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A HYDROGEN ENGINE INDUCTION TECHNIQUE FOR BACKFIRE-FREE OPERATION

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16. Abstract (Limit: 200 words)

Details are given of a fuel-induction system for a hydrogen-fueled engine. The induction system was designed to preclude backfire otherwise common in hydrogen-fueled engines. System features and test data are presented and discussed in the context of the requirements peculiar to use in an underground mine environment. The fuel induction-system, when tested in conjunction with a modified Caterpillar 3304 engine, provided backfire-free engine operation and satisfactory engine performance.

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FORWORD

This report was prepared by Eimco Mining Machinery International under USBM contract H0202034, with Lars Olavson as Program Manager. The contract was initiated under the Metal and Nonmetal Health and Safety Program. It was administered under the technical direction of the Twin Cities Research Center, with Mr. Lito Mejia acting as the Technical Project Officer. Mrs. Sandra Schlesier was the Contract Administrator for the Bureau of Mines.

The report summarizes work completed under this contract from April 1981 through May 31, 1982.

ACKNOWLEDGEMENT

Appreciation is expressed to the Technical Project Officer of the U.S. Bureau of Mines, Mr. Lito Mejia for his cooperation and guidance and also for the direct participation on the part of Mr. Mejia in collecting data.

Appreciation is expressed also to Mr. Lloyd Johnson of Leejon Engineering for collaboration and assistance on engine design.

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INTRODUCTION AND SUMMARY

This study constitutes one phase in a long range program with the overall objective of producing a practical clean burning engine suitable for powering underground mining equipment.

The Program phases are:

I. Technical Assessment

II. Engine Development

III. Fuel System Design and Test

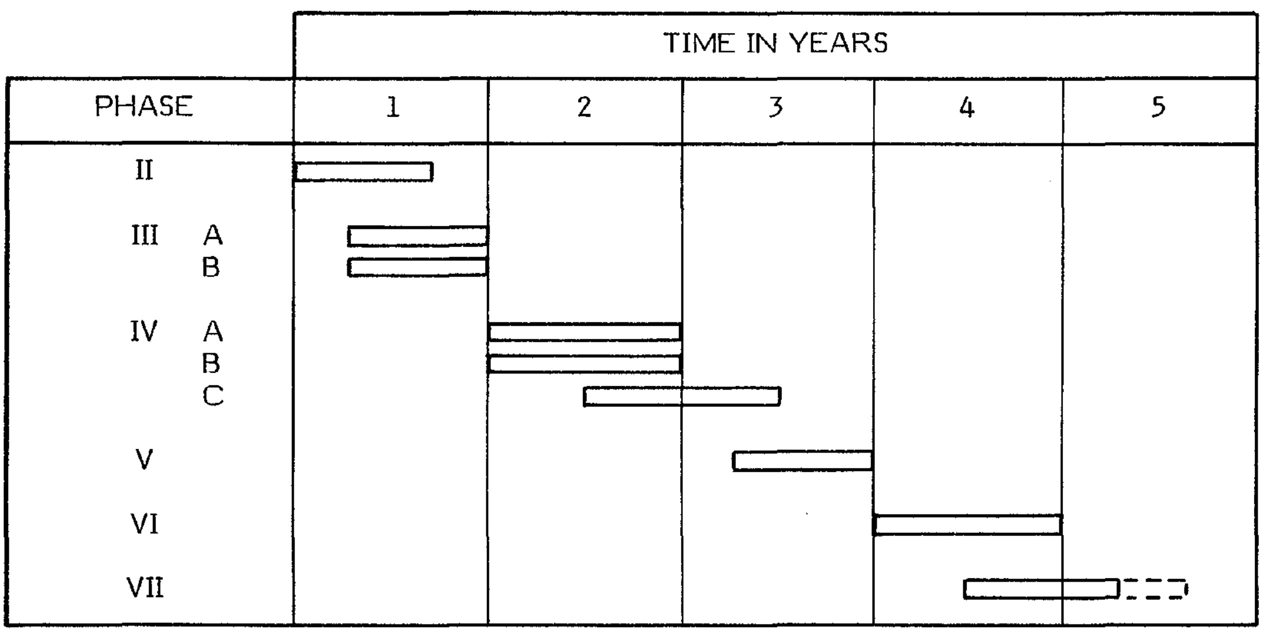
IV. Prototype Vehicle Development

V. Surface Demonstration

VI. Underground Demonstration

The time frame for this activity is shown in the bar chart below.

PROGRAM SCHEDULE



The objectives of Phase II of this project were to correct backfiring problems encountered earlier, when an engine was operated with premixed or "carbureted" hydrogen-air mixtures and to verify that the low emissions and excellent performance established earlier were not seriously compromised by the new design.

The testing was conducted at Hydride Energy Specialists* facilities in Denver, Colorado under a cost-sharing arrangement between the U.S. Bureau of Mines and Eimco Mining Machinery International.

The most significant result of the modifications reported herein is the elimination of intake system ignition or "back-firing." On several occasions during testing, the engine was pushed beyond its limits into conditions where preignition occurred. This would certainly have caused backfiring in a conventional intake carbureted system, however, at no time did the engine backfire with the new induction system.

The NO_x emissions were held to very low levels (lower than anticipated) by the use of very lean hydrogen air ratios with high boost pressure. Typical NO_x emissions were 0.3 g/kWh compared with 5-10 g/kWh emitted by the cleanest of mining diesels (see Figure 1).

Fuel consumption was lower than expected. Previous hydrogen consumption data indicated a minimum brake-specific fuel consumption of 95 g H2/kWh. The minimum consumption measured during testing of the new (Parallel Induction) fuel system was only 84 g/kWh. This is the equivalent of about 240 g/kWh of diesel fuel on a lower heating value basis.

The hydrogen engine produces torque levels about 20% greater than a naturally aspirated diesel at Denver's altitude of 1600 m.

BACKGROUND

The Eimco Hydrogen Mining Vehicle project began in 1977 with the objective of creating an ultra-low emission power system for underground mining machinery. A Caterpillar 3304 was dynamometer tested in its standard, pre-chamber diesel form to establish baseline information at Denver's altitude before conversion to hydrogen fuel. The engine was subsequently disassembled and modified for hydrogen testing. The 17.5:1 compression ratio of the diesel was reduced to 10.5:1 by substituting Caterpillar gas engine pistons. Spark plug adaptors, also from the natural gas version of the 3304, were installed in place of the diesel prechambers. Spark timing was provided by several experimental devices ranging from conventional breaker-point and coil systems to solid state breakerless timing circuits.(1)

During the early phases of testing, the hydrogen-converted engine was operated with naturally aspirated hydrogen-air mixtures formed in a propane-type gas mixer or "carburetor". Later a turbocharger and finally an aftercooler were added to the system. The naturally aspirated hydrogen engine could produce about 60% of the rated load of the naturally aspirated diesel without exceeding the NO_x emissions goal of the project (0.67 g/kWh). The turbocharged hydrogen engine gave virtually the same performance as the diesel, while the turbocharged-aftercooled engine outperformed the naturally aspirated diesel without exceeding the emissions goal (1). A metal hydride fuel storage container was coupled to the test engine and many hours of testing were conducted to gain information about the performance of the system (2).

*Hydride Energy Specialists (HES) is a joint venture between Ergenics, and Hydrogen Consultants Inc.

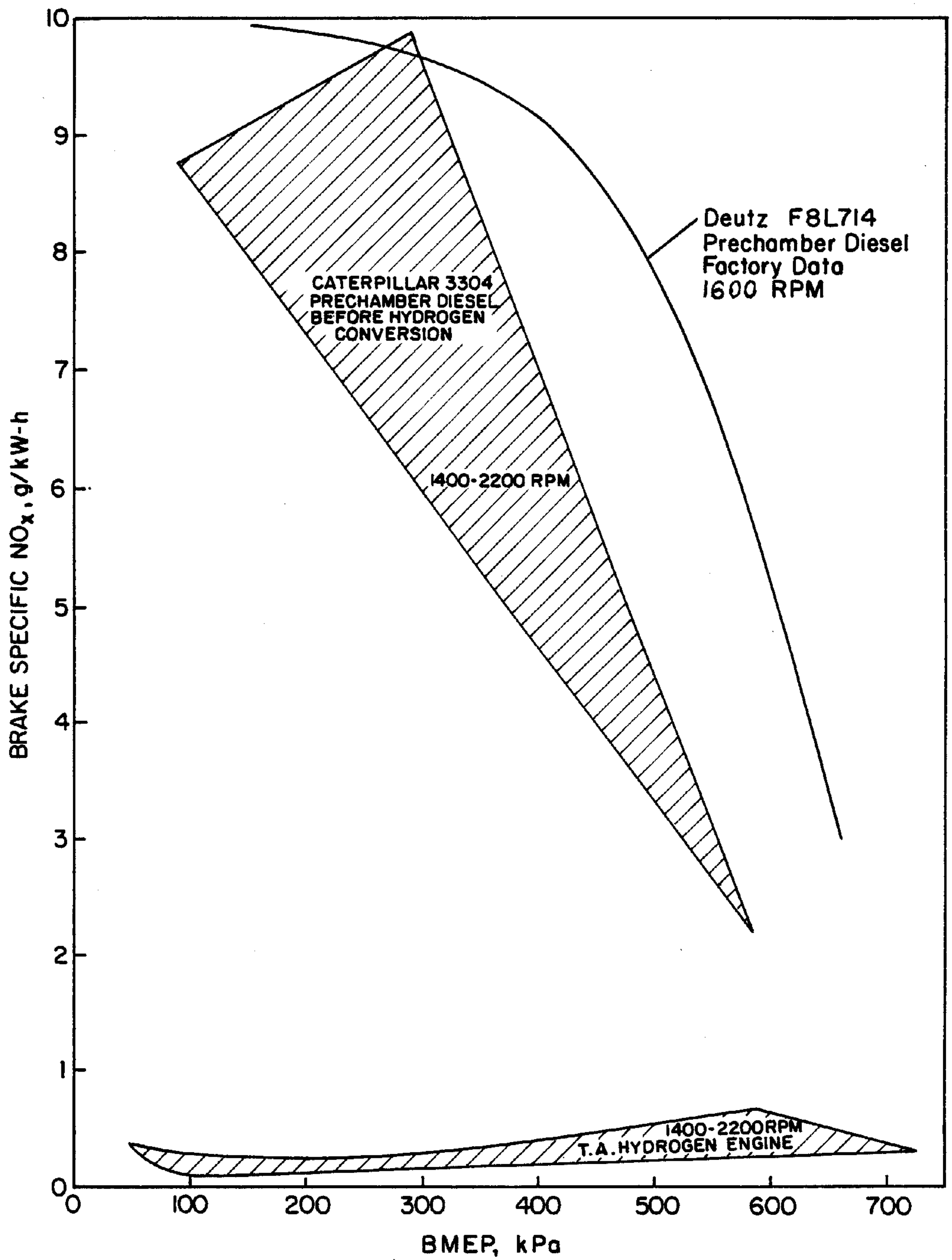


Figure 1.
Brake specific oxides of nitrogen vs brake mean effective pressure
for the hydrogen fueled Cat 3304 compared to two prechamber diesel engines.

Having proven the basic feasibility, performance and emissions potential of the power system, U.S. Bureau of Mines participation was sought to aid in formulating other criteria for the use of hydrogen technology in underground mining.

At this stage of development the engine was equipped for laboratory testing. There are many differences between a test engine with manual controls on each operating variable and a practical engine for use in a vehicle. In the latter case the only human input is the adjustment of engine torque via the accelerator pedal. Also the test engine, like all premixed hydrogen engines, had a propensity to backfire severely as a result of a bewildering number of causes. Many, if not all of these causes, could and would eventually arise if the engine were used in underground mining where these startling "bangs" would be unacceptable. A cost-shared effort by U.S. Bureau of Mines and Eimco was initiated in 1980 to assess solutions for the backfiring problem and other problems which had to be solved before undergound use could seriously be considered. A very broad survey was conducted of the various technical and economic aspects of hydrogen energy in the context of underground mining.(3) For the purpose of the present report a significant finding of Reference 3 concerns fuel system safety.

As a result, safety heavily influenced the selection of the engine control technique to be used in future development and demonstration activities. The maximum accidental fuel release of a metal hydride fuel storage container is determined to a great extent by the difference between the hydrogen pressure inside it and the atmosphere's pressure around it. The closer this difference is to zero, the less fuel can escape in the event of an accidental rupture of the container. However, some form of pressurized fuel injection was deemed necessary to solve the backfiring problem. Among the methods described in the literature for hydrogen injection, none have lower fuel pressure requirements than Parallel Induction (3).

PARALLEL INDUCTION

The hydrogen induction technique used on the Caterpiallar 3304 is called "Parallel Induction". Parallel Induction means that hydrogen and air enter the engine through separate, parallel paths. Each of these parallel systems has its own throttle, its own manifold, and its own valve system. The information provided on the following pages will show that:

- hydrogen fuel may be delivered to each cylinder in accurately metered pulses by simple, reliable, mechanical methods.
- these methods eliminate engine backfiring without the use of complex manual or computer controls like many research engines reported in the literature.
- there are no unproven components in the system. Poppet valves, throttles, pressure regulators and turbochargers have service lifetimes comarable to internal combustion engines and do not impose unusual service problems for the existing mine maintenance facilities.
- the principles of Parallel Induction are simple enough that no special training of operation or maintenance personnel is necessary.
- fuel energy consumption is comparable to diesel engines.

- exhaust emissions are negligible without controls other than maintaining a very fuel-lean mixture.

- fuel pressure requirements are very low.

The basic objectives of a Parallel Induction mixture control system are to separate and control the flow of fuel and air into the engine cylinders so that intake manifold explosions (backfiring) cannot occur. Backfiring in engines with premixed (carbureted) hydrogen-air mixtures has been a major obstacle to the practicality of hydrogen engines. Parallel Induction is conceptually similar to intake port injection in that fuel and air enter the cylinder during the intake stroke but are not premixed in the manifold. Figure 2 contrasts port injection with premixing or carburetion.

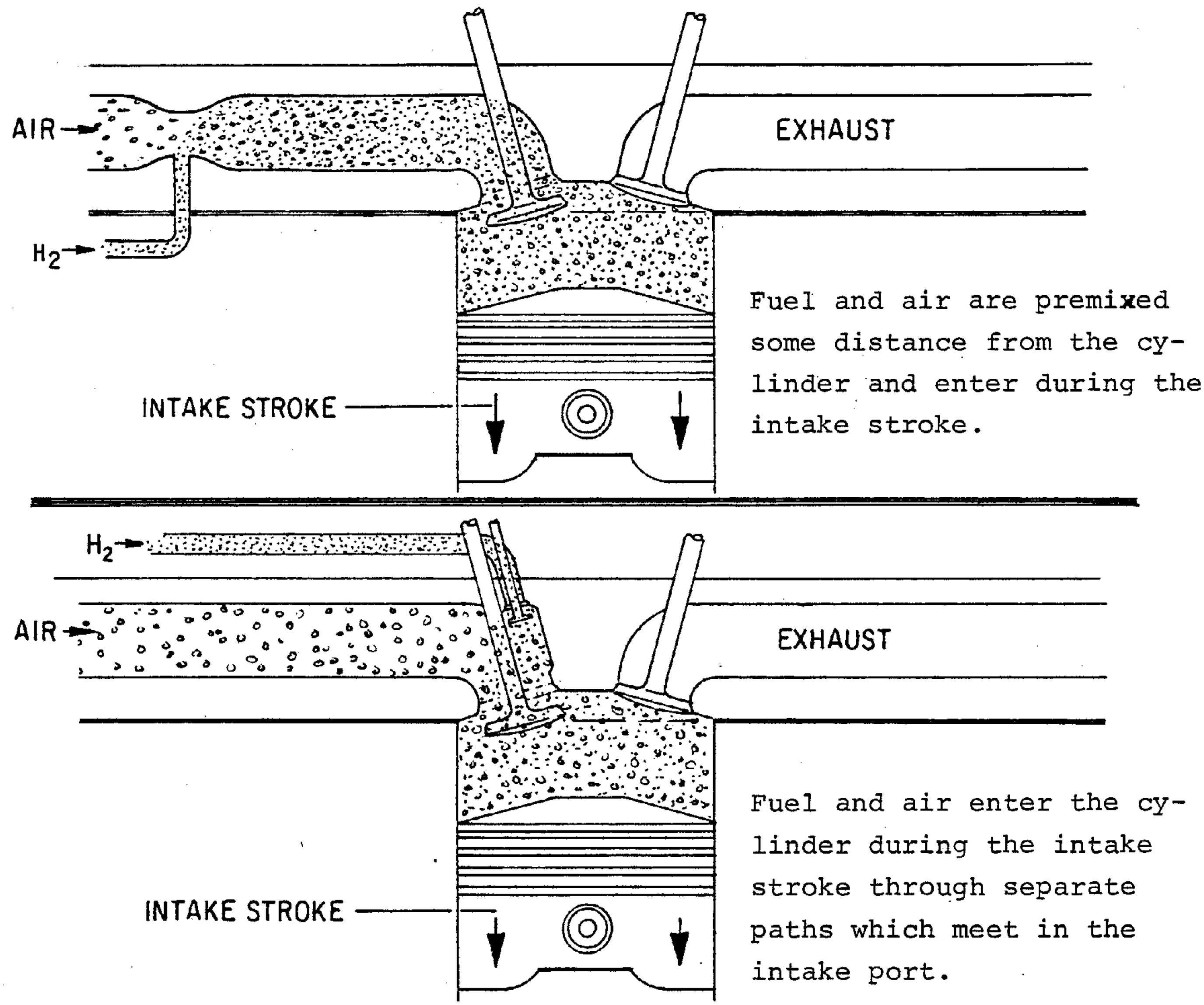


Figure 2.
Comparison between premixed or "carbureted" hydrogen engines and port-injected designs like the Parallel Induction system.

Intake stroke ignition or "backfiring" in "carbureted" engines results in startling "bangs", periodic engine stalls and, in some cases, engine damage. If the mixture in the cylinder of a Parallel Induction hydrogen engine is ignited during the intake stroke, the explosion cannot propagate through the intake manifold since it contains only air. The engine operator perceives such an incident as a misfire. Random isolated incidents of intake stroke ignition, which are so difficult to eliminate in backfiring "carbureted" engines, are barely noticeable with Parallel Induction.

Figure 3 is a schematic of a Parallel Induction hydrogen engine with turbocharging and aftercooling. Air is compressed in the turbocharger and cooled in the aftercooler before arriving at the air throttle. This dense air, when burned with hydrogen, produces power comparable to a naturally aspirated diesel.

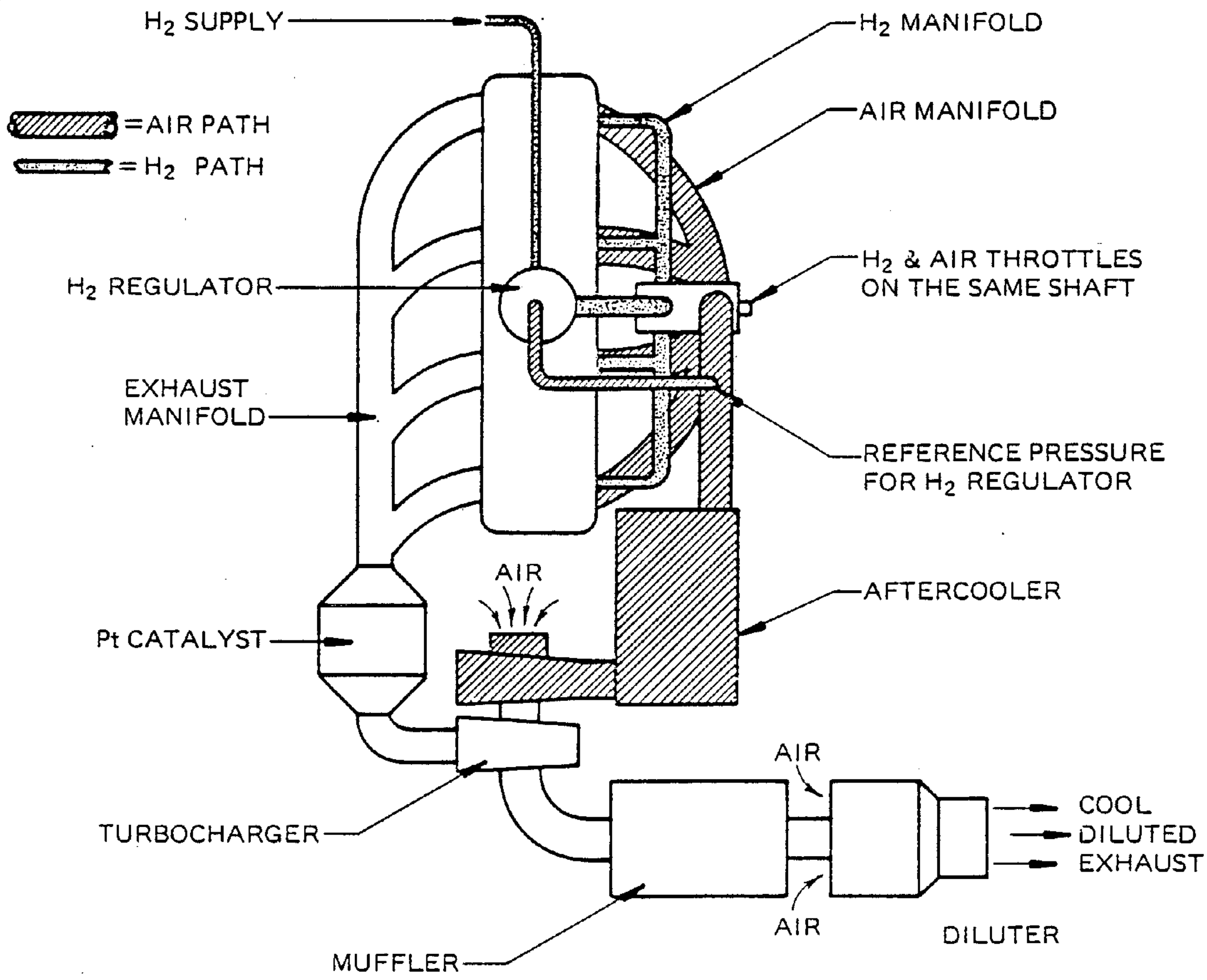


Figure 3.
Schematic of intake and exhaust systems in a turbocharged, aftercooled, Parallel Induction hydrogen engine.

The air throttle is a conventional "butterfly" throttle like those commonly used in gasoline carburetors. Air flows through the manifold and past the intake valves in the usual way except that it does not mix with fuel until the last instant before entering the cylinders.

The hydrogen path is similar to the air path. A regulator matches the hydrogen pressure above the hydrogen throttle to the air pressure above the air throttle. Hydrogen flows past its own throttle and through its own manifold and intake valves on the way to the intake ports. The Caterpillar engine has delayed hydrogen valves which open 30° of crank angle after the air intake valves have opened. This delay is not strictly in keeping with the concept of Parallel Induction since the hydrogen flows during a reduced portion of the intake stroke compared to the air. Despite this small dissimilarity, the hydrogen and air flows remain proportional throughout the speed and load ranges of the engine.

A throttle valve assembly is shown schematically in Figure 4. The larger throttle bore is for air, the smaller for hydrogen. Hydrogen and air mix in the cylinder during intake and compression, are spark-ignited, burned and exhausted in the normal way. An oxidation catalyst in the exhaust is used to prevent afterfiring or exhaust explosions by burning any fuel-air mixture which is not burned in the engine. The catalyst is not actually necessary since hydrogen is not toxic, but if the engine malfunctions (faulty ignition system, for example), loud explosions in the exhaust may result if the catalyst is omitted. The catalyst also eliminates a faint oil odor which normally is emitted by a hydrogen engine's exhaust. The present test set-up does not include a catalyst but data taken previously with the 3304 show its effectiveness (1).

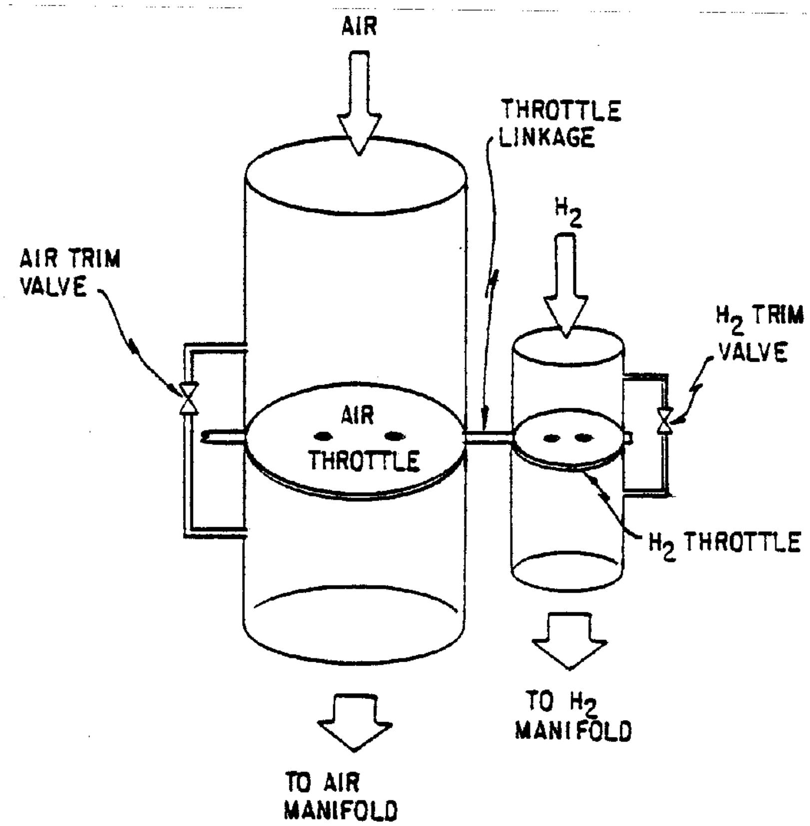


Figure 4.
Schematic of a Parallel Induction throttle valve assembly.

The exhaust flow drives the turbine in the turbocharger and exits through the muffler, as in any turbocharged engine. The turbine housing is considerably smaller than its counterpart in diesel engines of equal displacement since a smaller volume of gas at a lower density and a lower temperature is passed through it. A boost pressure regulator or "wastegate" regulates the intake manifold pressure to prevent preignition. The pressure limit is determined in part by the effectiveness of the aftercooler. Temperatures in excess of 93°C nearly always cause preignition.

The diluter shown in Figure 3 is intended to reduce the exhaust gas temperature before release by mixing the exhaust with ambient air. In certain cases, this is a requirement for mining vehicles.

The ignition system is set to produce the spark timing required for the hydrogen-air ratios supplied to the engine. One of the advantages of using Parallel Induction is that the equivalence ratio is nearly constant and this allows nearly constant spark timing. If large changes in equivalence ratio are designed into a hydrogen fuel control system then large changes in spark timing must be coordinated to equivalence ratio or poor performance will result. A Fairbanks Morse solid-state magneto provides ignition for the Caterpillar 3304. The timing is set near top-center for starting and increases automatically to a fixed value (ca. 13° BTC) at operating speeds. Data provided subsequently in this report will show that this proven and rugged ignition system is well matched to the ignition timing requirements of a constant equivalence ratio hydrogen engine.

Hydrogen-Air Metering Principles

The basic principle of Parallel Induction mixture control is to provide two parallel intake paths, one for hydrogen, one for air, whose relative sizes fix the proportion of the hydrogen and air entering the engine's cylinders.

The first aspect of hydrogen-air flow similarity to be established in the design of a Parallel Induction system is pressure. Air arrives at the mouth of an air throttle valve with a certain stagnation* pressure which is measured by a Pitot tube and sent to the hydrogen regulator as a control signal. The hydrogen regulator responds to this control signal by matching its output pressure to the air stagnation pressure. In this way the stagnation pressures of the hydrogen and air streams just above their respective throttle valves are made equal.

The hydrogen and air eventually arrive at the same place (the intake port) so their downstream pressures are identical. This downstream pressure pulsates as the pistons accelerate and decelerate through their intake strokes. For the purpose of understanding the control principles of Parallel Induction, these pulsations may be ignored and the flows may be thought of as steady.

Having established an equal pressure drop across each of the two parallel gas paths, the flow ratio of the hydrogen and air will be determined by the different properties of the two gases and by the different dimensions of the two paths.

The hydrogen intake system installed on the Caterpillar 3304 was fabricated as shown in Figure 5. The air intake valve and guide were altered to admit both hydrogen and air to the engine. A copper tube routes hydrogen through the air intake port to a sleeve-type

*"Stagnation" pressure of the air flow is the pressure which would exist if the air duct were expanded until the velocity becomes negligible.

valve and seat built onto the original air intake valve and guide. The sleeve valve delays the admission of hydrogen until 30 degrees of crank rotation after the air intake valve has opened. This type of valve mechanism and the benefits of delayed hydrogen injection were proven by Adt, Swain and Pappas (4).

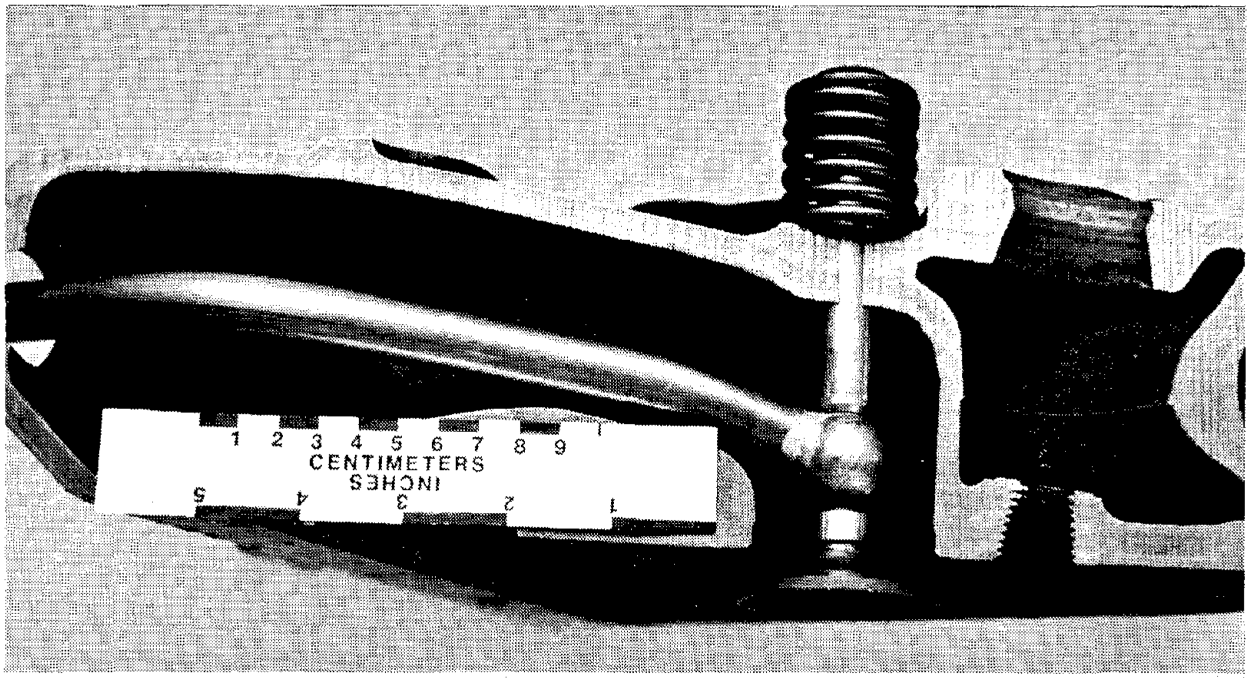


Figure 5.
Sectional view of hydrogen valve, seat and induction tube
in the intake port of the Caterpillar 3304.

There was some concern during the design of the Parallel Induction system that the hydrogen valve delay would interfere with the metering process, since the hydrogen flow period is not exaactly the same as the air flow period. This problem never materialized, as evidenced by the data of Figure 6.

Once an engine has been assembled and tuning has commenced, the only intake system dimensions which may be conveniently varied are the throttle bores. It is recognized, however, that the manifold, ports and valves add to the total restriction in the hydrogen and air systems, so care is taken to minimize restrictions in these places and to make each port, valve and manifold passage as nearly identical as possible.

The throttle bores must be small enough so that they are the most significant restrictions in each flow. On the other hand, the throttle bores must be large enough to allow adequate flow. Their sizes are roughly calculated using idealized flow equations* so as to give no more than about 3 kPa of pressure drop during wide open throttle operation below the engine speed where turbocharging begins. When the turbocharger begins to pressurize the air system, small pressure drops in the throttles are not very important. As shown in Figure 7, the Ishi-Warner turbocharger produces enough boost throughout the rated speed range to dwarf the small losses in the throttle valve.

*See Appendix D - Computer Modeling of Parallel Induction Mixture Control.

Having thus fixed the approximate size of the throttle valves so that they are small enough to control accurately, yet large enough to allow sufficient flow, the next task is to adjust their relative sizes to give the precise mixture control which idealized calculations cannot give. Up to this point in the explanation of Parallel Induction control techniques, hydrogen supply pressure has been described as being "equal" to air system pressure. This is not exactly true in some engine conversions produced by HES. It is sometimes beneficial to supply hydrogen at slight positive pressures to give good engine characteristics at low speeds. The exact size of the hydrogen throttle bore and the exact setting of the pressure regulator are determined by direct measurement of fuel-air equivalence ratio over the speed range of the engine. At very low engine speeds where turbocharging is of no help in producing engine torque, slight enrichment of the mixture makes the engine feel more responsive to the operator of a vehicle. The richer mixture will cause increased NO_x but only in the small portion of the useable RPM range where enrichment occurs. Not much low-speed enrichment has been designed into the Caterpillar engine, since the torque converter characteristic of a mining vehicle precludes speeds below the onset of turbocharge pressure. During idling, minor differences between the clearances in the hydrogen and air throttles cause mixture variations. The trim screws, shown schematically in Figure 4, are used to correct this problem. The Caterpillar 3304 only needed a trim screw on the air throttle to correct a tendency to idle richer than the design mixture.

The relative temperatures of the hydrogen and air flows can affect the fuel-air ratio. Under part-load conditions both flows are essentially at ambient temperature. At wide-open throttle the air temperature rises during compression in the turbocharger. The air manifold temperature was held at 40^{o}C during the testing described in this report by adjusting a tap water flow through the tube-side of a heat exchanger. Air-to-air aftercoolers are available which can perform this function by limiting manifold temperatures to 20^{o}C above ambient. The metering compensation needed to counter this minor heating effect is accomplished during tuning of an air-to-air aftercooled system (5).

Fuel System Modifications

HES' previous experience with Parallel Induction was limited to modifying Mitsubishi, MCA-Jet type engines which are factory equipped with two intake valves and two intake manifolds. The Caterpillar 3304 engine has no such separate, parallel path which may be modified for hydrogen delivery. A literature survey conducted earlier in the project provided several possibilities for constructing the required valving mechanism. The simplest most reliable device appeared to be the sleeve-valve design of Adt, Swain & Pappas (4). HES' version of this design is shown in Figure 5. The sleeve delays the flow of hydrogen until after the air flow has begun. This decreases the likelihood of the incoming hydrogen being ignited by hot residual gases from the previous cycle.

This delayed timing caused some apprehension about the possible effect on metering strategy. The concept of Parallel Induction requires a similarity between the way hydrogen and air flow into the engine. The delay caused by the hydrogen sleeve valve is dissimilar from the action of the air intake valve.

Figure 8 shows the components which are fabricated or modified to deliver hydrogen fuel to the intake port of the Caterpillar 3304. The sleeve valve is rough-machined from Invar, silver-soldered to the valve stem and ground to final dimensions. The cast iron sleeve valve seat is rough-machined, silver-soldered to the copper induction tube and pressed into the modified Caterpillar valve guide. After assembly the upper portion of the seat is reamed concentrically with the guide. The valve guide was also modified to accept a valve seal of the type commonly used on gasoline engines (TRW VP 36). The

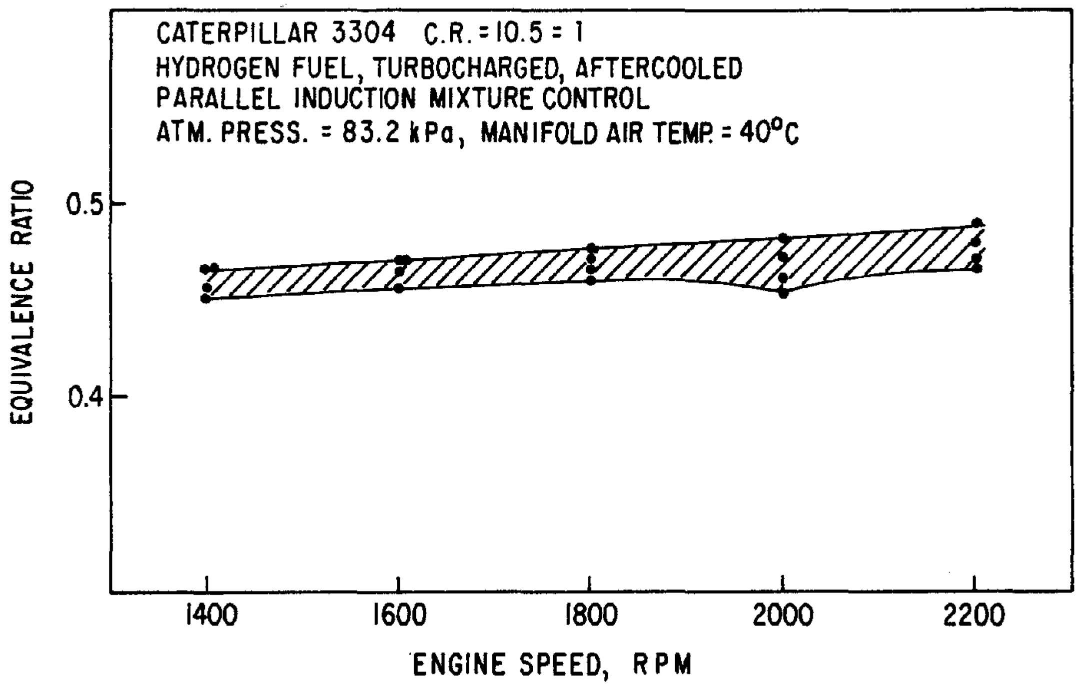


Figure 6.
Equivalence ratio vs rpm at various loads
shows nearly constant mixture under all operating conditions.

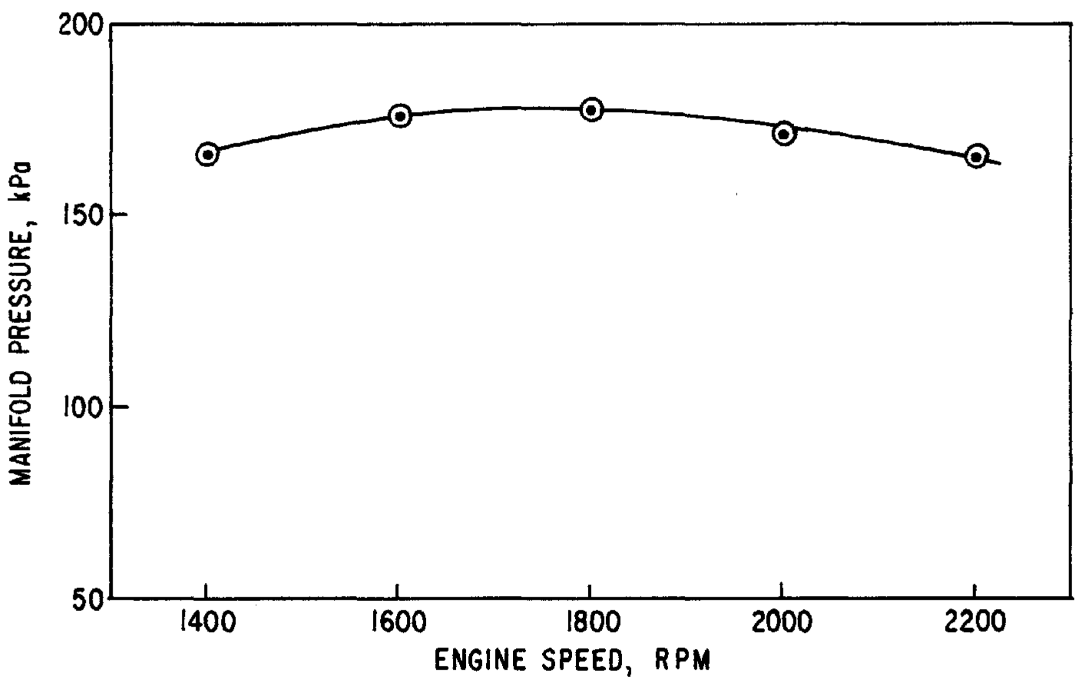


Figure 7.
Turbocharger supplies adequate boost at all speeds.

spring, retainer and keepers are standard Caterpillar parts. During final assembly the clearance between the sleeve valve and seat is lapped by spinning the valve in its corresponding guide. The resulting clearance should not change much as a result of changes in engine temperature since Invar has a very low coefficient of thermal expansion and the cast iron seat is kept relatively cool by the induction tube which acts as a cooling fin. The assembled components were shown in Figure 5.

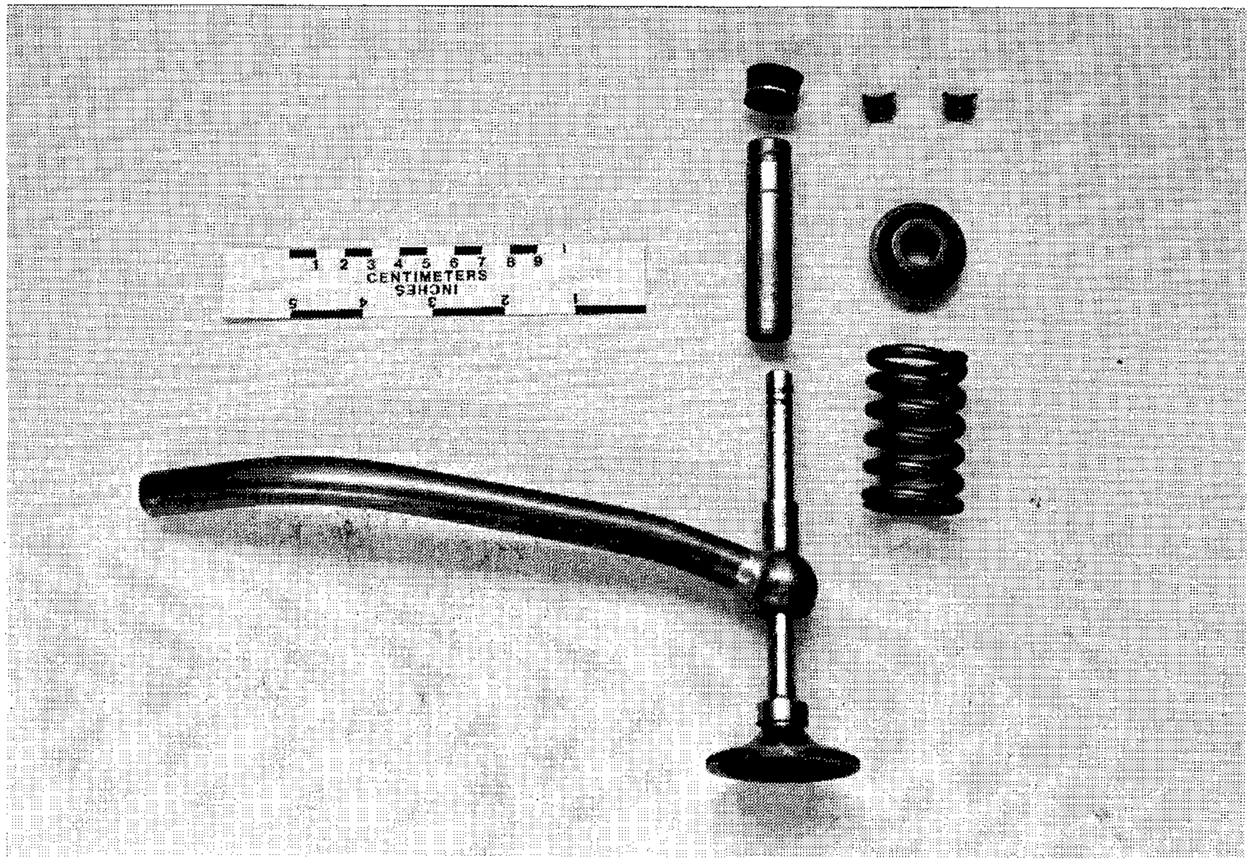


Figure 8.
Hydrogen sleeve valve and related components.

Hydrogen does not begin to flow until the upper edge of the sleeve clears the lower edge of the seat. This delays the onset of hydrogen flow until 30° ATC on the intake stroke. As the intake valve closes toward the bottom of the intake stroke the hydrogen valve closes at about 160° ATC. This permits the intake port to be purged of hydrogen by the last fraction of the air flow into the cylinder. The hydrogen sleeve-valve, seat and manifolding were sized large enough so that they are not significant flow restrictions compared to the hydrogen throttle.

A seal was added to the top of the valve guide to inhibit hydrogen leakage up the guide and into the rocker cover area. This seal may be a critical maintenance item as the engine wears and guide clearances increase.

The hydrogen intake manifold (See Figure 9) is bolted to the outside of the air intake manifold. Hydrogen is distributed to each cylinder through tubes bent to follow the air intake port. The tubes are silver-soldered to each sleeve valve seat at one end and enter the hydrogen manifold through O-ring seals at the other end.

The throttle valve shown schematically in Figure 4 regulates the proportion of hydrogen and air entering their respective manifolds. The 1.5 inch air throttle is large enough to flow sufficient air before the onset of boost. After boost pressures rise to operating levels, some pressure drop is acceptable for the sake of tight control of equivalence ratio. The throttle valve assembly also contains a relief valve to prevent shock waves from occuring in the intake system during rapid deceleration. Figure 10 is a photo of the assembly.

The Parallel Induction system worked without any mechanical problems during 25 hours of testing. The only need for improvement of the design comes from a significant loss in volumetric efficiency (Figure 11). The induction tube and sleeve valve seat disturb the geometry of the intake port which was designed by Caterpillar for maximum flow. This problem was anticipated and reduced with the assistance of Mr. Lloyd Johnson of Leejon Engineering, Peoria, Illinois.

The rapid loss of volumetric efficiency above 2000 RPM is due to an undersized turbine housing which causes excessive, exhaust pressures at higher engine speeds. This will be remedied during future phases of the project by a turbine with a greater area/radius ratio. Aside from this poorly chosen turbine housing, the Ishi-Warner RHB6 turbocharger worked well throughout the performance range of the engine. The compressor trim is designated 406 Bz2. The turbine housing was the P11 (A/R=.43) type which should be increased to the P13(A/R=.51) type for future work. A special adaptor plate was fabricated to adapt this turbocharger to the Caterpillar exhaust manifold. The aftercooler was a Young Radiator Company, Model F-602-EY-4P, Part No. 161141. All turbocharged operating conditions were regulated to $40 \pm 0.5^{o}C$ by manually adjusting tap water flow in the tube-side of the heat exchanger. An air-to-air type after-cooler may be used when the engine is installed in a vehicle.

IGNITION SYSTEM

Ignition systems used during past phases of this project were manually adjusted to the minimum best torque settings for a broad range of fuel-air ratios. These experimental units were not rugged enough for future phases of the project.

A Fairbanks-Morse (Model SCSA4BV36) solid-state magneto was adapted to the 3304 using Caterpillar gas engine magneto drive components. These heavy-duty components are typical of industrial gas engine ignition systems which may be expected to provide trouble-free service for thousands of hours. The weakest link in such ignition systems is the spark plug which may require periodic replacement to avoid fouling, although there was no evidence of fouling during the tests.

The magneto retards the timing by 10-20 degrees during cranking to facilitate engine starting. The exact value of this retardation is adjusted from a potentiometer supplied with the magneto.

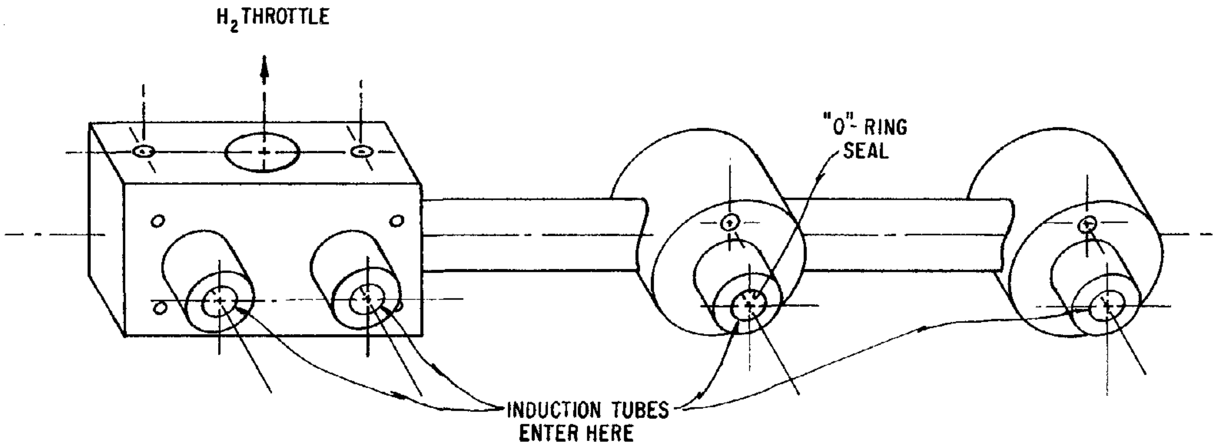


Figure 9.
Intake Manifolding
Hydrogen manifold bolts onto the side of the air manifold
of the Caterpillar 3304.
Hydrogen induction tubes enter through o-ring seal.

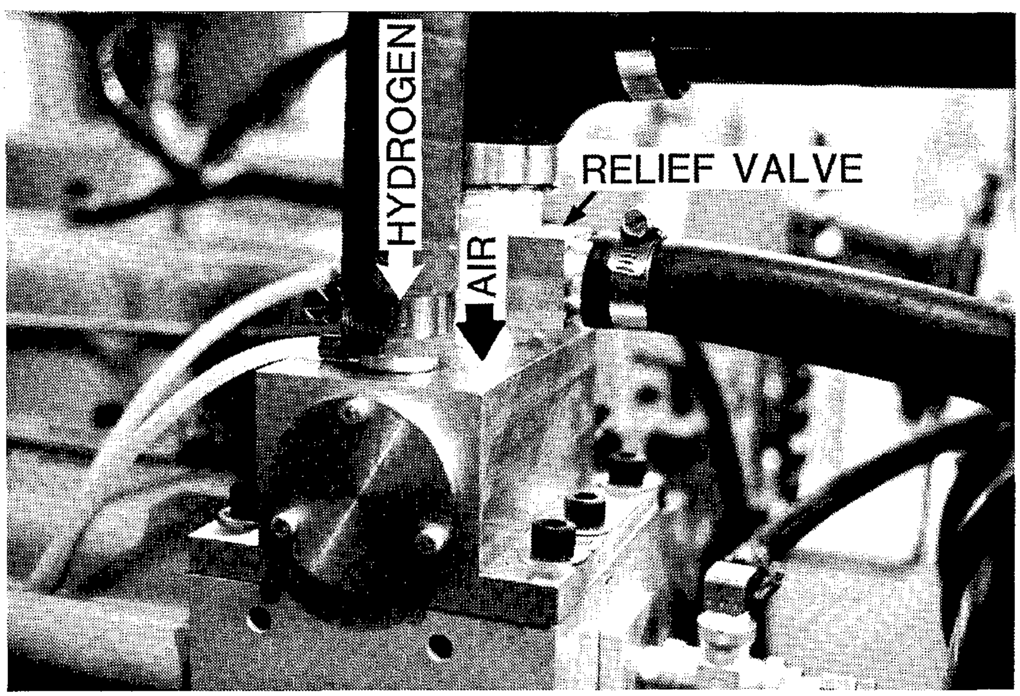


Figure 10.
Throttle valve assembly mounted on Caterpillar 3304

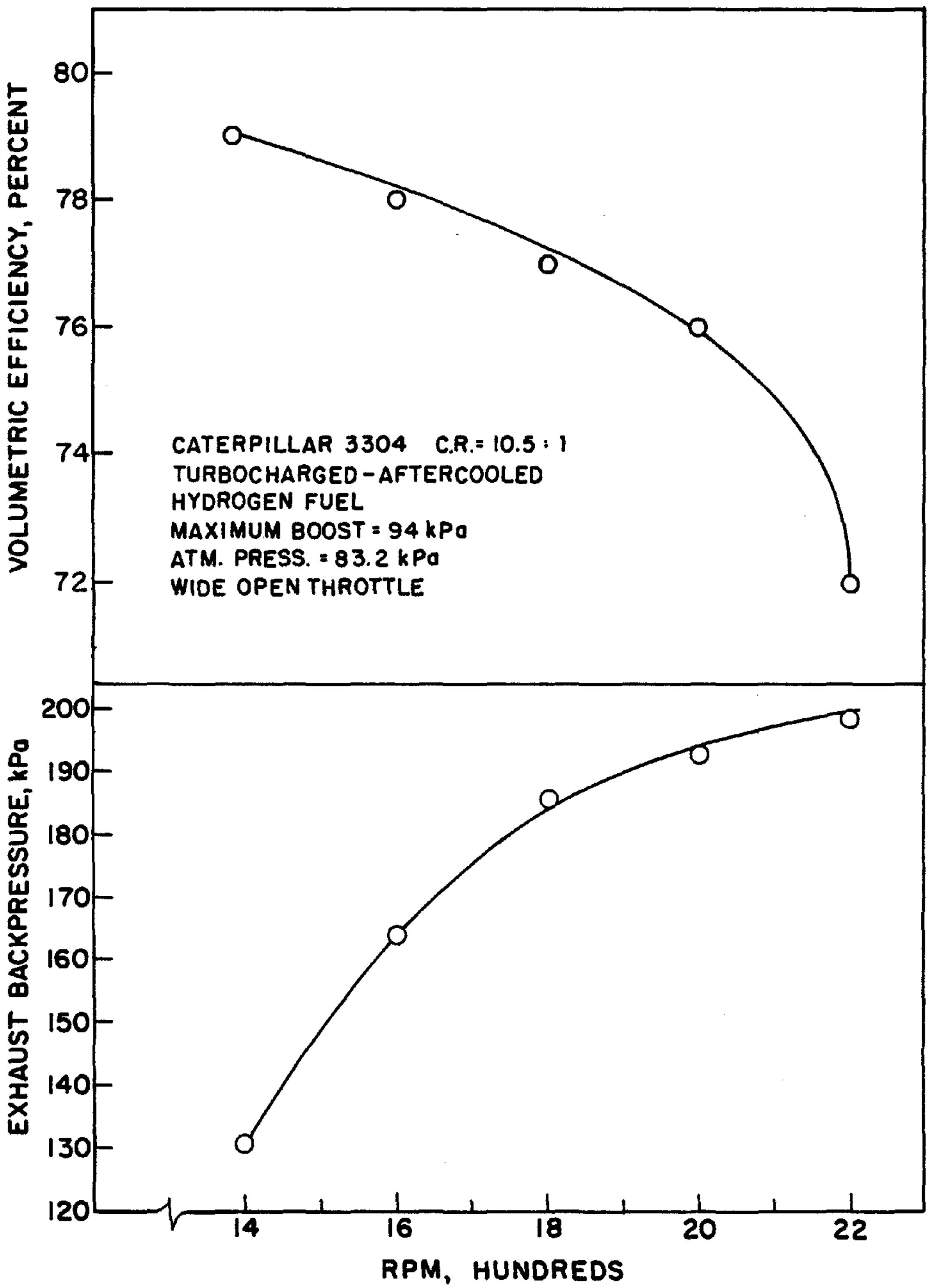


Figure 11.
Exhaust backpressure and volumetric efficiency vs engine speed
show a need for a larger turbine housing on the
Ishi-Warner turbocharger.

A 180 volt distribution system sequences the firing of individual coils screwed onto each spark plug. Fairbanks-Morse recommends the use of special spark plugs with threaded bodies with these coils. Since these plugs are not available in a variety of heat ranges, it was decided to modify conventional 14 mm spark plugs (Champion J-4J) to fit the coils. Figure 12 is an exploded view of the coil and plug. The hexagonal portion of the plug body was turned round and threaded for 13/16-20. A 3/4" snap-ring limits the depth of the plug in the coil.

The short, 3/8" reach spark plugs were used in the 3/4" reach Caterpillar spark plug adaptors, which were modified earlier in the project to form shroud chambers around the plug electrodes (1). This shroud, shown in Figure 13, prevents ignition of the hydrogen-air mixture until late in the compression stroke, when a fresh charge has been compressed into the shroud chamber. This prevents mistimed sparks or hot spark plug surfaces from igniting the mixture prematurely. It was known from previous work with constant equivalence ratio hydrogen engines that correct ignition timing would not change much with engine speed or load. To verify this, tests were conducted throughout the operating range of the 3304 to determine the correct ignition timing.

In a turbocharged engine, the determination of correct spark timing is complicated by a rise in boost pressure as spark timing is retarded. This results in a reduction in thermal efficiency, more energy in the exhaust stream, an increase in boost and the impression of constant performance while in actuality, the situation is higher fuel consumption. This precludes a simple "best-torque" determination. It was decided to use brake-specific fuel consumption as the criterion for determining correct timing. Figure 14 shows data taken over the range of speeds and loads for which the 3304 is designed. A normalized basis is used so that all data may be accurately plotted on the same scale. This was accomplished by dividing each datum by the minimum datum for that speed and load. A normalized $BSNO_x$ value of 2 means that twice the minimum NO_x is being produced. A normalized BSFC value of 1.03 means that fuel consumption is 3% higher than the minimum value.

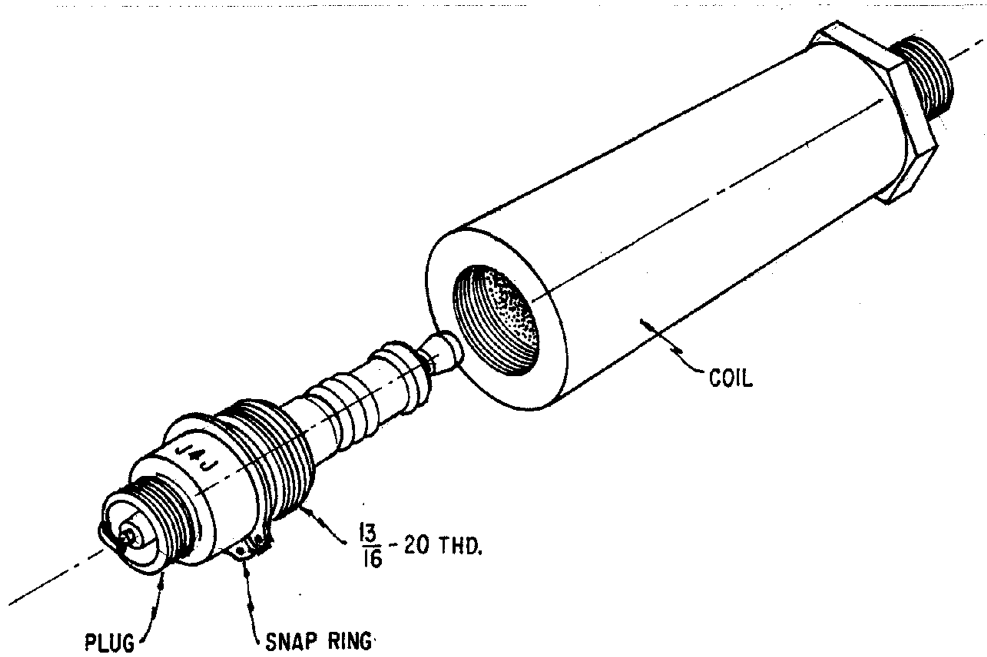


Figure 12.
Modified spark plug with snap-ring fits standard Fairbanks-Morse coil.

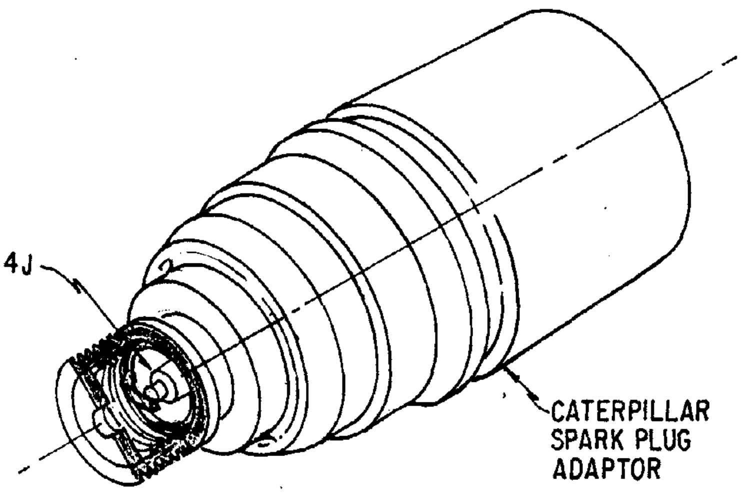


Figure 13.
Caterpillar spark plug adaptor has a plug with a hole in it welded to the end to form a shroud chamber around the plug electrodes.

The first thing to note in Figure 14 is that advancing the timing beyond 19^{o} BTC causes preignition at 2200 rpm and 100% load. This must be avoided regardless of fuel consumption or NO_x considerations, since severe preignitions would eventually damage the engine.

The brake-specific fuel consumption data indicate that 13^{o} BTC is the best timing for 1400 rpm and 17^{o} BTC is best for 2200 rpm. These conclusions are not affected by load. It is inferred that 4^{o} of centrifugal advance would give minimum fuel consumption at all speeds and that no vacuum advance or boost-pressure retard functions are needed. The Fairbanks-Morse magneto gives about 2^{o} of "centrifugal" advance between 1400 and 2200 rpm and has no manifold pressure control. The compromises involved in using the Fairbanks-Morse ignition system compared to an ideal ignition system may be as little as $\pm 1^{o}$. For example, it may be set for 14^{o} BTC at 1400 rpm being 1^{o} advanced from optimum. At 2200 rpm it will be 16^{o} BTC, 1^{o} retarded from optimum.

Also shown in Figure 14 are normalized NO_x data which clearly point out why minimizing advance is desirable. Regardless of speed or load, NO_x is reduced by retarding the spark to about 10^{o} BTC, below which further improvements are small.

The NO_x trend encourages decreasing spark advance from fuel consumption optimized values toward 10^{o} BTC. The region between 10 and 15^{o} BTC is the approximate range of compromise between minimum BSFC and minimum $BSNO_x$. The timing was set at 13^{o} BTC at 1800 rpm and hence ranged from 12^{o} BTC at 1400 rpm to 14^{o} BTC at 2200 rpm for the series of tests run during this test phase. The marginal improvements possible by more sophisticated timing control are not worthy of pursuit at this stage of development.

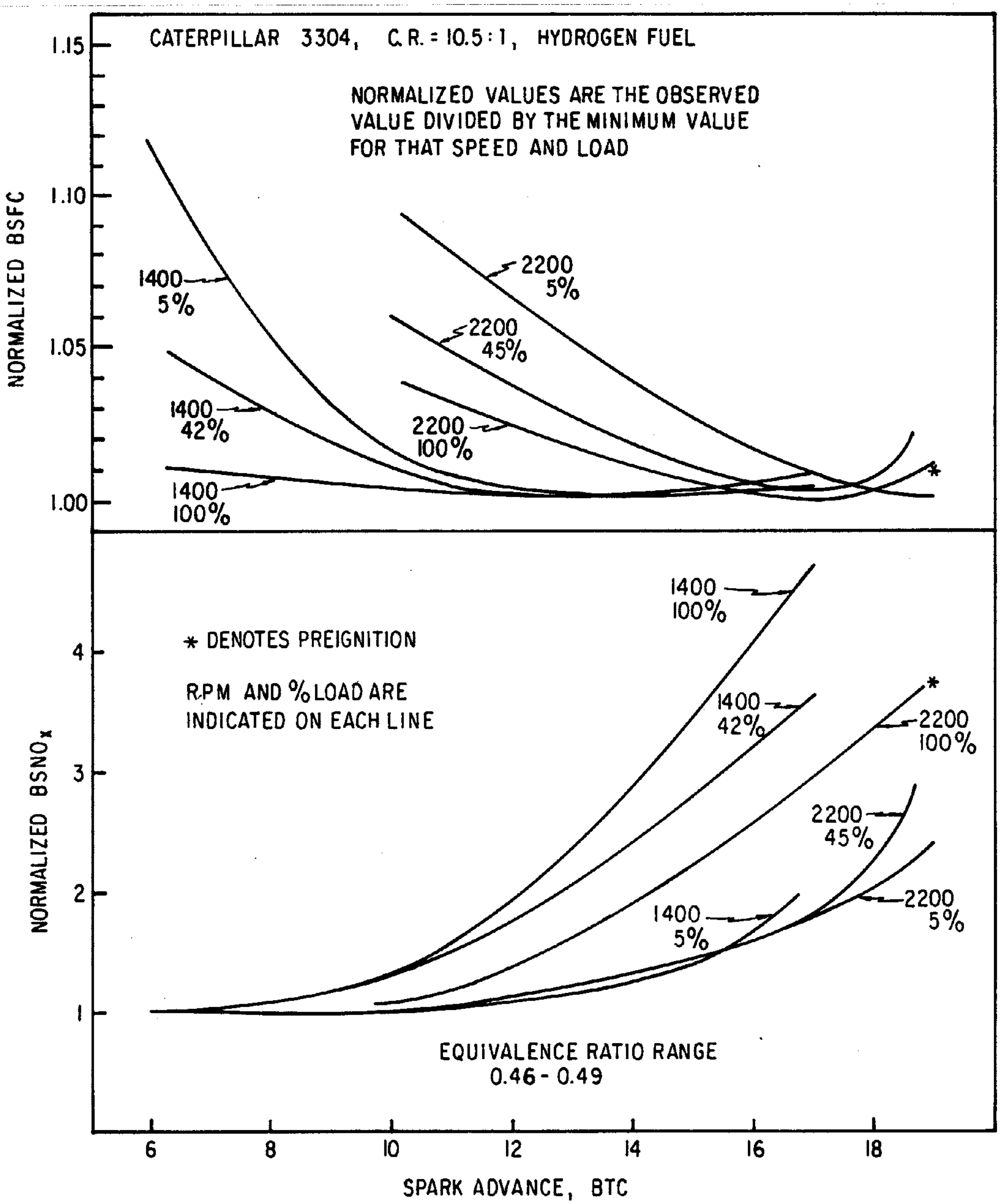


Figure 14.
Normalized fuel consumption and emissions show that the spark timing needed for best results is about 13^{o} BTC at all speeds and loads.

PERFORMANCE

In its present configuration the turbocharged-aftercooled (TA) hydrogen engine is tuned to approximate the sea level performance characteristics of the naturally aspirated (NA) diesel. This is about 20% more than the diesel's rated output at Denver's altitude of 1600 meters. Figure 15 plots the brake mean effective pressure (BMEP) power and brake specific fuel consumption capabilities of the TA hydrogen engine over the rated speed range. The NA diesel data at Denver's altitude are shown as dashed lines.

Figure 16 shows the fuel consumption of the engine at part load conditions. The previous data, taken when the engine was unthrottled and used premixed hydrogen-air mixtures, is indicated by the shaded region. The NA diesel data are shown as dashed lines for reference purposes. Diesel data were related to hydrogen data on a lower heating value basis. During high boost operating conditions the data from the hydrogen flowmeter were found to be incorrect (because of leakage in the differential pressure lines from the flow element to the manometer.) Therefore, hydrogen consumption was determined from air flow and exhaust oxygen concentration. These fuel consumption data (Figure 16) were verified by discharging known volumes of compressed hydrogen while accounting for compressibility and temperature effects. This verification is indicated with an asterisk in Figure 16 at 750 kPa BMEP. The only significant differences between the premixed and Parallel Induction data are that the unthrottled, premixed engine is less efficient at light loads due to incomplete combustion. At full loads the Parallel Induction engine also has an efficiency advantage but this is because of the greater BMEP which results from the greater boost pressure used. The peak brake mean effective pressure is limited to just over 700 kPa (102 psi) by restricting equivalence ratio to about 0.48 and by a turbocharger selection which yields about 2:1 pressure ratio. If the engine were operated "as is" at sea-level it would have a significant increase in output. Low altitude users could either take advantage of the increased power or detune the engine to produce lower NO_x emissions. Very high altitude users could achieve standard performance by increasing equivalence ratio, boost or both. At 690 kPa (100 psi) of BMEP there is little risk of preignition. The engine will yield 900 kPa (130 psi) of BMEP for indefinite periods if the equivalence ratio is richened to about 0.52 and the ignition timing is correspondingly retarded. This heavy loading is very near the preignition limit and is not recommended for continuous service at Denver's altitude. It may be acceptable to do so at low altitude with leaner mixtures which are less susceptible to preignition.

The equivalence ratio control of the Parallel Induction system over the speed range of the engine was shown previously in Figure 6. The mixture is constant at 0.48 ± 0.01. The slight enrichment at 2200 rpm full load, is enough to cause a relatively high NO_x emission compared to all other data taken from this engine. This will be remedied during future phases of the project.

Cylinder-to-cylinder distribution was determined by oxygen analysis in each exhaust port. No errors were found within the precision of the method, which is $\pm 0.05\%$ oxygen or about 0.4% uncertainty of the stated value for equivalence ratio, assuming complete combustion.

Other evidence for good cylinder-to-cylinder distribution of mixture comes from the NO_x data presented in the Emissions section of this report. Figure 19 shows that the NO_x data for Parallel Induction are just what they were with premixed hydrogen air mixtures. This should be a very sensitive indicator of distribution errors or even cycle-to-cycle differences in equivalence ratio, since a rich cycle and a lean cycle yield much higher NO_x than that of the average mixture.

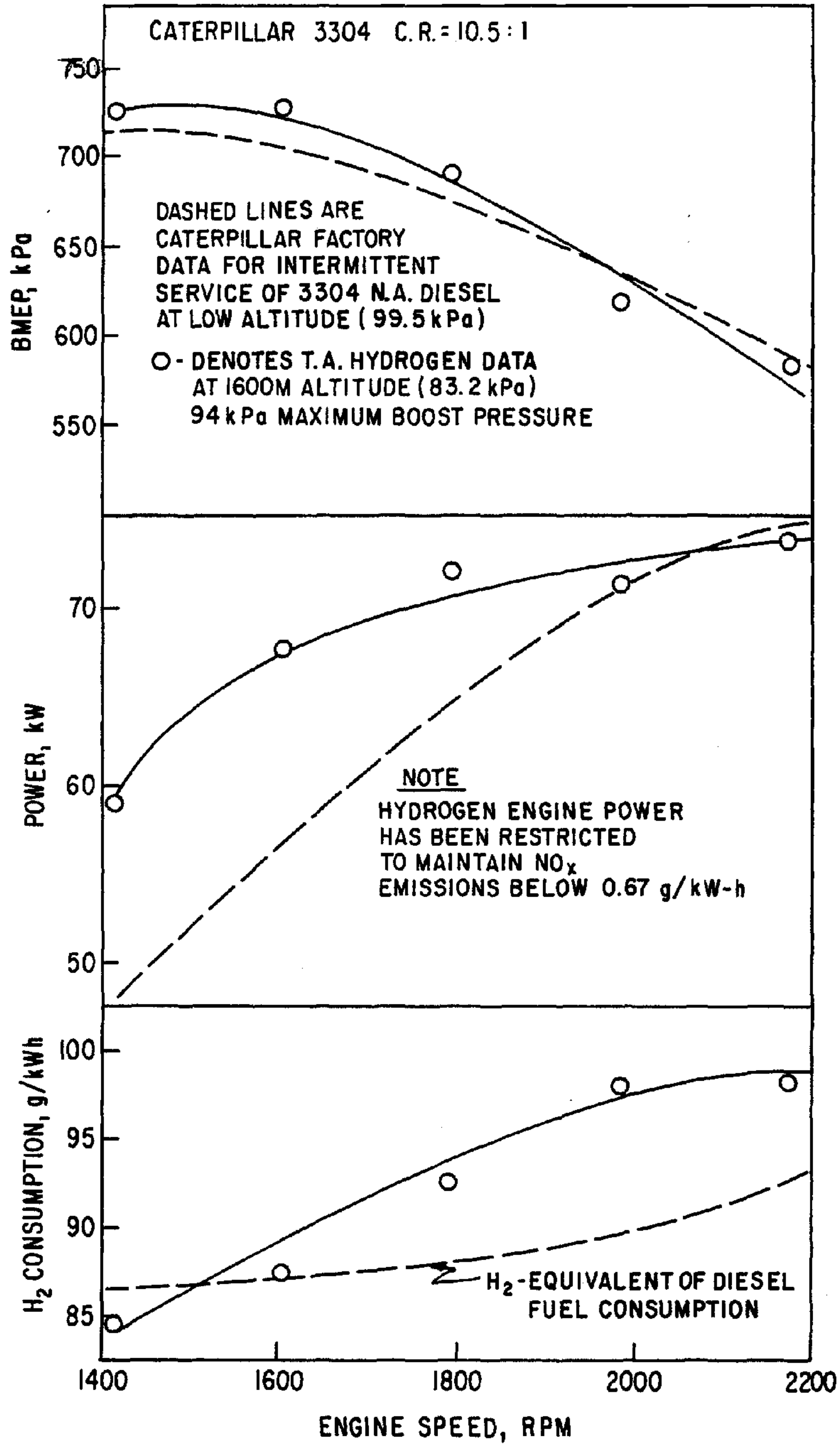


Figure 15.
Brake mean effective pressure, power and fuel consumption comparison between the TA hydrogen engine at 1600m altitude and NA diesel at low altitude.

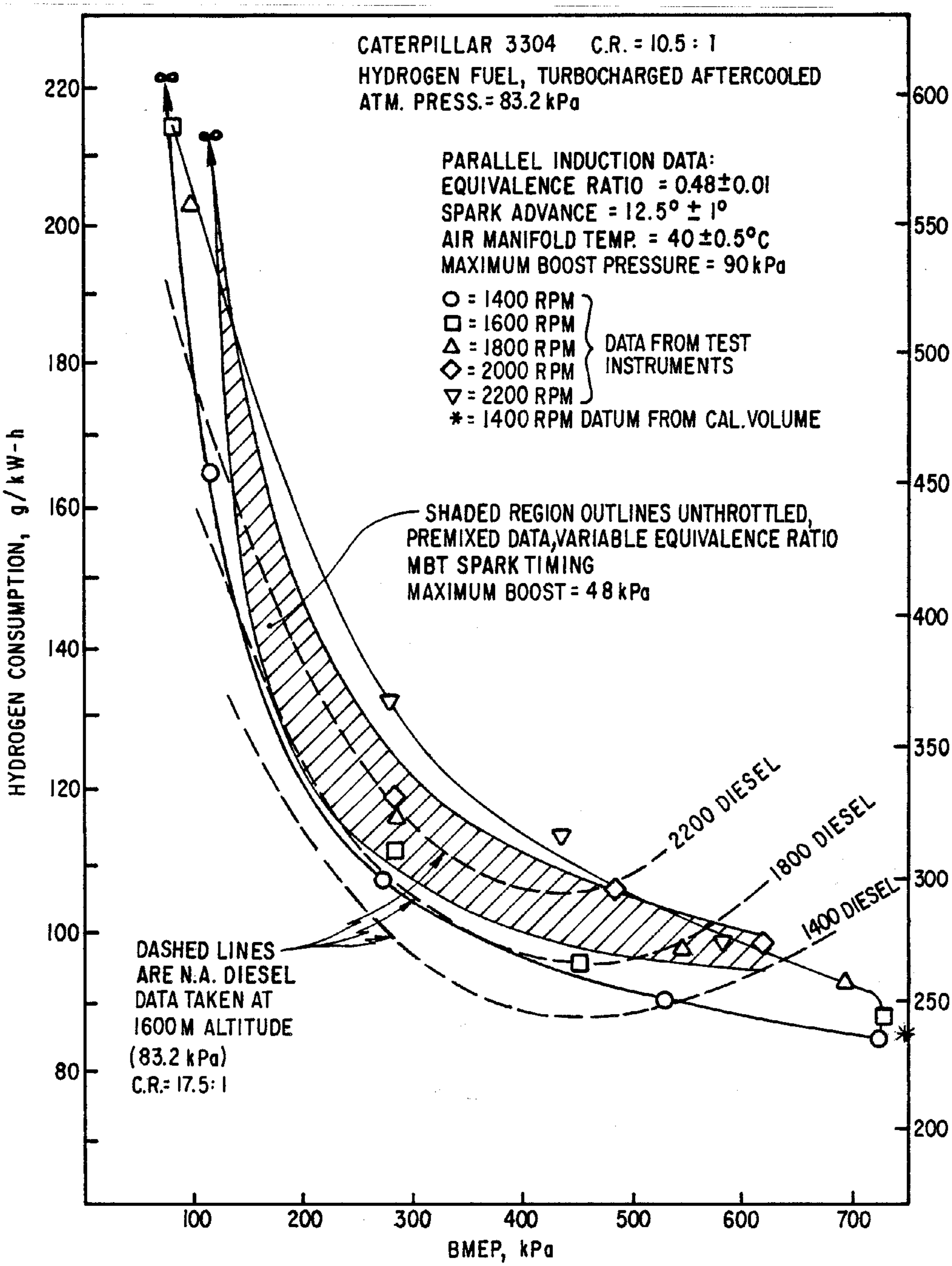


Figure 16.
Brake specific hydrogen consumption vs brake mean effective pressure for the Parallel Induction hydrogen engine compared to premixed hydrogen data (shaded region) and diesel data (dashed lines).

EMISSIONS

Most of Eimco's mining vehicles are naturally aspirated prechamber diesels. These engines are derated to about 690 kPa BMEP at sea-level to limit smoke and CO emissions to acceptable levels. The objective of the hydrogen engine conversion was to equal this BMEP while reducing smoke and CO emissions to zero and limiting NO_x to no more than 10% of what the diesels emit.

It is known from previous work on this engine that these objectives could be met through turbocharging, especially if an aftercooler were used to remove heat from the compressed intake air. There is another dimension to be considered in achieving a fixed BMEP objective in a turbocharged-aftercooled hydrogen engine. The greater the density of the intake manifold air, the lower the equivalence ratio required for a given power level. Figure 17 illustrates the nature of this choice. Several different boost pressures were set by controlling the wastegate of the turbocharger. The mixture was then brought to a value which produced 690 kPa of BMEP and NO_x emissions, fuel consumption, etc. were recorded. It is clearly better, from an emissions viewpoint, to use high boost and a lean equivalence ratio than less boost and a richer equivalence ratio. There is little improvement possible by decreasing equivalence ratio below 0.48 so this is where the present design was placed. This corresponds to a pressure ratio of about 2:1 in the compressor of the turbocharger.

Figure 1 compared the NO_x emissions of the hydrogen-converted 3304 TA engine with two prechamber diesels commonly used in underground mining. The 3304 NA diesel data were taken from the test engine at Denver's altitude, before conversion to hydrogen. The Deutz F8L714 data are from the factory at low altitude. The diesels peak near 10 g/kWh and average* about 6 or 7 g/kWh. The TA hydrogen engine with Parallel Induction produces peak NO_x near 0.7 g/kWh and averages* about 0.3 or 0.4 g/kWh.

The same hydrogen engine NO_x data are presented in more detail in Figure 18. If it were not for the relatively high emissions at 2200 rpm and 590 kPa, the data would all group below 0.5 g/kWh. The reason for this high emission is shown in Figure 19. The wide-open throttle data from Figure 18 are replotted along with premixed engine data in Figure 19 to point out the consequences of a slight mixture enrichment which occurred at 2200 rpm.

It is in the emissions area where Parallel Induction compares unfavorably with the unthrottled form of the hydrogen engine. Figure 20 shows brake-specific NO_x vs brake mean effective pressure for the Parallel Induction engine, data taken from the same engine unthrottled and estimated data for an unthrottled engine with the same turbocharger as the present engine. This shows that there is significant room for improvement in NO_x emissions if the problems of developing a practical variable equivalence ratio engine can be solved.

*"average" brake-specific emissions do not include data below 50 kPa of BMEP since brake-specific values are undefined at zero BMEP.

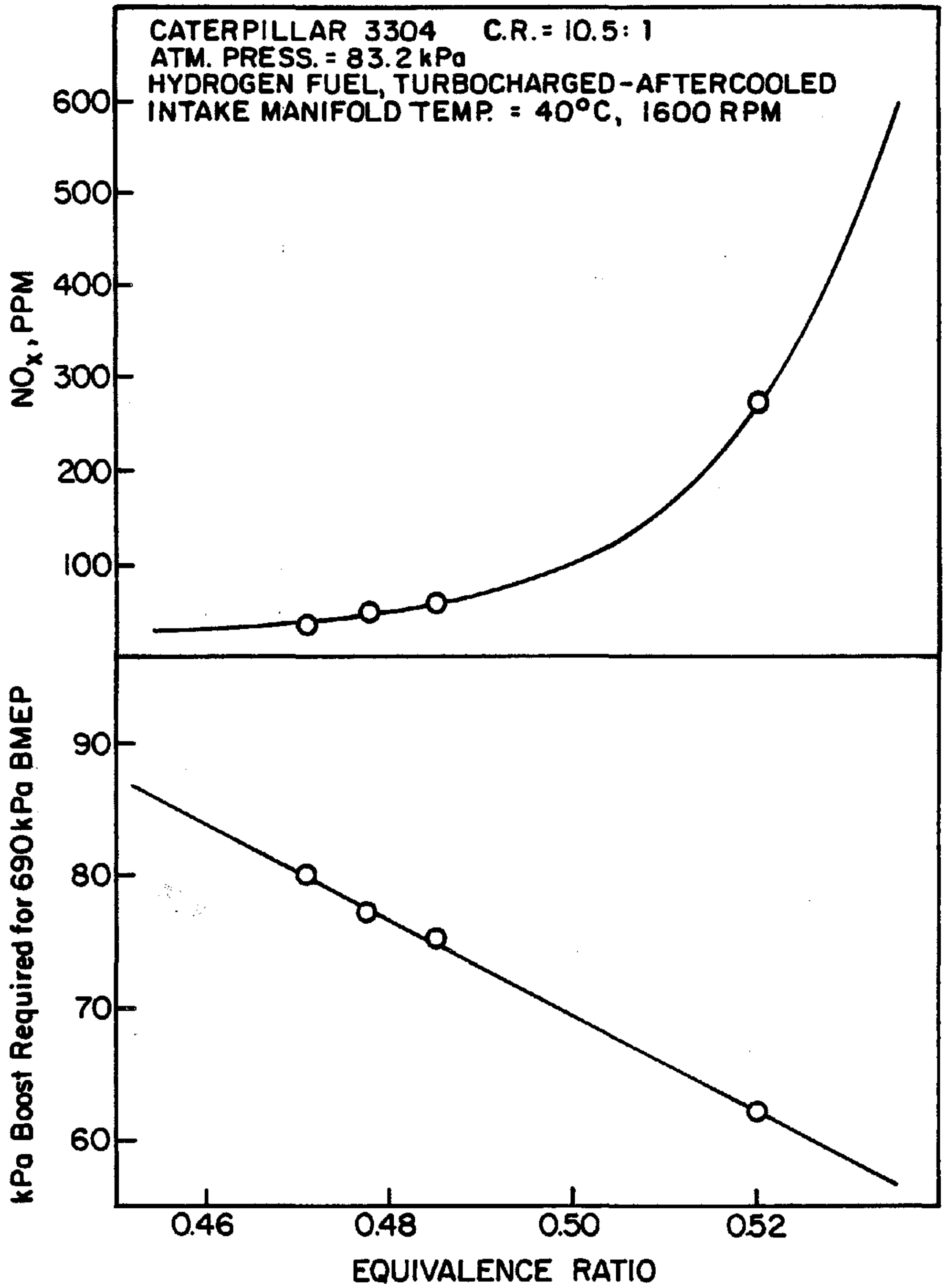


Figure 17.
NO_x and boost pressure variations while maintaining constant brake mean effective pressure with various equivalence ratios.

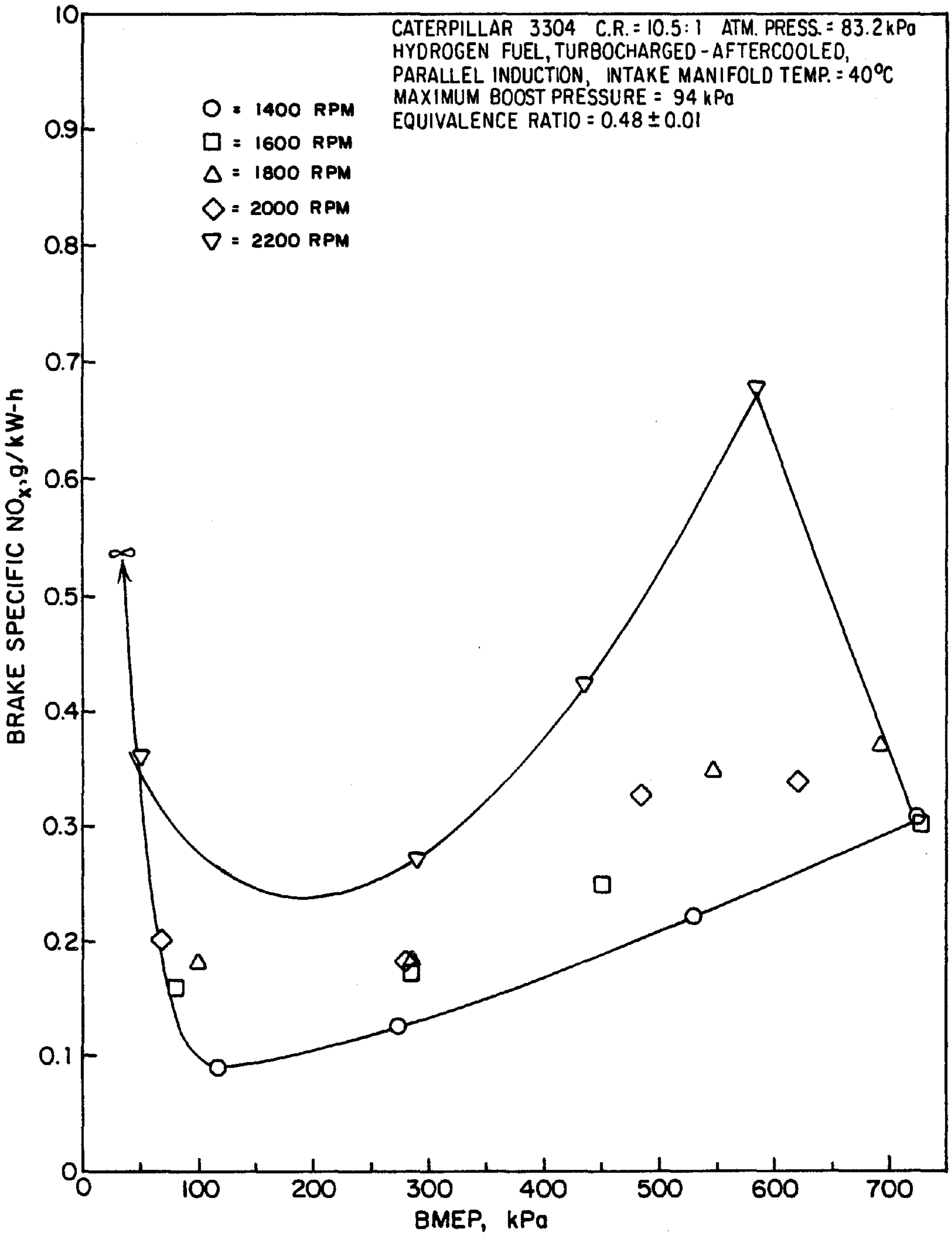


Figure 18.
Parallel Induction hydrogen engine NO_x vs BMEP data are replotted from Figure 1 on an expanded scale.

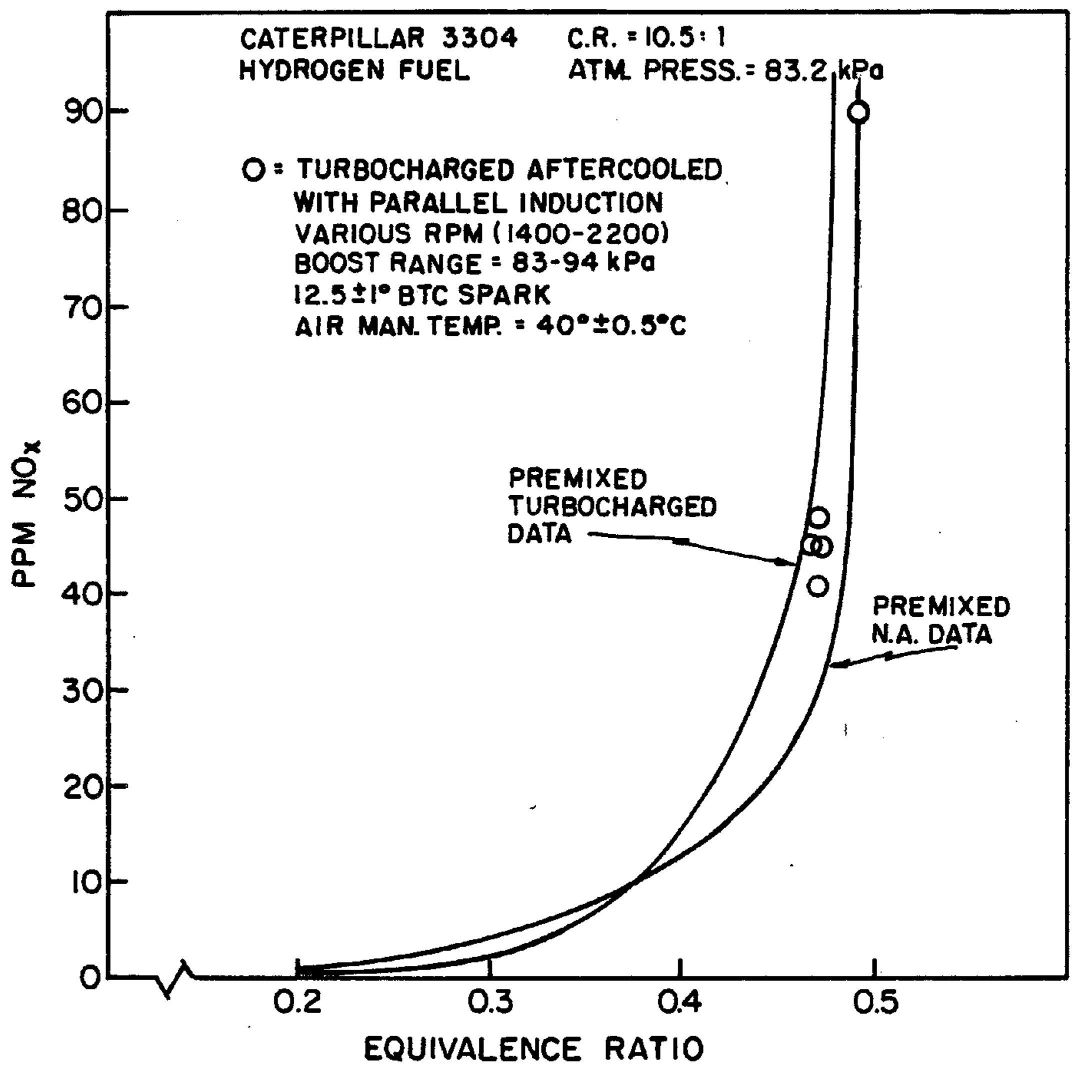


Figure 19.
NO_X concentrations vs equivalence ratio shows the high sensitivity of NO_X emissions to small changes in equivalence ratio.

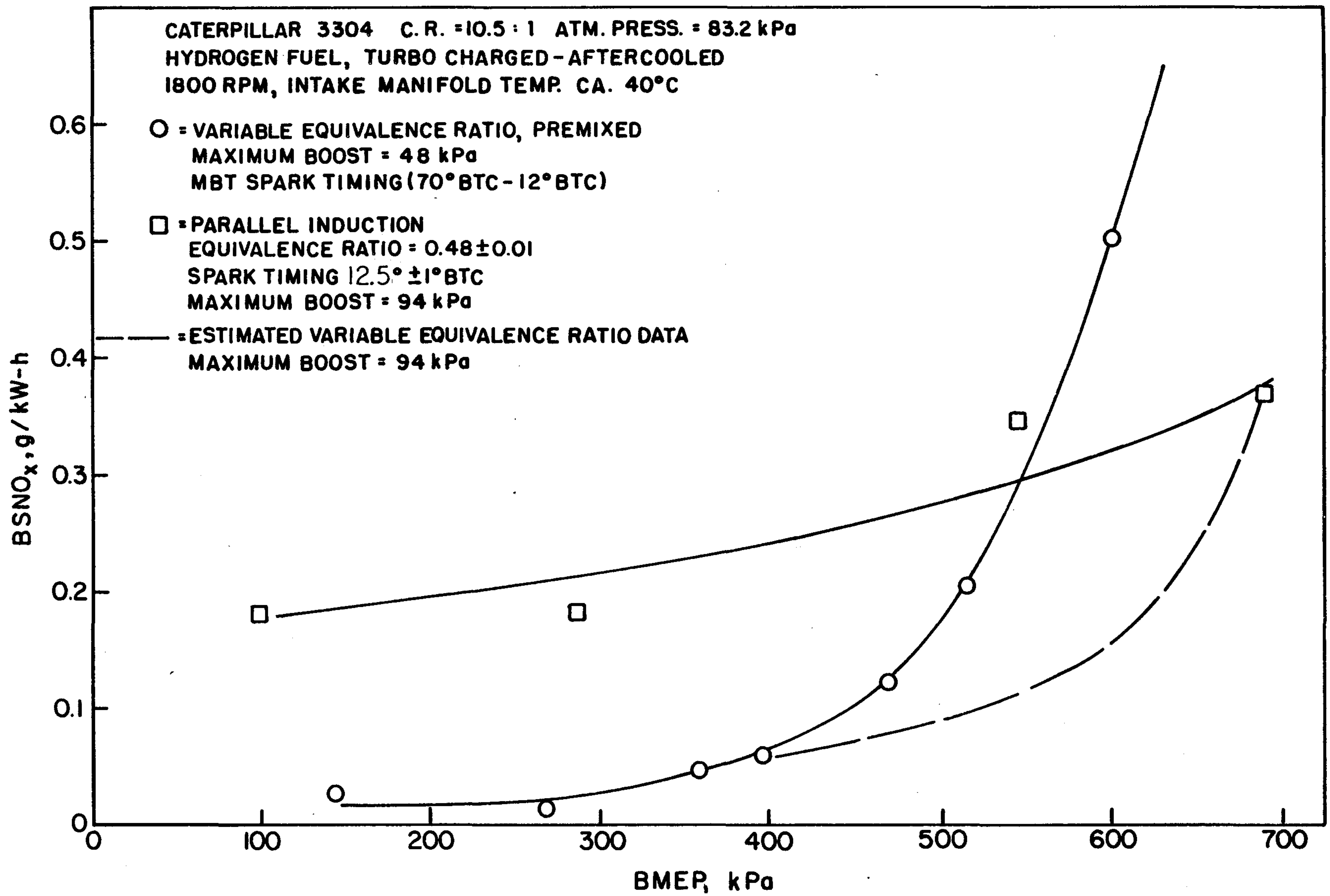


Figure 20.
Brake specific NO_x vs brake mean effective pressure for the Parallel Induction engine is somewhat higher than that which could be obtained from unthrottled hydrogen engines.

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1. F.E. Lynch, "Eimco Hydrogen Engine Project", Final Report, Eimco Mining Machinery, October 1, 1980.

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3. N. Baker, L. Huston, F. Lynch, L. Olavson, G. Sandrock, "A Clean Internal Combustion Engine for Underground Mining Machinery", Final Report, Phase I, United States Bureau of Mines Contract No. H0202034, December 31, 1981.

4. R. Adt, M.R. Swain, J.M. Pappas, "Hydrogen Engine Performance Analysis Project" - Dept. of Energy Contract No. E(04-3)-1212, 2nd Annual Rept., Jan. 1980.

5. F.E. Lynch, "Parallel Induction: A Simple Fuel Control Method for Hydrogen Engines", Proc. 4th World Hydrogen Energy Conference, Los Angeles, June, 1982.

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APPENDIX A

LUBRICATING OIL PERFORMANCE

Some of the usual engine oil degradation processes, such as particulate build-up and fuel dilution, are not possible with hydrogen fuel. On the other hand, the high moisture content of the combustion gases and the presence of NO_x could lead to corrosion of internal engine parts. Throughout the Eimco hydrogen engine development project the engine oil has been analyzed at regular intervals so that any unusual changes in the lubricant would be noted.

All testing has been performed with Chevron "Gas Engine Oil HDAX", SAE 30. This oil is specially formulated with low ash content so that deposits in the combustion chamber are minimized. When the cylinder head was removed in preparation for recent engine modifications, a brown, tar-like stain was found on each piston crown. This is undoubtedly from lubricating oil which had not burned completely as a result of the relatively low combustion temperatures in a lean burning hydrogen engine. These resinous deposits should not cause a problem as long as accumulations are modest and ash does not form. The main concern is for preignition from hot glowing ash or carbon deposits on combustion chamber surfaces. (see Figure A)

The brown stains were also found on the combustion chamber surfaces of the cylinder head and intake valves. There were some solid carbonaceous deposits near the exhaust valve seats on the cylinder head, but the exhaust valves were relatively free from deposits of any kind.

The pistons and exhaust valves were not cleaned during the recent modifications so that a continued observation of deposit accumulation may be made. However, a new cylinder head was used for the Parallel Induction modifications and new intake valves were installed in it.

The oil and filter were changed after the installation of the Parallel Induction fuel system as a precautionary measure since the engine had been out of service for over a year. However, the analysis of the used oil was not much different after the long period of inactivity. Test Report No. 81-2191 lab #4023 on 11-3-81 reflects the properties of the oil after the inactive period while lab #2617 (transcribed from an earlier report No. 80-1507) describes the oil before the inactive period. Aside from minor discrepancies in silicon and sodium levels, the wear metals have not changed. Oxidation which was incipient in 2617 was not found in the later sample but water and "fuel dilution" were. In their comments Hauser Labs notes that the apparent fuel dilution was mistakenly diagnosed because of water content. The water may simply be condensation of atmospheric moisture, since it was never noted during any active operating period.

Lubricating oil samples were taken from new, as received Chevron HDAX SAE 30, from the engine after using this oil for 10 hours and again after 20 hours (see attached report No. 82-564). This short test period is not a very strong indication of oil performance or of engine wear metal accumulation. However, since a history of lube-oil analysis is available from the 120 hour test period reported in (1) a fair understanding of engine wear and oil performance is beginning to form, and more experience will be gained during subsequent phases of the project. The only surprise in report No. 82-564 is that viscosity seems to be decreasing rather than increasing. The changes in viscosity are not large enough to cause concern but the phenomenon bears watching.

Figure A.
Stains on the piston crown after 120 hours of operation on hydrogen fuel.

HAUSER
LABORATORIES
5680 CENTRAL AVE. P.O. BOX G, BOULDER, COLORADO 80302 • PH. 303-443-4662

November 30, 1981
Test Report No. 81-2191

USED OIL ANALYSIS REPORT

Client Hydrogen Consultants, Inc.
P. O. Box 10454
Denver, CO 80210
Attention: F. Lynch

Program
Wearcheck X
Oilcheck X

Unit#
Model 3304
Make CAT
Oil Type Chevron HDAX SAE 30

										Metals in Parts Per Million							
Lab #	Sample Date	Unit Miles or Hours	Oil Miles or Hours	Viscosity SUS @ 100°F	Fuel Dilution	Water	Ethylene Glycol	Oxidation	Blotter Test	Aluminum	Silicon	Iron	Lead	Chromium	Copper	Tin	Sodium
4023	11-3-81		120	558	P?	P	N	N	A	< 5	10	37	31	<1	97	< 5	47
2617	8-8-80		120	597	N	N	N	P	A	<5	25	38	37	<1	100	<5	25

Reporting Codes: A - Acceptable, B - Borderline, E - Excessive, N - Negative, P - Present, U - Unacceptable

Comments: #4023: No previous data for comparison. Flashpoint (for fuel dilution) is below 240°F. Positive water result confused with high fuel content. Copper level may be high. Other metals appear to be acceptable.

HAUSER LABORATORIES

5680 CENTRAL AVE. P.O. BOX G, BOULDER, COLORADO 80302 • PH. 303-443-4662

April 8, 1982
Test Report No. 82-564

USED OIL ANALYSIS REPORT

Client: Hydrogen Consultants Inc.
P.O. Box 10454
Denver, CO 50210
Attn: F. Lynch

Program: Unused Oil
Wearcheck: X
Oilcheck: X

Unit#: A
Model: 3304
Make: CAT
Oil Type: HDAX SAE 30 Chevron

Metals in Parts Per Million

Lab #	Sample Date	Unit Miles or Hours	Oil Miles or Hours	Viscosity SUS @ 100°F	Fuel Dilution	Water	Ethylene Glycol	Oxidation	Blotter Test	Aluminum	Silicon	Iron	Lead	Chromium	Copper	Tin	Sodium
3427	4-3-81			656	N	N	N	N	A	<5	<10	2	3	<1	<1	<10	2
4282	2-24-82		10 hrs	627	N	N	N	N	A	<5	<10	14	6	<1	27	<5	18
4353	3-18-82		20 hrs	612	N	N	N	N	A	<5	<25	15	5	<5	26	<5	13

Reporting Codes: A - Acceptable, B - Borderline, E - Excessive, N - Negative, P - Present, U - Unacceptable

Comments: #4282: Elevations in iron, copper and sodium levels. Other metals are acceptable. Recommendatio Resample in regular interval.

#4353: Wear metals are acceptable.

APPENDIX B

TEST EQUIPMENT AND PROCEDURES

The hydrogen engine described in this study was converted and tested at the HES facility in Denver, Colorado. Within this facility are the engine/dynamometer test cell, and the fuel, air, and exhaust gas instrumentation.

Some of the values present in this study require the use of only one instrument, e.g., equivalence ratio (% oxygen) or BMEP (torque). Several important values require several instruments: Computing brake specific fuel consumption requires data from instruments measuring pressure, differential pressure, temperature, engine speed and engine torque. Associated with each instrument is its accuracy of measurement, defined as the extent to which the results from the measurement approach the true (error-free) value. A second source of error, or inaccuracy, is the reading error, which determines how closely the instrument's scale or meter can be read, a function of the number of divisions on the scale, or digits on a meter.

Measurement accuracy is reported as maximum percent deviation from the true value. Reading error is reported in the units of the instrument's scale or meter. Therefore the percentage contribution of the reading error to total inaccuracy of the measurement depends on the actual scale or meter reading for a particular data point. The contribution is higher at downscale readings than at upscale readings.

ENGINE/DYNAMOMETER TEST CELL

The test engine was coupled to a Stuska Engineering Model 800D absorbtion brake dynamometer. The principal data outputs from the dynamometer are the engine's torque and speed. By calibration, the error in the torque measurement was found to be insignificant; the reading error in the torque gauge dominates the error analysis with an uncertainty of ±0.78Nm (1.05 ft lb). The engine speed (mechanical-drive) tachometer has a scale reading error of ±10 rpm. Stuska Engineering provided a calibration curve generated against a ±0.25% reference.

The engine and dynamometer are housed with a sound-proofed cell, suitably ventilated and containing all the appropriate safeguards in case of an accidental fuel release.

Hydrogen fuel was supplied by a 1203m^3 (42,543 scf) tube trailer, located in a yard adjacent to the building. A permanent hydrogen piping system, constructed according to local safety codes, is installed between the outside wall and the engine test cell. The tube trailer hydrogen pressure is reduced to about 345 kPa (50 psig) and supplied to the interior hydrogen plumbing via a flexible fuel line. Manual shut-off valves are located at the tube trailer outlet, at the dynamometer control panel and at the site where the hydrogen plumbing enters the interior of the building.

FUEL FLOW INSTRUMENTATION

Hydrogen flow to the engine is measured with a Meriam Instruments Laminar Flow Element, Model 50MW20-2, which produces a differential pressure directly related to volume flowrate. Calculation of the mass flow requires knowledge of differential pressure, absolute pressure and temperature of the flowing gas, provided by the following instrumentation:

Differential Pressure: Dwyer model 246 inclined manometer; 0-6" water column (WC); Reading error, 0.01" water column; Accuracy, ±1%.

Absolute Pressure: Dwyer Magnehelic model 2215; 0-15 psig; Reading error, 0.1 psig; Accuracy ±2%.

Temperature: Omega Engineering Type J (Iron-Constantin) Thermocouple, SS 304 sheath; Readout: Omega Engineering model 199 Digital Meter; Overall error ±1.5°C (±0.5% at test conditions).

The calibration curve for the LFE (±0.5% accuracy) provided by Meriam was converted to a polynomial equation, for computing purposes, and this conversion produced an additional inaccuracy of 0.25%.

Early computations for hydrogen mass flow were suspiciously low compared to data obtained earlier when the engine was in its premixed or carbureted form. Several data points were checked by measurement of hydrogen pressure change in a cylinder of a known volume, over a measured time period. Since the hydrogen tube trailer was already the fuel source, it was fitted with a more accurate pressure gauge (0-600 psig), a Weksler model AA14P, which has a stated measurement accuracy of ±0.5% and a reading error of 2.0 psi. The internal volume of the trailer was certified to within ±0.1%, and the elapsed time, measured by a stop-watch, was sufficiently long to reduce timing inaccuracies to insignificance. Gas withdrawal was very slow, and therefore should approach isothermal conditions.

These tests were duplicated with a second much smaller volume, which was known to be within .06%. Pressure measurement was also more accurate (±0.25%) with the use of a Weksler model TA1 gauge, having a reading error of 0.5 psi. There was good agreement between the results from both volumes but not with the flowmeter. To compensate for the erratic hydrogen flowmeter, the air flow and exhaust oxygen data were used to deduce hydrogen flow. This gave good agreement with the volume discharge data.

Further hydrogen instrumentation, for diagnostic purposes, was also installed on the test engine. The pressure drop across the hydrogen throttle valve could be measured by three Dwyer Magnehelic differential pressure gauges with full-scale reading of 8"WC, 60"WC and 15 psi. A type J thermocouple was installed in the hydrogen intake manifold (important for volumetric efficiency calculations), and measured by the Omega Engineering model 199 digital readout described earlier. The pressure difference between the hydrogen pressure above the throttle and the stagnation air pressure was measured by a Dwyer Magnehelic model 2020 differential pressure gauge with an accuracy of ±2% and a 0.25"WC reading error.

AIR FLOW INSTRUMENTATION

Air flow to the engine is measured by a Meriam Instruments Laminar Flow Element Model 50MC2-4F. The instrumentation for measurement of differential pressure and air temperature are identical to those used for hydrogen flow, discussed above. Absolute air pressure was obtained from the U.S. Weather Service station located in Denver, and corrected for the altitude difference between the station and the engine test site. The actual pressure of the air flowing within the LFE will be lower than the atmospheric pressure from the effect of the air filter upstream of the LFE. Measurement of the air filter pressure drop showed the maximum effect on accuracy will be less than ±0.3%. Conversion of the supplied calibration curve for the LFE to a polynomial produced an additional inaccuracy of 0.2%.

Additional air flow instrumentation was installed on the test engine for both diagnostic and control purposes. Air intake manifold pressure was measured by a Weksler model BA1 compound (30"Hg to 15 psig) gauge (±1% accuracy, 0.25"Hg or 0.1 psi reading error) and temperature by Type J thermocouple and Omega model 199 digital readout. For all these tests reported in this study, air manifold temperature was set at 40^oC ($\pm 0.5^oC$) by manual control of water flowrate through the aftercooler between the turbocharger compressor and air throttle valve. Aftercooler performance data, for design of the air-cooled exchanger anticipated in subsequent work, required the measurement of turbocharger air temperature by Type J thermocouple and the Omega digital readout. Pressure drop across the air throttle valve was measured by the same gauges discussed earlier for the hydrogen throttle valve.

Certain measurements of engine blowby gas flows were made with a flow of compressed air purging the crankcase. The volume flowrate of air was determined by a Dwyer Rate-Master model RMC-121 flowmeter, 2% accuracy, 0.1 sfcm reading error. All blowby gases were ducted to the air intake system. During measurement of engine air flow, no purge air was admitted to the engine, since such flow would affect the flowrate through the LFE, which has a higher accuracy than the flowmeter measuring purge air.

EXHAUST GAS FLOWS

Information on the performance of the exhaust turbine side of the turbocharger was gained by Type J thermocouple measurement of exhaust gas temperature immediately up- and downstream of the turbine. Gas pressure upstream of the turbine (exhaust manifold backpressure) was measured with a 0-30 psi Weksler model AA13P pressure gauge, having a ±0.5% accuracy and a 0.1 psi reading error. Exhaust pressure downstream of the turbine was found to be only several inches water column.

EXHAUST ANALYSIS

Exhaust gases were extracted from the exhaust system at six locations: the four exhaust manifold cylinder flanges and up- and downstream of the turbocharger turbine. The individual cylinder exhaust sampling was only used for the cylinder-to-cylinder mixture distribution tests, while the majority of data were taken at the sampling point downstream of the turbine. Extended testing of the backpressure gases upstream of the turbine tended to over-pressurize and overheat the gas sample handling system.

The exhaust gas sampling points are connected to a series of Snap-Tite quick-connects by 0.25" OD SS 304 tubing, as shown in Figure B-1. Individual samples are made by connecting the flexible SS 316 and Teflon TFE line to the Snap-Tites. Exhaust gas then flows into the Balston Type 33G filter housing (all stainless and Pyrex glass) containing a type CX filter (fluoro-carbon binder, 95% collection efficiency at 0.6 microns). The filter housing also served as a first-stage water trap.

The filtered gases exiting the filter are drawn into the all stainless and Teflon conditioning system by a Metal Bellows Type MB-141 pump. The gas is pressurized to 6 psig and chilled in the shell-side of a Young SSFO-201-HY-1P stainless steel heat exchanger. Cooling is provided by a recirculating loop of a refrigerated antifreeze/water solution. Entrained water condensate in the exhaust drops out in the trap at the heat exchanger outlet. The exhaust gas then enters a series of valves and manifolds for subsequent flow to the gas analyzers.

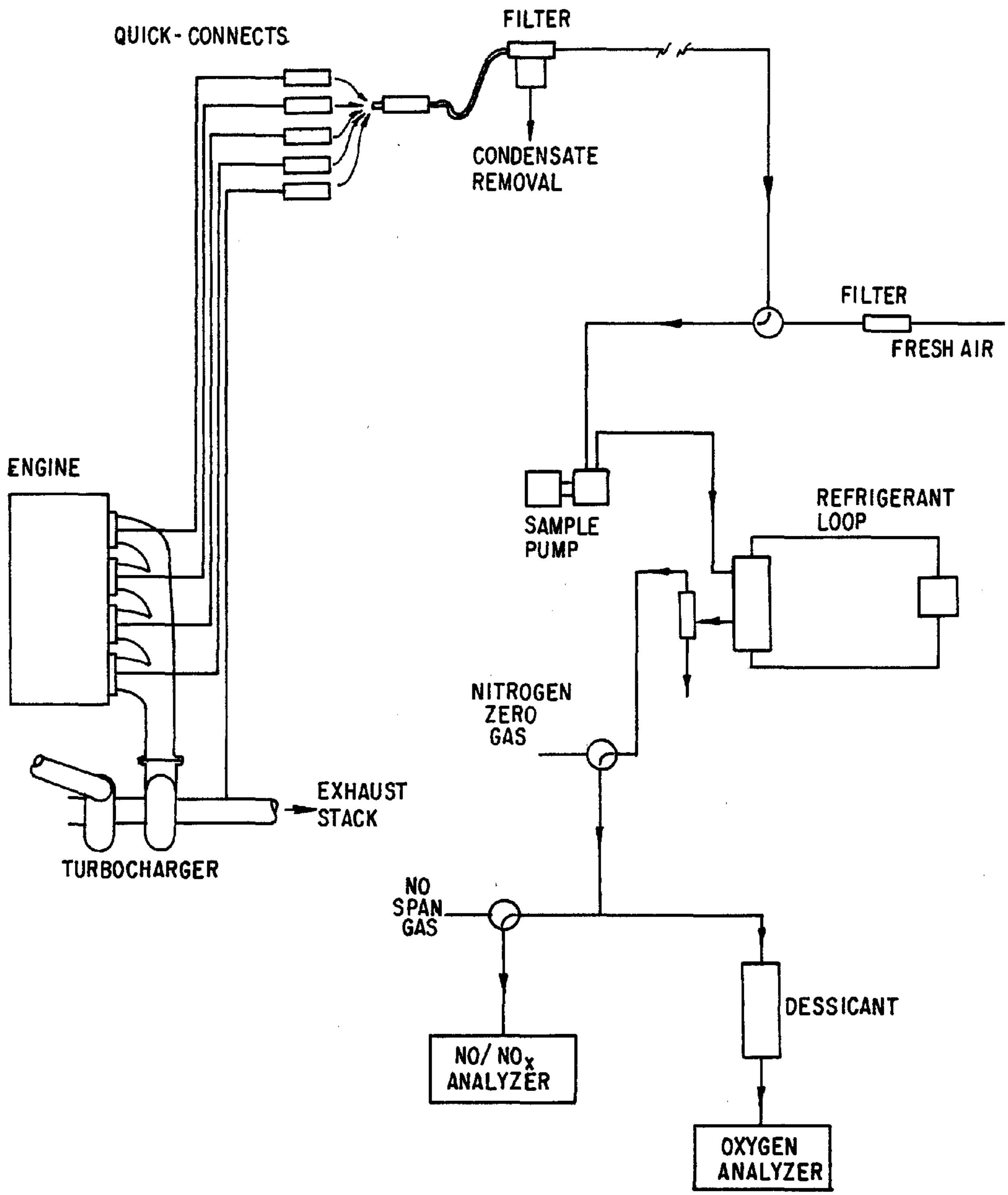

Figure B-1.
Exhaust gas sampling system.

OXYGEN ANALYSIS

The most important piece of instrumentation used in tuning a hydrogen engine is the oxygen analyzer, since the dry exhaust oxygen content (expressed as a fraction) is directly relatable to equivalence ratio (ø) by the equation

$$ø = \frac{0.20946 - O_2}{0.20946\ (1-O_2)}$$

The Beckman Instruments OM-11EA oxygen analyszer utilized in this study has an overall measurement and reading accuracy of ±1% of full scale (25% O_2) or ± .25% O_2. Exhaust gas was dried to a -40°C (-40°F) dew point by passing the gas through a column of Drierite dessicant. The instrument was calibrated before and after each series of tests with dry zero-grade nitrogen (containing 3 ppm oxygen) and dry outside air (20.946% oxygen). If instrument drift exceeded 0.1% oxygen, the data for that series of runs were eliminated from the study.

OXIDES OF NITROGEN ANALYSIS

The only significant emissions from a hydrogen engine are the total oxides of nitrogen (NO_x), comprising nitric oxide (NO) and nitrogen dioxide (NO_2).

These gases were measured with a Thermo Electron Model 10A Chemiluminescent NO-NO_x analyzer. Since the instrument has a linear response, the unit can be calibrated on any of the selectable full-scale meter ranges of 0-2.5, 10, 25, 100, 250, 1000 ppm NO/NO_x, and data taken on any other range. The analyzer has a stated accuracy of ±1%, and a reading error of 0.01 full scale units. The instrument was zeroed and spanned with 84 ppm NO in nitrogen before and after each series of runs. Instrument drift was insignificant.

Because NO_2 is an important mine air component from a health standpoint, measurement of this gas requires special discussion. NO_2, unlike NO, is very soluble in liquid water, forming nitric acid. Since any NO_2 lost from the exhaust sample via water condensation removal would compromise the emission data, a series of tests were run to determine this effect. The engine was run under conditions which produced about one hundred ppm NO and NO_2 when sampled using the normal methods, including condensate removal. The entire sample handling system was then isolated and dried, with the exception of the condensate trap, which was bypassed since drying this unit would have required many hours. Air was injected at the exhaust sampling point to lower the dew point and the sample handling pumps were activated. The oxygen, NO and NO_2 concentration in this condensate-free sample were then measured. Comparison of the dilution ratios determined from the oxygen, NO and NO_2 measurements showed good agreement (within 4%). Had the dilution ratio based on the NO_2 measurement been different from the oxygen or NO-based ratio, the implication would be that NO_2 is lost by reaction with liquid water, when the normal sample handling system is employed. Since the opposite was found, the NO_2 data are not compromised. This may not be true for exhaust streams containing high concentrations, but it appears that the NO_x concentrations typically produced by the hydrogen engine (less than 100 ppm) are unaffected by this water condensation and removal system.

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APPENDIX C

CRANKCASE BLOW-BY ANALYSIS

The blow-by gas composition is of special interest in a hydrogen fueled engine for underground mining, since the crankcase contains a respectable volume of oil of a known fire hazard potential, regardless of the fuel used. When the engine is running, the crankcase contains a mixture of air, exhaust gases, intake gases and oil mist. There are three conceivable problems which must be considered:

- Excess moisture from exhaust gases can emulsify the engine oil and lead to excessive engine wear.
- Exhaust gases contain NO_x which can lead to corrosion by combining with moisture to form acids.
- Combustible concentrations of hydrogen in the crankcase gases are unacceptable because of possible crankcase explosions.

The engine was operated at 1400 rpm and full load while the crankcase gases were analyzed for oxygen and NO_x using the exhaust gas analyzers. Tests were conducted while various amounts of purge air where blown through the crankcase. By knowing the oxygen content of the pure crankcase gases and with vartious measured flows of dilution air (20.95% O_2) the blow-by flow was deduced. This flow was 1.75 l/sec. This is less than 2% of the intake flow.

The proportion of the blow-by which has the composition of exhaust gas was determined to be roughly 25% (see later discussion). In this diluted state the dew point of the blow-by should be 30-35^oC, which is well below the normal operating temperature of the crankcase. Water condensation is not a problem as further shown by oil analyses.

The NO_x concentrations observed were on the order of 10 ppm in undiluted crankcase gases. Such low concentrations cannot cause significant engine corrosion.

Two bag-samples of crankcase gases were anlyzed by gas chromatography. The data are listed below in Table C along with the sampling conditions.

TABLE C - GAS CHROMATOGRAPH ANALYSES OF BLOW-BY GASES

	Sample A - 1400 rpm - 360 kPa BMEP
71%	N_2
16%	O_2
13%	H_2
	3000 ppm Total HC

	Sample B - 1400 rpm - 727 kPa BMEP
72%	N_2
16%	O_2
11%	H_2
	2600 ppm Total HC

The relative concentrations suggest a blow-by composition of roughly 25% exhaust gases and 75% intake gases. The low fraction of exhaust gases in the blow-by is confirmed by the real-time oxygen and NO_x analyses.

These hydrogen-oxygen concentrations are clearly within the combustible range. The vapor phase hydrocarbon concentrations are too low to add much to the problem but if there were a crankcase explosion, oil mist would probably contribute to the intensity. The largest fire hazard is, of course, the 20 liters of oil in the crankcase which could be released if the oil pan were damaged by an explosion.

This hydrogen can come from only two places: the piston rings leaking fuel-air mixture during compression or the intake valve guide leaking pure hydrogen from the sleeve-valve. As mentioned above, the proportion of exhaust gas in the crankcase is rather low. Therefore the majority of the leakage takes place during intake and compression. Balancing the oxygen and hydrogen content of the blow-by gases against the intake and exhaust compositions indicates that the major source of hydrogen is the piston rings rather than the valve guides.

It was an objective of this develoment phase to determine what problems may exist in the crankcase of a hydrogen engine. Although no other problems were found, the hydrogen content of the blow-by gases is clearly something which must be dealt with in future phases of the program. There is at least one solution which is apparent at this early stage of considering the problem. The blow-by flow rate is a small fraction of the air flowing to the engine if the engine is in a reasonable state of repair. By passing some of the engine's air flow through the crankcase, the blow-by gases can be diluted well below combustible hydrogen concentrations.

The present 1400 rpm blow-by rate of 1.75 l/sec at 12% hydrogen could be diluted below the combustible limit by drawing about 3 l/sec of fresh air through the crankcase. At 1400 rpm the engine is drawing about 85 l/sec of air, so 3.5% of the total air flow is needed.

As the engine wears, the blow-by rate will increase, so the dilution air flow should be high enough to render all reasonable blow-by rates non-combustible.

APPENDIX D

COMPUTER MODELING OF PARALLEL INDUCTION MIXTURE CONTROL

Before proceeding to the design of actual hardware for the Parallel Induction system on the Caterpillar 3304, an idealized, steady flow model of the system was computer simulated. The model was based on an approximate flow equation which is a proposed standard of the National Fluid Power Association:

$$Q = 22.5\, C_v \sqrt{\frac{\Delta P\, (P_2)}{T_1\, (G)}}$$

Where: Q is the flow in scfm
ΔP is the pressure drop in psi differential
T_1 is the upstream temperature in ^oR
P_2 is the downstream pressure in psia
G is the specific gravity (air = 1.0, H_2 = 0.0695)

The flow coefficient, C_v is 1.0 for a valve which flows 1 gpm of water with a 1 psi pressure drop. Of more use in designing Parallel Induction components is the "equivalent orifice" relationship, $C_v = 18\, D_o^2$ where D_o is the diameter of a sharp-edge orifice.

It has been found that the butterfly throttles in a Parallel Induction system can be suitably approximated by sharp-edged orifices whose areas are the same as the area of the cylindrical throttle bore minus the throttle shaft cross sectional area, as viewed from the axis of the throttle bore.

The modeling was only performed for wide-open throttle conditions, since the objective was to establish the minimum size throttles which could be used without excessive restriction of the intake system. The criterion for this was arbitrarily selected as 0.4 psi (about 3 kPa) pressure drop at the maximum engine speed, before the turbocharger began to pressurize the intake system. This speed was estimated from experience gained on other engines to be 880 rpm for the 3304 Caterpillar.

The "minimum equivalence ratio" (defined as the equivalence ratio at constant temperature) was set at a value somewhat lower than the target mixture so that the effects of turbocharger heating would be compensated for. This heating was included in the flow equation via the T_1 term by an empirical equation derived for the air-to-air aftercooler on one of HES' hydrogen vehicles.

Pressure rise vs rpm was likewise an empirical function patterned after actual test data on another engine using a turbocharger similar to the one used on the 3304 Caterpillar.

The program first calculated total intake flow in acfm at standard pressure and temperature for the 880 rpm, unturbocharged condition. Next, the hydrogen and air flows were determined as fractions of the total flow and the required C_v for each flow was determined so that the pressure drop was at the desired value.

The equivalent orifice sizes for each valve were then calculated and then the program begins increasing rpm with corresponding boost pressure and air temperature increases. At each rpm interval the resulting equivalence ratio was calculated.

A typical printout for the program is shown below.

PARALLEL INDUCTION HYDROGEN ENGINE
THROTTLE SIZING
AND
MIXTURE CONTROL DATA

THE FOLLOWING INPUT DATA WERE SUPPLIED:

ENGINE DISPLACEMENT	=	425 CUBIC INCHES
ATMOSPHERIC PRESSURE	=	14.7 PSI
AMBIENT TEMPERATURE	=	530^{o}R
VOLUMETRIC EFFICIENCY	=	87%
MINIMUM EQUIVALENCE RATIO	=	43%
RPM WHERE BOOST BEGINS	=	880 RPM
MAXIMUM ENGINE SPEED	=	2200 RPM
H2 OFFSET PRESSURE	=	2 INCHES H20

AIR FLOW BEFORE BOOST	=	79.6790463 SCFM
H2 FLOW BEFORE BOOST	=	14.3557738 SCFM
AIR SYSTEM Cv	=	36.8210794
H2 SYSTEM Cv	=	1.75139022
EQUIVALENT AIR ORIFICE	=	1.43025016 INCHES
EQUIVALENT H2 ORIFICE	=	.311928608 INCHES

ENGINE RPM	EQUIVALENCE RATIO	AIR TEMP	AIR PRESS
220	.9	530.	14.69
440	.58	530.	14.69
660	.5	530.	14.69
880	.47	530.	14.69
1100	.45	535.98	18.38
1320	.47	571.11	22.06
1540	.52	626.4	24.7
1760	.51	626.4	24.7
1980	.51	626.4	24.7
2200	.51	626.4	24.7