas and quality governed hydrogen operation is displayed in Figure 12. The temperatures experienced during part load hydrogen operation are lower than those for LP-gas. As hydrogen fuel-air mixtures approach stoichiometric conditions, the measured temperatures resulting from hydrogen combustion generally approach that of LP-gas operation. This is due to much cooler combustion flame temperatures generally associated with lean operation of hydrogen. The exhaust gas temperature of hydrogen and LP-gas combustion vary greatly at lean hydrogen mixtures, but are very close when both are operating at stoichiometric conditions and similar power levels. The difference in spark plug temperatures narrows for the two fuels, but a $60^{o}C$ difference still exists at maximum power for hydrogen. This is due primarily to the different spark plug heat ranges used for the two fuels.

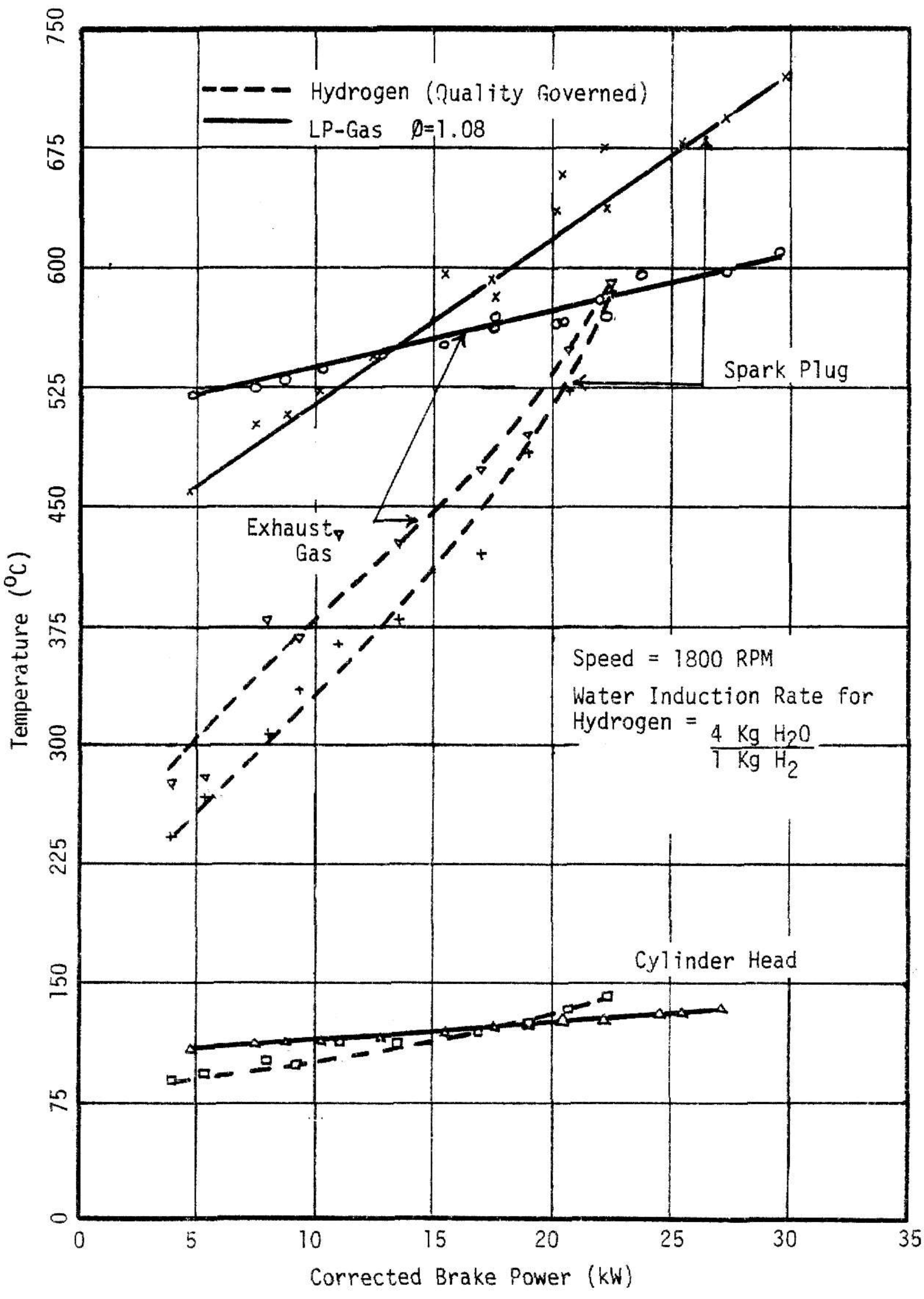
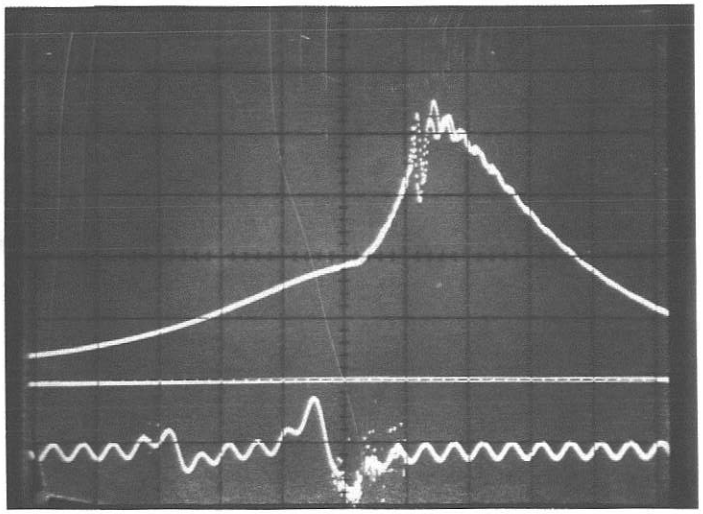


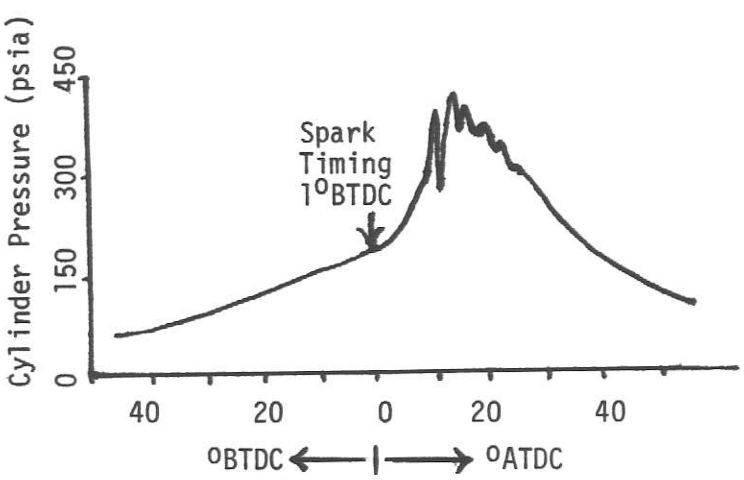
Figure 12: Temperature of Various Points in the Cylinder vs. Power Level

Cylinder Pressure Characteristics

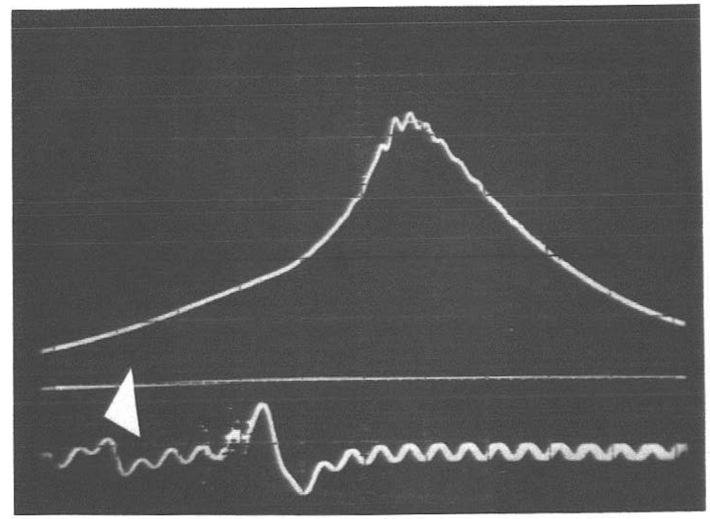
Observations of pressure rise of hydrogen and LP-gas also reveals some interesting differences (Figure 13). Pressure diagrams of fuel-air mixtures near stoichiometric conditions are characterized by a very rapid pressure rise with a number of sharp erratic pulses occuring near the peak pressure. With water induction, these erratic pressure pulses become fairly evident at about 15 kilowatts ($\emptyset$=0.5) for operation at 1800 R.P.M. Audible knock was not noted along with the occurrence of the pressure pulses. Studies by the University of Florida indicate that this inaudible "knocking" condition is caused by rapid combustion which results in a rapid pressure rise. This would be similar to knocking experienced in a compression ignition engine (Escher, 1975). Figure 13 displays the rapid pressure rise of hydrogen as compared to the smoother pressure carve of LP-gas. The rapid pressure rise experienced near top dead center also giver hydrogen combustion a much closer resemblence to the ideal Otto cycle.

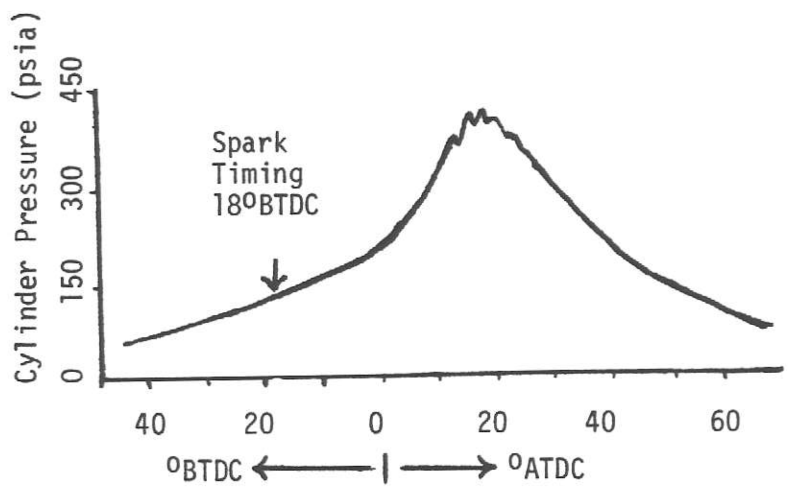
Water injection appears to provide some slowing or smoothing out of the initial pressure rise during lean operation (Figure 14). Water induction creates a longer period of ignition delay under the lean fuel-air condition. This effect disappears at richer mixtures. The delay between spark timing and the beginning of a rapid pressure rise is approximately two or three crankshaft degrees for $\emptyset$=1.0 (Figure 13). No comparison of the effect of water induction upon the pressure rise for fuel-air mixtures approaching stoichiometric conditions was made.





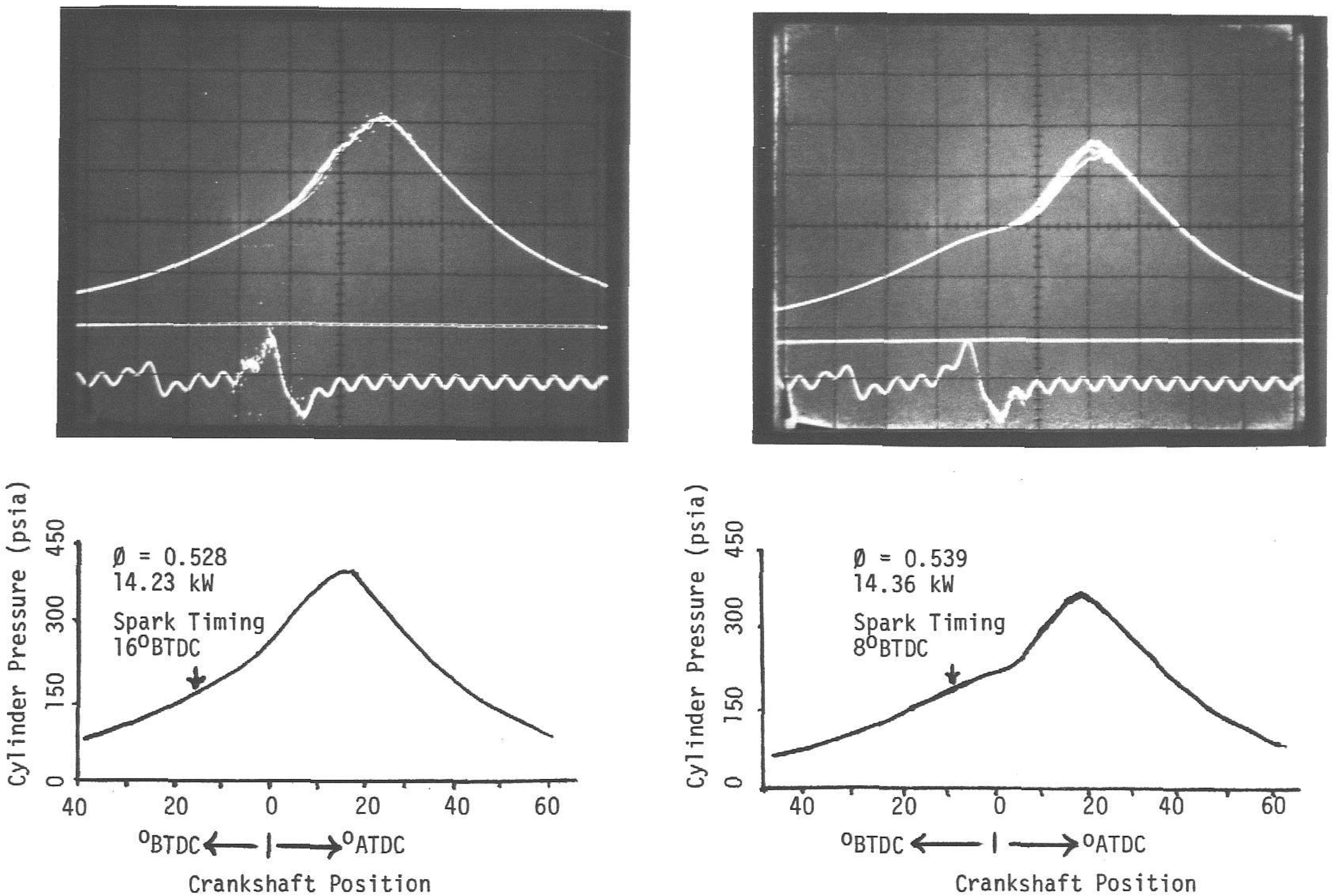
Hydrogen operation at 1805 R.P.M. and 22.56 kW.





LP-Gas operation at 1797 R.P.M. and 22.37 kW.

Figure 13: Comparison of the Cylinder Pressure Diagrams of LP-Gas and Hydrogen Combustion



Water Induction = 3.71 Kg. of water per Kg. of hydrogen.

No water induction.

Figure 14: The Effect of Water Induction on Cylinder Pressure

Application of Hydrogen to Agricultural Engines

The application of hydrogen to agricultural engines was a major concern of a number of these studies. The use of hydrogen in a naturally aspirated tractor engine with water induction is going to cause a number of differences in operation. A 15% reduction in maximum power can be expected in the switch from LP-gas to hydrogen. Since most tractor engines are often loaded to more than 75% of their maximum power during continuous operation, the conversion to hydrogen will restrict performance of many tractor operations previously possible with the same engine. In other words, the same size engine fueled by hydrogen will be restricted to smaller tillage tools and jobs as compared to operation on LP-gas or gasoline.

This particular engine when used in a tractor was rated for operation at 2200 R.P.M., the point of maximum power. Hydrogen experiences it's maximum power at 2600 R.P.M. and will probably need to be operated at this speed in order to achieve it's potential.

Another important aspect of tractor operations is it's ability to pull through a momentary overload without shifting gears to a lower speed. As a tractor encounters a rough spot, hopefully, the pulling ability or torque will increase as speed drops off (Table 9). This characteristic is represented by the effect of speed at maximum power on torque. Hydrogen operation at 2200 R.P.M. experiences only small increases in torque as speed drops off. The ability of a hydrogen fueled engine to "lug" through a momentary overload is minimal. If hydrogen operation was rated for a higher speed, the engine's ability to meet momentary overloads would improve slightly. However, this adjustment would be at the expense of fuel efficiency. During these tests no checks were made to see if hydrogen operation with water induction could react to sudden changes in load without experiencing backfiring. This could be a severe problem that needs to be tested.

Hydrogen operation in tractors appears to be limited due to power losses and its poor ability to meet momentary overloads. Of course, the naturally aspirated hydrogen fueled engine studied in this thesis is also severly limited in application to tractor engines due to the trend away from naturally aspirated spark ignition engines.

Application of hydrogen to irrigation engines appears to be brighter. Presently, spark ignition natural gas, gasoline, and LP-gas engines supply a significant portion of the power for irrigation. These engines are normally designed for continuous operation at 75 to 80% of their maximum power, although this may vary considerably for individual situations. The load will remain nearly constant under most conditions. An application such as irrigation should be satisfactory for hydrogen operation. Our tests reveal hydrogen can achieve the power necessary to supply the designed power requirements for LP-gas. This should also be true for natural gas which has a lower power to displacement ratio than LP-gas. An engine switched from gasoline to hydrogen will lose about 25% of it's power, possibly causing some loss in pumping rate. Of course, with any switch from hydrocarbon fuel to hydrogen, much less, if any, reserve power will be available.

The constant load operation will allow continuous operation under conditions where backfiring will be no problem. A readily accessible water supply will make water induction a practical modification. The one major remaining question concerning application of hydrogen to an irrigation engine is its effect upon engine lifetime. The inaudible knock or pressure pulses noted for a fairly wide range of power levels may create problems with certain engine components. Further consideration should be given to this potential problem.

CONCLUSIONS

1. Quality governing of a hydrogen engine provides an adequate means of controlling power except when the engine is idling.
2. Backfiring becomes a problem for fuel-air mixtures greater than 50% of stoichiometric conditions. Mixing of water with the fuel-air charge at a rate of four kilograms of water per kilogram of hydrogen provides successful control of backfiring except when the engine is placed under heavy loads at low speeds. Rates of water induction much greater than four kilograms of water per kilogram of hydrogen cause detrimental effects upon engine power and fuel efficiency.
3. Hydrogen operation experiences higher brake thermal efficiency than LP-gas for similar power levels. The relative magnitude of this advantage varies from 35% for lean hydrogen-air mixtures to 5% at maximum power levels of hydrogen.
4. In similar naturally aspirated engines, hydrogen will supply only 85% as much power as LP-gas. Hydrogen will produce its maximum power at approximately 400 R.P.M. faster than LP-gas.
5. Preignition at the exhaust valve or spark plug electrode, the two hottest spots in the combustion chamber, does not appear to be a problem until fuel-air mixture near stoichiometric conditions are attained. Backfiring at lean fuel-air mixtures does not originate from these two spots.
6. Hydrogen combustion experiences a very rapid and erratic pressure rise for fuel-air mixtures approaching $\phi=1.0$. The erratic pressure pulse experienced may be the result of extremely rapid combustion

and resulting pressure rise.

7. Hydrogen combustion in an engine with water induction to control backfiring appears to be most applicable for irrigation power units. Application of hydrogen to a tractor engine appears to be less acceptible. The loss of power and the poor ability to lug during overload of hydrogen engine will hinder its acceptance for use in a tractor.

SUGGESTIONS FOR FUTURE RESEARCH

The use of hydrogen in an internal combustion engine has three major areas which need further exploration. Further evaluation of hydrogen injection directly into the cylinder and the effect of the rapid and erratic pressure rise resulting from hydrogen combustion is needed. Field demonstration of agricultural applications of hydrogen are also needed.

Hydrogen injection has the potential to eliminate power loss problems and backfiring. Injection appears to be the only possible means of achieving compression ignition of either hydrogen-diesel mixtures or hydrogen. Design of equipment that will meet the needs of injecting a low density gas at the proper time needs further development. Such factors as energy requirements and economic cost of an injection system must be important considerations.

Further studies need to be made of the irregular pressure pulses experienced during the combustion pressure rise. An evaluation of this phonomenon's effect upon the life of various engine parts should be made. If this occurrence needs to be suppressed, such factors as spark timing and intake charge dilution need further consideration.

The best evaluation of hydrogen's application to an agricultural engine would be an in field study. Conversion of a tractor or irrigation engine to hydrogen for actual field tests would illustrate problems and advantages of hydrogen that may be overlooked during laboratory studies. Of course, field demonstration of such an application would also greatly enhance the general public's acceptance of hydrogen as an energy source.

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APPENDIX A

ENGINE SPECIFICATIONS

Engine model:	Ford Industrial 172 GF	
Displacement:	2.8 liters	
Number of cylinders:	Four in line	
Firing order:	1-2-4-3	
Compression ratio:	8.6 to 1	
Valve timing:	Intake opens	15^{o} BTDC
	Intake closes	35^{o} ABDC
	Exhaust opens	41^{o} BBDC
	Exhaust closes	15^{o} ATDC

Spark plug gap: 0.0625 cm for LP-gas operation
0.0381 cm for hydrogen operation

Spark timing: Minimum spark timing for maximum brake torque.

APPENDIX B

INFORMATION FOR FIGURES

Table 3: The Effect of the Variation of Water Induction Rate on Hydrogen Engine Performance

Speed (RPM)	Corrected Brake Torque (Kg-m)	Corrected Brake Power (kW)	Water Induction Rate $(\frac{Kg\ H_2O}{Kg\ H_2})$	Brake Thermal Efficiency (%)	Equivalence Ratio	Spark Timing (^{o}BTDC)
1799	8.36	15.45	0.	27.5	0.476	11
1804	8.46	15.67	0.28	28.1		11
1800	8.46	15.63	0.58	27.8	0.477	10
1800	8.51	15.73	1.28	28.3		10
1792	8.54	15.72	1.72	27.6	0.484	10
1792	8.55	15.74	2.46	28.2		11
1780	8.55	15.63	3.14	28.1	0.476	12
1775	8.48	15.46	3.76	27.2		14
1768	8.51	15.44	4.54	27.9	0.484	15
1765	8.46	15.33	5.27	27.7		17
1755	8.42	15.17	6.00	27.5	0.487	17

The information is displayed in Figure 3.

Data is Corrected to Standard SAE Conditions:

Atmospheric Pressure = 74.63 cm. of Hg

Dry Bulb Temperature = 29.4^{o}C

Wet Bulb Temperature = 17.5^{o}C

Power Control: Quality Governed

Table 4: The Effect of the Variation in Water Induction Rate on Cylinder Temperatures and Performance During Operation on Hydrogen

Speed (RPM)	Corrected Brake Torque (Kg-m)	Corrected Brake Power (kW)	Water Induction Rate $\left(\frac{Kg\ H_2O}{Kg\ H_2}\right)$	Brake Thermal Effi-ciency (%)	Equiv-alence Ratio	Spark Timing (^oBTDC)	Spark Plug Temper-ature (^oC)	Exhaust Temper-ature (^oC)	Cylinder Head Temper-ature (^oC)
1805	7.75	14.36	0.	24.8	0.539	8	376	457	116
1803	7.74	14.33	0.92	25.3	0.529	13	396	443	117
1800	7.74	14.30	1.86	25.5	0.524	13	381	438	115
1808	7.74	14.36	2.81	25.4	0.518	15	378	431	114
1801	7.69	14.23	3.71	25.0	0.528	16	379	427	113
1791	7.63	14.03	4.86	24.9	0.527	16	368	424	111
1781	7.62	13.94	5.88	25.0	0.524	16	362	424	110

Information is displayed in Figure 4 and Figure 10.

Data is Corrected to Standard SAE Conditions:

Atmospheric Pressure = 74.63 cm. of Hg

Dry Bulb Temperature = 29.4^oC

Wet Bulb Temperature = 17.5^oC

Power Control: Quality Governed

Table 5: The Effect of Fuel Richness on Hydrogen Performance at Wide Open Throttle

Speed (RPM)	Corrected Brake Torque (Kg-m)	Corrected Brake Power (kW)	Water Induction Rate $(\frac{Kg\ H_2O}{Kg\ H_2})$	Brake Thermal Efficiency (%)	Equivalence Ratio	Spark Timing (^{o}BTDC)
1813	3.84	7.15	0.	19.8	0.273	25
1795	4.47	8.25	0.	22.0	0.292	21
1801	5.26	9.72	0.	23.0	0.335	19
1795	6.16	11.36	0.	25.2	0.366	15
1802	7.00	12.96	0.	25.9	0.412	13
1799	7.92	14.63	0.	26.4	0.468	12
1800	8.42	15.56	0.	27.6	0.485	10
1800	8.85	16.36	0.	27.2	0.527	8
1795	9.45	17.42	0.	28.6	0.541	7
1795	9.85	18.15	0.	28.6	0.577	6
1797	10.03	18.52	0.	28.7	0.584	6
1810	3.36	6.24	3.83	18.5	0.279	32
1794	5.49	10.11	4.02	24.5	0.360	22
1798	7.09	13.09	3.96	25.8	0.462	19
1802	8.00	14.81	3.79	26.9	0.513	14
1799	10.05	18.57	4.13	28.2	0.634	11
1801	10.65	19.70	4.02	27.9	0.697	7
1802	11.01	20.38	4.11	28.0	0.738	6
1806	12.23	22.69	4.04	26.7	0.916	2
1795	12.71	23.42	3.89	25.4	1.041	1
1797	12.59	23.23	3.83	24.8	1.06	1
1790	12.54	23.06	3.74	24.0	1.10	1

The information is displayed in Figure 5.

Data is Corrected to Standard SAE Conditions:

Atmospheric Pressure = 74.63 cm. of Hg

Dry Bulb Temperature = 29.4^{o}C

Wet Bulb Temperature = 17.5^{o}C

Power Control: Quality Governed

During tests on hydrogen with no water induction, operation was ended due to backfiring at $\emptyset > 0.584$. Occasional backfiring was noticed first at $\emptyset = 0.527$.

Table 6: Performance of Engine Operated on LP-Gas at Constant Speed

Speed (RPM)	Brake Torque (Kg-m)	Brake Power (kW)	Brake Thermal Efficiency (%)	Equivalence Ratio
1800	2.71	5.00	12.8	1.03
1801	5.33	9.87	17.4	1.12
1796	7.50	13.83	20.3	1.15
1795	8.77	16.16	21.4	1.16
1797	10.48	19.35	23.3	1.14
1798	12.47	23.02	24.6	1.13
1791	13.85	25.47	26.1	1.11
1791	15.82	29.09	28.6	1.05
1801	15.90	29.41	28.7	1.06

The information is displayed in Figure 6 and Figure 7.

Conditions during data acquisition:

Atmospheric Pressure = 72.39 cm. of Hg

Dry Bulb Temperature = 25.6°C

Wet Bulb Temperature = 11.7°C

Power Control: Throttle

Table 7: Performance of Throttled Engine Operation on Hydrogen at Constant Speed

Speed (RPM)	Brake Torque (Kg-m)	Brake Power (kW)	Water Induction Rate $\left(\frac{Kg\ H_2O}{Kg\ H_2}\right)$	Brake Thermal Effi-ciency (%)	Equiv-alence Ratio	Spark Timing (^oBTDC)
1820	4.49	8.40	3.32	18.6	0.889	2
1804	6.50	12.04	3.53	22.1	0.930	1
1802	7.74	14.33	3.50	23.6	0.947	1
1798	9.02	16.66	3.58	25.2	0.943	0
1812	10.46	19.47	3.66	25.7	1.01	1^oATDC
1806	11.72	21.73	3.81	25.8	1.02	1^oATDC
1810	2.64	4.90	3.14	12.7	1.01	0
1815	5.33	9.94	3.35	18.9	1.01	0
1802	12.50	23.13	3.75	25.8	1.01	0

The information is displayed in Figure 6 and Figure 7.

Conditions during data acquisition:

Atmospheric Pressure = 73.46 cm. of Hg

Dry Bulb Temperature = 25^oC

Wet Bulb Temperature = 12.2^oC

Power Control: Throttle

Table 8: Performance of Quality Governed Engine Operated on Hydrogen at Constant Speed

Speed (RPM)	Corrected Brake Torque (Kg-m)	Corrected Brake Power (kW)	Water Induction Rate $(\frac{Kg\ H_2O}{Kg\ H_2})$	Brake Thermal Efficiency (%)	Equivalence Ratio	Spark Timing (^{o}BTDC)
1813	3.74	6.97	0.	19.8	0.273	25
1795	4.37	8.05	0.	22.0	0.292	21
1801	5.14	9.50	0.	23.0	0.335	19
1795	6.03	11.11	0.	25.2	0.366	15
1802	6.86	12.69	0.	25.9	0.412	13
1799	7.75	14.33	0.	26.4	0.468	12
1800	8.25	15.25	0.	27.6	0.485	10
1800	8.68	16.04	0.	27.2	0.527	8
1795	9.26	17.08	0.	28.6	0.541	7
1795	9.65	17.80	0.	28.6	0.577	6
1797	9.84	18.16	0.	28.7	0.584	6
1810	3.27	6.07	3.83	18.5	0.279	32
1794	5.36	9.88	4.02	24.5	0.360	22
1798	6.94	12.81	3.96	25.8	0.462	19
1802	7.83	14.50	3.79	26.9	0.513	14
1799	9.85	18.20	4.13	28.2	0.634	11
1801	10.44	19.31	4.02	27.9	0.697	7
1802	10.79	19.98	4.11	28.0	0.738	6
1806	12.00	22.26	4.04	26.7	0.916	2
1795	12.46	22.97	3.89	25.4	1.041	1
1797	12.35	22.78	3.83	24.8	1.06	1
1790	12.30	22.62	3.74	24.0	1.10	1

This information is displayed in Figure 6 and Figure 7.

Data is Corrected to:

Atmospheric Pressure = 72.39 cm. of Hg

Dry Bulb Temperature = 25.6^{o}C

Wet Bulb Temperature = 11.7^{o}C

Power Control: Quality Governed

During tests on hydrogen with no water induction, operation was ended due to backfiring at $\emptyset > 0.584$. Occasional backfiring was noticed first at $\emptyset = 0.527$.

Table 9: Engine Performance at Maximum Power at Various Speeds

Speed (RPM)	Corrected Brake Torque (Kg-m)	Corrected Brake Power (kW)	Water Induction Rate $\left(\frac{Kg\ H_2O}{Kg\ H_2}\right)$	Brake Thermal Efficiency (%)	Equivalence Ratio	Spark Timing (^{o}BTDC)
Power Control: Throttle Fuel: LP-Gas						
1410	16.89	24.46	0.	27.3	1.031	10
1598	16.76	27.50	0.	29.7	1.015	12
1794	16.15	29.76	0.	29.1	1.027	14
1984	15.56	31.70	0.	28.4	1.032	15
2198	14.55	32.84	0.	28.4	1.034	16
2401	13.27	32.72	0.	27.5	1.033	17
2598	12.22	32.61	0.	26.7	1.037	19
2794	11.26	32.31	0.	26.0	1.033	20
Power Control: Quality Governed Fuel: Hydrogen						
1600	12.72	20.91	6.15	26.3	0.968	0
1798	12.59	23.25	3.89	25.9	1.009	0
1998	12.45	25.55	4.02	26.1	1.011	0
2199	11.94	26.97	4.10	26.2	0.976	0
2412	11.04	27.35	4.00	25.9	0.941	0
2597	10.40	27.75	3.94	25.2	0.944	0
2797	9.63	27.66	3.76	24.0	0.955	0

The information is displayed in Figure 8 and Figure 9.

Data is Corrected to Standard SAE Conditions:

Atmospheric Pressure = 74.63 cm. of Hg

Dry Bulb Temperature = 29.4^{o}C

Wet Bulb Temperature = 17.5^{o}C

Operation on hydrogen at speeds below 1600 RPM was impossible due to backfiring even at high water induction rates. At 1600 RPM, a higher rate of water induction was needed to control backfiring. Operation on propane at wide open throttle was limited by vibration problems in the dynamometer at speeds below 1400 RPM.

Table 10: LP-Gas Engine Performance and Temperatures at Various Power Levels for a Constant Speed

Speed (RPM)	Brake Torque (Kg-m)	Brake Power (kW)	Brake Thermal Efficiency (%)	Equivalence Ratio	Spark Plug Temperature (°C)	Exhaust Temperature (°C)	Cylinder Head Temperature (°C)
1808	15.97	29.65	25.8	1.10	719	608	135
1807	9.48	17.60	20.9	1.11	581	571	122
1797	12.12	22.37	23.0	1.12	635	583	127
1796	13.80	25.46	23.8	1.12	678	595	131
1800	14.80	27.37	24.6	1.11	693	598	133
1805	11.03	20.44	22.4	1.10	659	567	127
1800	8.36	15.46	20.3	1.06	596	552	120
1802	6.81	12.61	18.5	1.05	548	546	117
1802	5.54	10.26	16.2	1.05	523	537	114
1799	4.80	8.88	17.8	1.00	509	533	113
1802	2.59	4.80	11.8	0.99	460	520	109
1796	4.03	7.44	14.0	1.02	503	526	112
1803	9.48	17.56	20.9	1.10	593	563	122
1800	10.97	20.27	21.9	1.12	636	568	126
1799	12.04	22.24	22.9	1.11	677	570	129

The information is displayed in Figure 4 and Figure 12.

Conditions during data acquisition:

Atmospheric Pressure = 72.90 cm. of Hg

Dry Bulb Temperature = 25.6°C

Wet Bulb Temperature = 18.9°C

Power Control: Throttle

Table 11: Hydrogen Engine Performance and Temperatures at Various Power Levels for a Constant Speed

Speed (RPM)	Brake Torque (Kg-m)	Brake Power (kW)	Water Induction Power $\left(\frac{Kg\ H_2O}{Kg\ H_2}\right)$	Brake Thermal Efficiency (%)	Equivalence Ratio	Spark Timing (°BTDC)	Spark Plug Temperature (°C)	Exhaust Temperature (°C)	Cylinder Head Temperature (°C)
1806	2.07	3.84	3.52	13.6	0.262	36	243	276	91
1799	4.31	7.96	3.72	21.2	0.349	19	306	379	102
1803	5.93	10.98	3.89	23.6	0.442	12	364	432	113
1803	9.17	16.97	3.73	26.3	0.640	8	419	473	121
1797	5.03	9.28	3.51	20.3	0.401	13	336	367	99
1805	2.88	5.33	3.79	16.1	0.273	35	268	279	94
1819	11.09	20.71	4.40	26.5	0.857	4	523	548	134
1805	12.17	22.56	4.58	25.4	1.056	1	585	589	143
1805	10.22	18.94	3.27	27.3	0.750	6	485	494	126
1801	7.39	13.66	3.71	25.0	0.528	12	379	427	113

The information is displayed in Figure 11 and Figure 12.

Data is corrected to:

Atmospheric Pressure = 72.90 cm. of Hg

Dry Bulb Temperature = 25.6°C

Wet Bulb Temperature = 18.9°C

Power Control: Quality Governed

THE APPLICATION OF HYDROGEN TO AN
AGRICULTURAL INTERNAL COMBUSTION ENGINE

by

RICHARD K. KOELSCH

B.S., Kansas State University, 1975

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MASTER OF SCIENCE

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KANSAS STATE UNIVERSITY
Manhattan, Kansas

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Hydrogen, an explosive gas feared by past generations, is being given growing consideration as a future replacement for some fossil fuels. The major concern of this thesis is the evaluation of it to one particular application, an agricultural internal combustion engine. Much consideration will be given to a hydrogen fueled engine's ability to meet the specialized requirements of present agricultural engines.

An LP-gas industrial engine was adapted for operation on hydrogen or LP-gas so that a comparative analysis of the two fuels could be made. Most of the alterations were an attempt to eliminate backfiring experienced with hydrogen. The easily ignited and highly explosive nature of hydrogen is the underlying cause of this annoying and possibly destructive phenomenon. Unless backfiring is eliminated the operation of a hydrogen engine is confined to lean fuel air mixtures and low power outputs.

One possible alteration to control backfiring involves the mixing of water with the fresh fuel air mixture. Water induction in conjunction with other changes that eliminate hot spots in the cylinder is an effective means of preventing backfire. The addition of an inert substance such as water increases the heat capacity of the fuel air mixture surrounding a dangerous heat source tending to retard or prevent any undesirable thermal reaction. A rate of four kilograms of water per kilogram of hydrogen proved to be effective in preventing backfire under most operating conditions except for heavy loads at low engine speeds. This rate also has very little if any detrimental effect upon engine power output or thermal efficiency.

Temperature studies of various points within the cylinder during hydrogen combustion provided some interesting insights into the backfire problem. The exhaust valve face and center electrode of the spark plug are considered prime sources of preignition that could lead to backfire. Replacement of these engine components with sodium filled exhaust valves and cooler operating spark plugs does not eliminate this nagging problem. However, temperature studies of these points do not indicate that they are the source of backfire at least during lean fuel-air operation. It appears that the exhaust gases or hot particles in the exhaust gas may be the source of this irritating problem.

Performance of a hydrogen engine in comparison to an engine fired by common hydrocarbon fuels is a critical concern. Quality governed hydrogen continuously demonstrated improved thermal efficiency over the same engine operated on LP-gas. However, available power from an engine fueled by hydrogen is less than the power available from LP-gas operation. Hydrogen's lower volumetric energy density than common hydrocarbon fuels accounts for most of this loss. The quantity of lost power is a major concern if present agricultural engines are converted to hydrogen. Hydrogen operation also revealed a poor torque rise at maximum power as speed is reduced. This factor may limit its application to an agricultural tractor which must be able to react to rapidly changing load conditions in the field.

Hydrogen's application to an agricultural tractor has several serious drawbacks. A farmer simply could not expect the same performance out of a similar sized engine fueled by hydrogen as more common fuels have demonstrated. However, the constant load of an irrigation engine may prove viable for hydrogen application. Presently, hydrogen's replacement of LP-gas or natural gas in irrigation engines provides the most encouraging potential use.