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HYDROGEN ENGINE PERFORMANCE ANALYSIS

First Annual Report

By
Robert R. Adt, Jr.
Michael R. Swain
John M. Pappas

August 1978

Work Performed Under Contract No. EC-77-C-03-1212

University of Miami
Department of Mechanical Engineering
Coral Gables, Florida

U. S. DEPARTMENT OF ENERGY

Division of Transportation Energy Conservation

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FOREWORD

In the Spring of 1976, the Energy Research and Development Administration, ERDA, (now the Department of Energy, DOE) distributed RFP No. E(04-3)-1212 - REQUEST FOR PROPOSALS - HYDROGEN ENGINE PERFORMANCE ANALYSIS PROJECT.

Background

According to the RFP: "In view of the fact that domestic petroleum will continue to be an increasingly scarce commodity, and that expanded reliance on imported petroleum by the United States is not feasible, alternatives to petroleum must be found and deployed. Transportation is almost wholly based on petroleum derived fuels at the present time.

Studies made for the Energy Research and Development Administration on alternative fuels for highway vehicles identify synthetic gasoline and synthetic distillate from coal and from oil shale and methanol from coal as the probable fuels for the 1985-2000 time period and beyond. Hydrogen was identified as another long-range alternative with timing primarily dependent on abundant availability from non-petroleum sources at competitive prices. It is possible that hydrogen may find earlier usage in limited or special areas where low emissions or other factors are paramount.

In contemplating hydrogen's use as a fuel for highway vehicles, two problem areas are evident which require early technical research and development: (1) on-board storage means and (2) power system design and operation.

This procurement focuses on the hydrogen-fueled power system. More specifically, it is limited in scope to the hydrogen/air-breathing internal combustion engines of the conventional reciprocating piston type.

At this point, a significant data base on engines has been developed by both 'historical' and contemporary researchers. However, it is observed that fully comprehensive, accurate and consistent technical data for the hydrogen engine remains to be achieved.

Hence, the required design-data baseline necessary to enable future internal combustion piston engines to be engineered for transportation application is at present incompletely documented.

Objective

ERDA desires to have the necessary experimental hydrogen piston-engine research conducted to provide the data needs for possible future engine development projects. A number of alternative power system types capable of operating on hydrogen-energy may be addressed in future programs. However, since the internal combustion piston engine is far and away the preeminent transportation power system today, it will be given initial attention.

The output of the project will be the published documentation of a substantial and coherent design data-base covering Performance, operational characteristics, and emissions of hydrogen/airbreathing, piston, reciprocating-internal combustion engines. This documented data-base should be sufficient to permit engine designers to directly execute practical prototype hydrogen engine designs.

The design, assembly and testing of one or more automotive type 4-cylinder, water-cooled, piston engines is viewed as the principal basis for the technical work to be conducted under the contract contemplated by this procurement action.

Conventional gaseous hydrogen supply systems will suffice for supporting the engine research, since the development of practical on-board storage means is outside the scope of the project.

Qualifications Criteria

Offerors submitting proposals in response to this RFP should, as a minimum, be experienced in modification of automotive type engines for use of hydrogen fired in testing thereof."

Contract Award

Based on the proposal, prepared as a joint effort by the authors of this document, the University of Miami, with Hawthorne Research and Testing, Inc. acting as subcontractor, was awarded the cost reimbursement contract to perform the HYDROGEN ENGINE PERFORMANCE ANALYSIS PROJECT. The project is expected to take three years to complete.

Present Work

In the present project, hydrogen-fueled engines are grouped into two major classifications: Pre Intake Valve Closing (Pre IVC)

Engines in which the fuel is introduced into the cylinder prior to the closing of the intake valve and Post Intake Valve Closing (Post IVC) Engines in which the fuel is introduced into the cylinder after the intake valve has closed. Thus, with the Pre IVC engine, the hydrogen is at a relatively low pressure when it is ingested by the engine, with the Post IVC engine the hydrogen is at a high pressure.

Within each of these two major classifications a number of configurations have been identified including 10 Pre IVC and 4 Post IVC engine types. They are delineated in this report. Following this delineation, each of the 10 Pre and 4 Post IVC engines are reviewed and evaluated with respect to their actual and/or potential performance, emissions and practical-use characteristics. In general, Pre IVC engines are found to be preferable over the Post IVC type for automotive applications.

This delineation, review, and evaluation was conducted by the authors during the planning stages of the project. It is presented because it was instrumental in the authors' formulation of the present-study program and should therefore aid the reader in understanding the rationale behind the particular course of action taken by the authors.

During the course of this three year program, all 14 (10 Pre- and 4 Post-IVC) configurations will be subjected to performance and emissions characterization tests. During the first year, baseline data for throttled and unthrottled, carburetted, Pre IVC hydrogen-fueled engine configurations, with and without exhaust gas recirculation (EGR) and water injection were obtained. In this report, these data along with descriptions of the test engine and its components, the test apparatus, experimental techniques, experiments performed, and the results obtained are given. Analyses of other hydrogen-engine project data are also presented and compared with the results of the present effort.

The unthrottled engine vis -à-vis the throttled engine is found, in general, to exhibit higher brake thermal efficiency. The unthrottled engine also yields lower NO_x emissions which were found to be a strong function of fuel-air equivalence ratio.

The presence of hot spots and operation with lean mixtures was found to cause flashback; hot spot-induced flashback was

characterized. Large valve overlap was found not to increase flashback tendencies.

Peak-power-output calculations showed that the potential, non flashback-limited power output of Pre IVC engines is larger than that usually reported if advantage is taken of the properties of hydrogen as a fuel.

Two independent methods were used to generate indicated thermal efficiency from brake thermal efficiency measurements giving results consistent with each other and allowing for the elimination of much of the apparent inconsistency of thermal efficiency results as reported by others.

Details of the above-mentioned findings and others are given in the report; the busy reader is referred to Section 13 for a brief summary of findings and conclusions.

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1. PROJECT OVERVIEW

Many problems associated with the design and development of hydrogen - air breathing internal combustion engines for automotive applications have been identified by various domestic and foreign researchers. The domestic work through circa August, 1975 has been reviewed and summarized by Escher (1975). Listed in the Reference section of this report are foreign as well as more recent domestic publications concerned with hydrogen-fueled engines for automotive applications.

An objective of this project is to address the problems identified in the literature, as well as others described herein, and to evaluate potential solutions to these problems. Another objective is to obtain and document a design data-base covering the performance, operational and emissions characteristics essential for making rational decisions regarding the selection and design of prototype hydrogen-fueled, airbreathing engines suitable for manufacture for general automotive use.

The project is expected to take three years to complete. The project sponsor is the Department of Energy (DOE; formerly the Energy Research and Development Administration). Contractual and business matters affecting the project are handled by the DOE San Francisco Operations Office, 1333 Broadway, Oakland, California, 94612. Technical considerations are handled by the Division of Transportation Energy Conservation, 20 Massachusetts Avenue, N.W., Washington, D.C., 20545 [E. E. Ecklund, (202-376-4892)] with Escher Technology Associates, P.O. Box 189, St. Johns, Michigan, 48879 [W. J. D. Escher (517-224-6726)] acting as project Monitor on behalf of DOE.

The contractor is the University of Miami, Department of Mechanical Engineering, Coral Gables, Florida 33124, [R. R. Adt, Jr., (305-284-4855)] and the subcontractor is Hawthorne Research and Testing, Inc., P.O. Box 248203, Coral Gables, Florida 33124 [M. R. Swain, J. M. Pappas].

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2. PRESENT REPORT

This document is the first annual report of the three-year project. It commences with a review of operational performance and emissions characteristics of hydrogen-fueled engines, followed by a delineation of potential hydrogen engine configurations and an evaluation of them. This review, delineation, and evaluation was conducted by the authors during the planning stages of the project. It is presented here because it was instrumental in the authors' formulation of the present-study program and should therefore aid the reader in understanding the rationale behind the particular course of action taken by the authors.

Following this background review is a reporting of the first year's project activities including descriptions of the test engine and its components, the test apparatus, experimental techniques, experiments performed, and the results obtained. Analyses of other hydrogen-engine project data are also presented and compared with the results of the present effort.

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3. HYDROGEN ENGINE CHARACTERISTICS - A BRIEF REVIEW

3.1 PRE- AND POST-INTAKE VALVE CLOSING (IVC) ENGINES

A wide variety of airbreathing piston engine types are potential candidates for use with hydrogen fuel. Two basic classes of engines have been selected with respect to the potential of hydrogen-fuel useage. The first involves ingestion of hydrogen into the cylinder prior to closing of the intake valve; the second after closing of the intake valve. In this report, the term Pre Intake Valve Closing (abbreviated Pre IVC) is used to denote the former engine type and Post Intake Valve Closing, (Post IVC), to denote the latter.

Hydrogen-fueled engine types have been summarized and described in detail by Escher (1975): The Pre IVC engines are similar in operation to conventional gasoline-fueled engines in that both hydrogen* and air** are introduced into the cylinder during the intake stroke. They can be introduced as a mixture, the mixture being formed by a carburetor-like device (a gas-mixer), or they can be introduced separately, as in a fuel-injection gasoline engine, with the hydrogen being injected (inducted) in the intake manifold or in the vicinity of the intake valve. The injection can be continuous or timed. With the Pre IVC engines, spark-plug ignition is used. In addition to using the conventional throttle to control charge density, Pre IVC engines have been operated with the load being controlled by changing the fuel-air ratio. This method of operation termed unthrottled or quality-controlled mode of operation is made possible by the high flame speed of hydrogen over a wide range of fuel-air ratio. Post IVC engines are similar in operation to compression-ignition engines, but some positive source of controlled combustion initiation is needed such as spark plug (single or multiple spark discharge) or a glow plug. Gaseous hydrogen is injected directly into the cylinder, commencing during

* In engine applications to date, hydrogen is introduced while in the gas phase. The present discussion and work is limited to gaseous hydrogen fuel.

** Oxygen has also been used in Pre IVC engines; the present discussion and work is limited to the use of air.

or near the end of the compression stroke and continuing on into the expansion stroke. Mechanical, electrical and hydraulic-pulse actuated hydrogen injectors have been used.

In addition to the usual performance and emissions characteristics of concern in a gasoline-fueled internal combustion engine, automotive-type engines exhibit a number of characteristics which are uniquely affected by the use of hydrogen as fuel. In the following Sections these characteristics of concern to hydrogen-fueled engines are discussed.

3.2 OXIDES OF NITROGEN EMISSIONS

Hydrogen-fueled engine oxides of nitrogen (NO_x) emissions have been identified as a definite problem area of concern. NO_x emissions are known to be mainly dependent on the fuel-air equivalence ratio (ϕ)*. This dependence is illustrated in Figure 1 where typical trends in NO_x emissions are plotted versus ϕ at constant engine speed. The solid curve in the figure was determined by analysis of the chemical kinetics in a homogeneous mixture of hydrogen and air (deBoer, July, 1975). It is observed that the NO_x levels are low for lean ($\phi < 0.6$) and rich ($\phi > 1.1$) mixtures with maximum levels occurring for ϕ near 0.8. Experimentally-determined NO_x emissions are found to be in qualitative agreement with theoretical predictions showing similar trends. It is thus apparent that if nitric oxides are to be reduced without resorting to: (1) exhaust gas recirculation; (2) water injection; (3) excessive spark-advance retard; or (4) exhaust-gas reduction catalysts; some method of engine operation must be found which will allow for operation away from $0.5 < \phi < 1.1$, or perhaps, for operation with a stratified-charge fuel-air mixture with some means of internal quenching of the NO_x -formation reactions.

3.3 INDUCTION SYSTEM FLASHBACK

Flashback in the induction system occurs when the fresh hydrogen-air charge ignites prematurely, prior to the closing of the intake valve. The flame thus generated propagates through the fuel-air induction system. The suspected causes of early ignition associated with flashback are: (1) hot spots in the

* See Section 16 for Glossary of Terms.

combustion chamber; (2) particulate matter in the residuals from the oil; (3) communication of fresh charge with burning exhaust gases during valve overlap; (4) communication of fresh charge with burning exhaust gases from another cylinder during operation with misfire; (5) very lean operation with the presence of still-burning gases from the previous cycle when the intake valve opens (due to the low flame speeds at lean operation). These causes are individually discussed below.

(1) At lean equivalence ratios, flashback is inhibited by the reduction in temperature of hot spots, increases in ignition energy of the mixture and reductions in flame speeds thus facilitating quenching. If typical gasoline-fueled engine heat-range spark plugs are used, as the load is increased from idle, flashback commences. Lower operating-range temperature spark plugs can be used to alleviate this cause of hot spot. However, as the equivalence ratio is increased further the exhaust valve eventually attains a high-enough temperature to cause preignition and flashback. Sodium-filled exhaust valves, whose peak temperatures are lower by about 200°F (111°C), can be used to extend the range of operation. Further extension of the range can be accomplished by many methods such as the use of two smaller-diameter, sodium-filled valves. Testing of water-filled exhaust valves (Danis, 1973), which utilize the enthalpy of vaporization of water to increase heat transfer rates, has shown an additional 300°F (166°C) reduction in peak exhaust-valve temperatures. Since the peak temperatures for water-filled valves are found to be near the seat, improved seat design will reduce peak temperatures even further (about 50°F). Lithium-filled exhaust valves exhibit temperature characteristics similar to water-filled ones (Danis, 1977).

(2) The amount of oil-generated particulate matter in the residual gas of a hydrogen-fueled engine in good condition is expected to be minimal. But in older, worn engines, which burn oil, troublesome amounts may be present.

(3,4) Flashback suspected of being caused by communication of unburned fuel with burning fuel during valve overlap may exist when cams designed to increase volumetric efficiency at high

engine speeds are employed or when flame-initiation-failure misfiring occurs.

(5) With the unthrottled engine operated at idle, the mixture is very lean and flame speeds are low. Such operation can result in a still-burning mixture in the combustion chamber when the intake valve opens causing flashback.

3.4 IRREGULAR COMBUSTION

Irregular combustion occurs whenever a flame is initiated at an improper point in time or space. The previously-described flashback is a special case of irregular combustion in which ignition of the mixture is initiated during the intake stroke. In addition to preignition, hydrogen-air mixture nonuniformity within the combustion chamber also has an effect on irregular combustion, especially in Post IVC engines. This irregular combustion is reflected by abnormal variations in cylinder pressure traces (Escher, 1975). The characteristics of irregular combustion in a Post IVC engine are not well understood (See 3.5).

3.5 THERMAL EFFICIENCY

It is accepted that when hydrogen is used as an automotive-type engine fuel high efficiencies are obtainable if the engine is designed to take advantage of the characteristics of hydrogen (Escher, 1975): The characteristics leading to high-efficiency engine design are high flame speed, even at lean air-fuel ratios, and low flammability limit. In order to take advantage of these characteristics the design of a hydrogen-fueled airbreathing engine should utilize unthrottled operation for light-load conditions*. This method of control reduces throttling losses, and permits operation with relatively low specific heats of the working fluids in the combustion chamber. Unthrottled operation is made possible by the high flame speeds and the low flammability limit of hydrogen-air mixtures which allow for operation with ultralean mixtures. The extremely high flame speeds experienced at richer air-fuel ratios can be used to advantage to reduce burning time and heat losses thus offsetting somewhat the disadvantage of higher specific heats only if the combustion process

* It is found, however, that some throttling is desirable at very light loads to limit extremely lean operation which is disadvantageous as will be discussed.

can be controlled to prevent flashback, preignition, rough ignition, incomplete combustion, and nitric oxide formation. If these phenomena cannot be controlled, however, the high flame speeds lead to decreasing thermal efficiency, particularly in Post IVC engines. For Post IVC engines it is suspected that the low thermal efficiencies observed are due to non-uniform combustion occurring when a flame front travels from a lean to a rich region in a mixture with a large hydrogen concentration gradient: the flame front may progress so rapidly so as to not allow sufficient diffusion of oxygen thereby resulting in extinction of the flame when the rich limit of combustion is reached; if, after further diffusion, the mixture reaches a combustible concentration and is ignited again, rough combustion and poor efficiency will result. Further investigation is needed to understand Post IVC engine operation.

3.6 PEAK POWER OUTPUT

There is much debate in the literature regarding power output from hydrogen-fueled engines as compared to gasoline-fueled engines. Usually 100% volumetric efficiency* is assumed for gasoline engines. In actuality, however, the volumetric efficiency for gasoline-fueled engines is less than 100%. Its value depends on engine speed during wide open throttle operation, being equal to about 75% at the maximum horsepower RPM and about 95% at the maximum torque RPM (Campbell, 1964). These values are identical for hydrogen-fueled engines only if the gasoline-fueled engine carburetor, intake manifold, cylinder head and cam design configurations are used.

In the case of a gasoline-fueled engine the fuel takes up a very small fraction of the volume of the fresh charge** and so the amount of air ingested per intake stroke is essentially proportional to the volumetric efficiency. For Pre IVC hydrogen engines, however, the fuel can take up a significant portion of

* Volumetric efficiency being defined in terms of the ratio of actual mass of air inducted in an engine during the intake stroke to the mass of air equal to the displacement volume times the density of atmospheric air (See Obert, 1973).

** 1.7% by volume maximum: for fully-vaporized, stoichiometric C_8H_{18} and air mixture.

the intake manifold thereby displacing an equally significant amount of air. For example, at stoichiometric conditions about 30% of the air is displaced. Thus, since the amount of air ingested per intake stroke at maximum-power conditions in a gasoline-fueled engine converted to operate on hydrogen is only 70% of that for gasoline-fueled operation, there would be a $[100 - 70(1500 \text{ BTU/lb}_{\text{air}})_{\text{H}_2} / (1300 \text{ BTU/lb}_{\text{air}})_{\text{gasoline}}]\% = 19.2\%$

reduction in power output. In performing this calculation it was assumed that the volumetric efficiency of the engine when fueled with hydrogen was the same as that when fueled with gasoline. If this assumption is true then full advantage was not taken of the characteristics of hydrogen as discussed below.

When the carburetor, intake manifold, cylinder head and cam were designed for gasoline-fueled operation, a design constraint was to provide for good atomization of a liquid fuel at low engine speed. This necessitated using small passageways to maintain high velocities of the fresh charge in the intake manifold resulting in large pressure drop and low volumetric efficiency at high engine speed. Since the above-mentioned design constraint can be relaxed for gaseous-fueled hydrogen engines, the intake manifold, cylinder head and cam can be re-designed to yield high volumetric efficiency at all engine speeds, especially at the gasoline-fueled-engine peak-power RPM.

In order to give reasonable estimates of the expected power output at both peak-horsepower RPM and peak-torque RPM, calculations were performed taking into account anticipated gains in volumetric efficiency for engines incorporating intake manifolds, cylinder heads and cams specifically designed for hydrogen-fueled operation. The calculations are given in Appendix 1 (and discussed in Section 4.3.3). It should be pointed out here that the Post IVC engine cylinder head design constraints are more stringent than those of a Pre IVC engine. In Appendix 1, however, the additional constraints imposed on the Post IVC engine are ignored. It should also be pointed out that a Post IVC hydrogen engine, if it can be operated with stoichiometric mixtures, will exhibit peak power levels about 30% greater than that of a similar-size gasoline engine: The expression

$$[(1.13) (1500 \text{ BTU/lb}_{\text{air}})_{\text{H}_2} / (1300 \text{ BTU/lb}_{\text{air}})_{\text{gasoline}} - 1] 100 = 31\%$$

is more exact where the factor 1.13 accounts for the greater volumetric efficiency of the Post IVC hydrogen engine compared to that of a gasoline engine.

3.7 HETEROGENEOUS MIXTURES - COMBUSTION CHAMBER

In previously tested Post IVC engines, combustion-chamber mixture nonuniformity is suspected of causing problems resulting in low levels of maximum power output and reduced efficiency (Homan, 1978).

In the Pre IVC engine, however, nonuniformities (stratification of the charge) can lead to improved operation. A hydrogen concentration gradient can be established, with rich, easily ignited mixtures near the spark plug and lean mixtures away from the plug. Such a stratification allows operation at very lean mixtures in the Pre IVC engine. In (Swain, 1973), a fuel-air equivalence ratio of 0.09 was achieved which is well below the usually accepted lean limit of combustion in hydrogen engines of around 0.20 (King, et al, 1955). It is also lower than the upward-propagating-flame lean limit of 0.1 (Strehlow, 1968). In addition to allowing operation at very lean mixtures, charge stratification in the Pre IVC engine can reduce the concentration of hydrogen in the blowby gases alleviating a potential safety problem of excessive hydrogen in the crankcase.

3.8 HETEROGENEOUS MIXTURES - INTAKE MANIFOLD

Cylinder-to-cylinder fuel-air ratio maldistribution resulting from heterogeneous mixtures in intake manifolds affect thermal efficiency and NO_x emissions for gasoline-fueled (Pre IVC) engines. Similar affects are expected for hydrogen-fueled Pre IVC engines.

Billings Energy Corporation (BEC), (1977) measured a 10% variation in cylinder-to-cylinder fuel-air equivalence ratio in their hydrogen-fueled Pre IVC engine. This variation explains why, when comparisons were made at equal brake mean effective pressure, the brake thermal efficiency for their engine was the same as that found for an engine with no distribution problems (Swain, 1973), even though the former engine had a compression ratio of 12 and the latter a compression ratio of 10.

The evidence of heterogeneous mixtures in the intake manifold suggests that the high diffusion rate of hydrogen is not the

dominant mechanism governing fuel-air mixture concentration patterns in the manifold and that turbulence effects predominate.

3.9 FLAME INITIATION FAILURE

Flame initiation failure in a cylinder caused by an ignition-system malfunction is a phenomena that will, in all likelihood, occur during operation of practical hydrogen-fueled engines. As discussed here, this failure can lead to some potentially serious problems especially with regard to safety. It is therefore important to gain an understanding of the events that occur due to flame initiation failure.

A potential occurrence after flame initiation failure is an overheating of the exhaust manifold or generation of high pressure in the exhaust system resulting from burning of the charge therein. Misfire could damage an exhaust-system catalyst should one be present for NO_x control purposes. Misfire could also result in the presence of a still-burning mixture in the cylinder when the intake valve opens; this mixture having been ignited in the exhaust system. Subsequent flashback could occur.

3.10 HYDROGEN CONCENTRATION IN THE CRANKCASE (BLOWBY)

Blowby can result in hydrogen concentration build-up in the crankcase with the potential for producing a combustible mixture therein. The presence of a combustible mixture in the crankcase could pose a serious safety problem. While blowby information is available for gasoline-fueled engines data are not reported to date for hydrogen-fueled engines. As a result of the different pressure history in a hydrogen-fueled engine cylinder and the high diffusion rates of hydrogen, the blowby characteristics could differ from those of a gasoline-fueled engine and they must therefore be experimentally evaluated. Blowby rates and the related restriction on the allowed hydrogen concentration in the crankcase will determine design requirements for hydrogen-engine crankcase-ventilating systems.

3.11 MISCELLEANEOUS EMISSIONS (CO, HC, H₂O₂)

Emissions of carbon monoxide, hydrocarbons (both from the lube oil) and hydrogen peroxide are expected to be small but should be monitored during engine research and development studies at selected conditions.

3.12 RELATED PRACTICAL CONSIDERATIONS

In addition to the engine-related problems discussed in Section 3.2 - 3.11 there are a number of important considerations of practical interest the understanding of which is essential for making a hydrogen engine design selection.

3.12.1 Initial Costs

The initial cost of a hydrogen-fueled engine is a function of complexity of design, materials, and manufacturer's tooling requirements which in turn depend on engine type (Post vs. Pre IVC, etc.).

3.12.2 Operating Costs (Exclusive of Fuel Costs)

As defined here, operating costs include maintenance, oil, and water for the water injection versions of the hydrogen-fueled engine. Maintenance cost estimates for the various potential hydrogen-fueled engine configurations are discussed in Section 4.3.10.

Oil costs are expected to be no more than those for gasoline-fueled engines. It should be pointed out that the cost of water for the water-injection-type engine could be significant. The last statement is based on the results of tests in Miami where city water injection was found to increase flashback problems as a result of deposits formed from material in the water. The problems were not present when distilled water was used. The collection of water by condensing exhaust gases on the vehicle is not expected to be a cost-effective solution for water-injected engines in the light of alternate solutions to flashback and NO_x control discussed in Section 4.3.4 and Appendix 2.

3.12.3 Complexity of Design

History shows that the most widely-accepted automotive powerplants are those that are inherently simple in design. The auto industry has always, for good reasons, avoided complex costly design configurations. Therefore, complex designs should be avoided unless it is demonstrated that the simple ones are inadequate.

3.12.4 Safety

It is of prime importance that practical automotive hydrogen-

fueled engines be safe with the possibility of hydrogen leakage being minimized. Thus, engine configurations requiring high pressure hydrogen are less desirable than those requiring low pressure levels. There are other safety considerations such as flashback and crankcase blowby which have been discussed previously.

3.12.5 Fuel System/Engine Compatibility

Three basic on-vehicle hydrogen storage methods have been identified; metal hydrides, liquid hydrogen, and compressed gaseous hydrogen.

If a metal hydride is used then there must be both sufficiently high temperature levels to effect desorption and adequate quantities of available heat, at or above this temperature, in the exhaust gases. For these reasons it is important to know exhaust-gas temperatures and flowrates (to assess heat quantity) for hydrogen-engine configurations of interest. Since practical metal hydrides may be operated at close to atmospheric pressure, it may be necessary to utilize a compressor with Post IVC engines which require high hydrogen pressure levels on the order of many hundreds of pounds per square inch.

The use of either liquid or gaseous storage does not impose the heat transfer requirements associated with the hydride storage system but still requires some form of compression for Post IVC engines. The compression could be avoided with gaseous hydrogen systems at the expense of an extreme decrease in storage capacity since the tanks would, in effect, be depleted when pressure levels reach values approaching fuel injection pressure levels (on the order of hundreds of pounds per square inch). Since gaseous storage of hydrogen is at best a marginal solution, the avoidance of a compressor by this method of operation does not seem very practical.

In the case of liquid hydrogen storage the use of some type of cryogenic pump is implied for the Post IVC systems.

4. POTENTIAL HYDROGEN-ENGINE CONFIGURATIONS - DESCRIPTION AND PRELIMINARY EVALUATIONS

In the following sections a wide variety of potential hydrogen engine configurations are described and the evaluation of them made at the onset of the present work is given. The engines are grouped into the two major classifications given in Section 3.1: Pre Intake Valve Closing Engines in which the fuel is introduced into the cylinder prior to the closing of the intake valve and Post Intake Valve Closing Engines in which the fuel is introduced into the cylinder after the intake valve has closed. Thus, with the Pre IVC engine the hydrogen is at a relatively low pressure when it is ingested by the engine, with the Post IVC engine the hydrogen is at a high pressure. Within each of these major engine classifications a number of configurations have been identified and are summarized in Table 1 where 10 Pre IVC and 4 Post IVC engine types are described.

4.1 PRE INTAKE VALVE CLOSING (PRE IVC) ENGINES

In Table 1, Pre IVC engines 1 through 6 utilize a carburetor to meter the fuel at the beginning of the intake manifold. With engines 7 through 10 the metering is done in the valve pocket of the intake port, with the fuel being injected into the port during mid portions of the intake stroke.

Both throttled (engines 1, 2, 3) and unthrottled (engines 4 through 9) Pre IVC engines are listed as well as one (engine 10) which utilizes a combination of throttling and unthrottled operation (quality control) as a means of load control (see Appendix 2). Finally, for nitric oxide control, either exhaust gas recirculation (EGR) or water injection is listed as a feature of some of the engines. The simultaneous use of EGR and water injection is not listed in the Table, as this approach is considered to be not practical in the light of alternate ones.

4.2 POST INTAKE VALVE CLOSING FUEL INGESTION ENGINES

The Post IVC engines are subdivided into two groups: one with a single combustion chamber (engines 11 and 12) and the other with dual combustion chamber - a main chamber and a pre-chamber (engines 13 and 14). Finally each of the Post IVC engines

are classified as to whether combustion commences during fuel injection (engine 11 and 13) or after fuel injection is complete. EGR and water injection have been intentionally omitted for the Post IVC engines because it is felt that if the Post IVC engines (which are already considered to be of sufficient complexity to be marginal for automotive applications) require EGR and/or water injection for NO_x control then they would come into use only as a last resort.

4.3 EVALUATION OF ENGINE CONFIGURATIONS

Prior to making a selection of engine configurations to be tested in the present work it was felt necessary to evaluate the various hydrogen engine configurations with regard to the various engine performance, emissions and practical use characteristics described in Section 3. To this end, Table 2 was prepared in which each of the 14 engine configurations described in Sections 4.1 and 4.2 is rated for each of the 15 characteristics given in the Table. Except for the characteristic maximum power, the ratings are inter-engine comparisons and by necessity, as a result of the lack of a design data base, qualitative in nature. (An objective of the present work is to generate sufficient information to make Table 2 more quantitative). Where deemed necessary, some of the characteristics are compared at both high- and low-load conditions. The maximum-power rating is given at both the operating condition of maximum-horsepower engine speed and maximum-torque engine speed. One hundred percent is the horsepower output of a gasoline engine operating at maximum-horsepower engine speed. Appendix 1, in conjunction with Section 3.6, explains the details of the calculation procedure used to estimate the Table 2 maximum-horsepower ratings. Some of the ratings are based on a review of the data available at the onset of the present study and are identified by their being typed in UPPER CASE letters. Other ratings were estimates, as determined by a review and analysis of the then available literature, and are identified by their being typed in lower case letters.

4.3.1 Oxides of Nitrogen Emissions

The first entry in Table 2 is NO_x emissions. At low loads NO_x emissions were found to be low for engines tested previously

and were expected to be low for those engines not tested (8 through 11). As discussed in Section 3.2, NO_x emissions are a problem at high loads unless some means of control is utilized such as EGR (engines 2, 5 and 8) or water injection (3, 6 and 9) or combinations thereof. An exception to this need of control is engine 10 which is expected to reduce the NO_x emissions problem to a minimum without EGR or water injection. A detailed description of this engine termed the Discontinuity Engine and the method it employs for nitric oxide control is given in Appendix 2.

It should be noted that for prechamber Post IVC engines there exists a unique solution to the problem of NO_x reduction. This solution involves the quenching of NO_x formation reactions and has been applied with success to gasoline-fueled prechamber engines (Newhall and Messiri, 1970) where reductions in NO_x emissions of an order of magnitude or more have been observed. The quenching of the NO_x formation reactions is accomplished by burning a rich mixture in a relatively large prechamber and letting it expand into the main combustion chamber thereby letting it cool before high combustion temperatures and the corresponding high equilibrium NO_x concentrations are reached.

4.3.2 Brake Thermal Efficiency

At high loads the available literature showed that all hydrogen engines tested exhibited high efficiency except for engines 12, 13 and 14. It was felt, however, that the latter engines have the potential for high efficiency at high loads as is indicated in Table 2. For low-load operation the efficiency is a function of engine design, with the throttled engines exhibiting low efficiency levels. All of the other engine types exhibit high efficiency at low loads.

4.3.3 Maximum Power

The maximum power output capabilities of hydrogen engines are seen to be a strong function of design configuration. All of the Post IVC engines yield higher maximum power levels as a result of the high pressure injection of fuel while, to varying degrees, the Pre IVC engines yield lower maximum power levels due to partial displacement of the air by the gaseous hydrogen

fuel in the cylinder after the intake stroke. For engines with intake manifolds, cylinder heads, and cams designed specifically for hydrogen-fueled operation, however, the reduction in peak power is not expected to be as great as that which occurs when gasoline-fueled engine components are utilized (see Section 3.6). In fact, with engine 10 a reduction in peak-power level of only around 4% is expected. Table 2 clearly illustrates the very large power output penalties incurred when EGR is used to reduce flashback and NO_x emissions.

4.3.4 Flashback

Flashback problems exist in Pre IVC engines. It has been demonstrated, however, that the problem can be alleviated by the use of water injection and/or EGR or techniques to cool hot spots. A potential technique is to use timed injection of hydrogen with the fuel being injected after allowing a sufficient quantity of fresh air to enter the combustion chamber and reduce the temperature levels of potential hot spots or, perhaps, eliminate reactions leading up to preignition and flashback. With the Post IVC engines flashback is not possible.

4.3.5 Uncontrolled Combustion

At light loads, uncontrolled combustion is not a problem for either Post or Pre IVC hydrogen-fueled engines. At high loads, however, the problem can exist for Pre IVC engines and would be related to hydrogen's high flame speed and low ignition energy. While uncontrolled combustion has been found to be a problem at high loads for Post IVC engines (12, 13 and 14) it is expected that, with proper design, the problem can be reduced as is indicated in Table 2.

4.3.6 Heterogeneous Mixtures - Combustion Chamber

In the Pre IVC engines problems associated with nonhomogeneous mixtures in the combustion chamber are expected to be slight as there should be sufficient time for mixing of the gaseous hydrogen with air after the intake stroke. Water injection or EGR could make the problem somewhat worse. It should be mentioned here that if charge stratification can produce reductions in NO_x emissions the charge may be intentionally stratified to yield rich mixtures near the spark plug as discussed in general terms in Section 3.7.

In the Post IVC engines nonhomogeneous mixtures have been found experimentally to pose problems. This fact is reflected in Table 2 for engines 12, 13 and 14. Based on these experiments, problems are anticipated for engine 11.

In summary, nonhomogeneity in the combustion chamber of Pre IVC engines presents small problems as compared to the situation in Post IVC engines where serious problems are encountered.

4.3.7 Heterogeneous Mixtures - Intake Manifold

Nonhomogeneous mixtures resulting from poor carburetor design can present a problem in carburetted engines but no problem in timed-injection Pre IVC engines or Post IVC engines as reflected in Table 2.

4.3.8 Hydrogen Concentration in the Crankcase (Blowby)

For all of the Pre IVC engines, leakage of unburned hydrogen from the hydrogen-air mixture past the piston rings may occur. With the Post IVC engines the leakage is less likely, especially for engines 12 and 14 where hydrogen is burned during injection.

4.3.9 Initial Costs

In Table 2 the cost of a carburetted, unthrottled engine (Pre IVC engine 1) is arbitrarily defined as low. It is observed that Pre IVC engines are of low cost except when EGR, water injection, or discontinuity engine design (Appendix 2) is used for flashback and NO_x emission control in which case the costs are medium. Post IVC engines, on the other hand, are expected to have high initial costs.

4.3.10 Operating Costs (Exclusive of Fuel Costs)

In Table 2 operating cost is defined as being low for engine 1. It is seen that all Pre IVC engines exhibit low operating costs except for the water injection engines where the costs are expected to be medium as a result of the need, in some regions of the country, to use treated water to remove nucleation-site-forming impurities. Post IVC engines, on the other hand, are expected to have high operating costs caused by anticipated high maintenance costs.

4.3.11 Complexity of Design

Complexity of design is thought to be a general indicator of dependability. Throttled (No. 1) or unthrottled (No. 4) carburetted engines and the unthrottled timed injection engine (No. 7) are rated good with regard to complexity of design. The design becomes more complex when EGR, water injection, or the combination throttling and quality-control engine (No. 10) is used for control of flashback and/or NO_x emissions. The design configurations of Post IVC engines are very complex and thus they all are rated as poor.

4.3.12 Safety

All of the Pre IVC engines have about the same safety problems and are given a rating of average. As a result of the high pressure requirements of Post IVC engines, they are all rated below average from a safety point of view. Flashback, another safety problem, has been considered elsewhere.

4.3.13 Fuel System/Engine Compatibility

Since all of the Pre IVC engines require a hydrogen supply close to atmospheric pressure, they are all given a rating of fair with regard to the problem of fuel system/engine interfacing. Post IVC engines require high hydrogen supply pressures, are thus rated poor, and are less desirable than Pre IVC engines from a fuel system/engine compatibility point of view.

5. TECHNICAL APPROACH

The technical approach to the gathering of the desired data base for hydrogen engine designers is to test and evaluate each of the potential hydrogen engine configurations of Table 1.

The first two years of the project are devoted to testing and evaluating the Pre IVC type of hydrogen engine. The review of Section 3 and the evaluation of Section 4.3 led the authors to conclude that for Pre IVC hydrogen engines there are two major problems which must be overcome: (1) flashback and (2) NO_x emissions at high load. To attain solutions to these problems it is essential that NO_x emissions from the various Pre IVC engines be characterized and that an understanding of the causes of flashback be ascertained. Thus, studies of flashback and NO_x formation for the various Pre IVC engine configurations were considered to be of paramount importance followed by more detailed performance characterization. This consideration governed to a large extent the design of the technical approach and the testing program.

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6. SELECTION OF TEST ENGINE CONFIGURATIONS

As stated in Section 5, the technical approach was to test the various potential hydrogen-engine configurations delineated in Table 1. The criteria for the selection of test-engine configurations was two-fold: (1) The testing of the engines selected must provide sufficient data to permit the filling in of Table 2 with as much quantitative information as practical; (2) The selection must include engine types that are suited for collection of baseline data, as well as engine types offering potentially-practical solutions to the problem of automotive hydrogen-fueled engine design. Baseline data are absolutely necessary for an understanding of the phenomena occurring in hydrogen-fueled engines and for making comparisons. Baseline data, however, are most readily obtained from test engines which do not necessarily exhibit satisfactory performance and emissions characteristics. Such test engines (viz Table 1 Engine Configurations 1 and 4) are of little practical importance, but they must be tested. In order to generate a comprehensive design base data set, it is necessary to test other engine types which are of practical interest even though they do not yield fundamental baseline data as a result of the coupled effects associated with their various emission- and performance-control features such as EGR or water injection (viz Table 1 Engine Configurations 2, 3, 5, 6, 8, 9, 10).

Prior to describing in detail the Pre IVC test engines selected, the following list, delineating some of the criteria used in making a selection of hydrogen-fueled test engines, is given (Swain, 1973): (1) A multicylinder engine, as specified in the Request for Proposal for this project, is obviously needed to study geometric (cylinder-to-cylinder) fuel-air ratio distribution; (2) The engine should have no more than four cylinders to minimize time spent machining new parts; (3) An engine incorporating fully-machined combustion chambers is desirable to ease the problems involved in analyzing combustion chamber deposits; (4) An engine without gamed exhaust ports is desirable to enable isolation of the exhaust gases from individual cylinders. All of these requirements are satisfied by the Pre IVC-type engine

configurations selected for the present work. For the Post IVC-type test-engine configurations it was decided to use a single cylinder engine.

Three test-engine configurations have been selected for determining Table 1 Pre IVC engine configurations 1-10 performance and emissions characteristics:

Configuration (1) - Toyota 2TC, spark-ignited four cylinder engine fitted with a gaseous carburetor and means for water injection and EGR. The engine is to be operated in both the throttled and unthrottled mode of operation thus yielding Table 1 Potential Hydrogen Engine Configurations 1-6.

Configuration (2) - The above engine with the cylinder head modified to allow for hydrogen induction into the intake port during a portion of the intake stroke thus yielding Table 1 Potential Hydrogen Engine Configurations 7-9.

Configuration (3) - The short block assembly and cylinder head of Configuration (2) with the intake manifold modified to allow for the discontinuous mode of operation thus yielding Table 1 Potential Hydrogen Engine Configuration 10.

Time allowing, Pre IVC engine tests will also be conducted using a second Toyota 2TC short block assembly fitted with different pistons to yield a higher-than-stock compression ratio (12:1 vs 9:1).

To determine the Table 1 Post IVC engine configuration performance and emissions characteristics, a single cylinder engine, constructed by using one of the Toyota 2TC engine cylinders, will be used.

During the first year of the project Configuration (1) was tested and it is described in detail in the following sections.

7. TEST ENGINE, COMPONENTS AND SELECTION RATIONALE

In the following sections the test engine used is described along with its components. Reasons for the selections made are also presented.

7.1 ENGINE

The specifications of the Toyota 2T-C, four cylinder, hemispherical combustion chamber test engine used during the first year to simulate potential hydrogen engine configurations 1 - 6 are given in Table 3. This assembly was selected because the authors have had considerable experience using this engine for hydrogen applications (Swain and Adt, 1972; Swain, 1973). In addition, parts are readily available and the parts suppliers have been found to be very cooperative. The engine satisfies the four criteria given in Section 6 and, further, it is judged readily adapted to establish all of the potential hydrogen engine configurations given in Table 1.

7.2 CAMSHAFT

Four camshafts were used (Table 4): Camshaft C1 - the stock Toyota camshaft; Camshaft C2 and C3 both low-overlap, low-residual designs; and lastly C4, a camshaft with zero degrees valve overlap. Camshaft C1 was used for break-in and flashback comparison testing. Camshaft C2 was used for the bulk of the testing program.

Based on considerations of engine displacement, intake- and exhaust-system flow characteristics, camshaft lobe profiles, and piston motion versus crankshaft angle, Camshaft C3 was originally expected to produce torque versus engine speed characteristics similar to those of modern American-made, passenger-car engines and would thus be used for the bulk of the testing. Preliminary testing showed Camshaft C2 to yield adequate torque characteristics and brake specific NO_x emissions similar to those of Camshaft C3. Since Camshaft C2, while no longer used in new Toyota engines, is a readily available stock Toyota camshaft for which gasoline-fueled operation performance and emission data are also available, it was chosen as the one for the bulk of the testing. It should be pointed out that Camshaft C2 is of the pre-emission control design. It does not have the currently used,

relatively-large overlap. The valve overlap chosen minimized internal EGR without penalizing high-RPM torque characteristics.

Camshaft C4 was used at an operating condition where flashback was experienced with Camshaft C2 in order to test the hypothesis that flashback is not caused by communication of inlet-manifold gases with expelled exhaust gases (the negative of item (3), Section 3.3). Since the exhaust lobe duration of Camshaft C4 was similar to that of Camshaft C2, the exhaust valve temperature was expected to be very similar during operation with these camshafts. If the engine continued to flashback when it was operated at the same condition when Camshaft C4 was substituted for C2, then the hypothesis that communication of expelled exhaust gas with intake manifold gas does not cause flashback would not be accepted. The results of the test of this hypothesis is given in Section 11.2.2.

Camshafts C3 and C4 were made by Crane Cam, Inc. In Table 4 the specifications for the four cams are given. The specifications are according to the method of Crane Cam, Inc. who define valve opening and closing angles as the angle where there is 0.050 in. lift at the lobe.

7.3 ENGINE OIL (LUBRICANT)

Valvoline gas engine oil AD-1, part number 413, SAE 30 weight was used as the engine lubricant because of its low ash content. This oil is the same one recommended for natural-gas-fueled engines.

7.4 CARBURETOR

Original plans called for the use of an Impco gas-mixer carburetor. Testing was begun with an Impco gas-mixer consisting of a CA 125M mixer (W/A3-60+AF1-5)*, a AJB-2 converter and a VFF 30-2 vacuum fuellock. With this mixer system, cylinder-to-cylinder fuel-air ratio variation was found to be quite large. In fact, idling performance in the unthrottled mode was not acceptable. A Beam gas-mixer consisting of a Model SC-60-HA carburetor, a 6C-177-177 orifice and a 341-29 venturi was subsequently installed on a one inch riser atop the standard intake manifold. Cylinder-to-cylinder fuel-air ratio variation improved

* Impco specification number.

from approximately $\pm 10\%$ with the Impco carburetor to approximately $\pm 2\%$ with the Beam carburetor in the unthrottled mode of operation. The Beam carburetor was therefore deemed the proper one to gather baseline data.

7.5 EXHAUST VALVES

It is apparent from Section 3.3 (flashback problem) that proper exhaust valve design will be an important ingredient for Pre IVC engines. The design scheme will depend to a large extent on the thermal characteristics of the valve in combination with the exhaust-valve seat, exhaust-valve guide, cylinder head, and coolant-passage geometry and flowrates. These characteristics and the methods that were used to ascertain some of them are described below.

The sodium-filled valve invented by Heron (1928) is well known and widely used to lower exhaust valve operating temperatures (Danis, 1973). Such valves, which reduce peak valve temperature by about 200°F (110°C), have been used to extend the useful range of hydrogen-fueled engines (Swain, 1973). Testing of water-filled exhaust valves (Danis, 1973) which utilize the enthalpy of vaporization of water to increase heat transfer rates, has shown an additional 300°F (166°C) reduction in peak exhaust valve temperatures under certain conditions. Since the peak temperatures for water-filled valves are found to be near the seat (Danis, 1977), improved seat design should reduce peak temperatures even further (about 50°F). A disadvantage of the water-filled valves is the possibility of "burnout" (Danis, 1973) which occurs when film boiling takes place in the stem thereby drastically reducing heat transfer rates out the stem. Another problem with water-filled valves is the potential of operation at the critical temperature (approximately 700°F).

It should be mentioned that the use of valves with larger stem diameters or valves with more efficient internal valve coolants can serve to lower peak exhaust-valve temperatures provided no other thermal resistances are present that would impede the potentially greater heat transfer rates afforded by these valves. Potential resistances include: film boiling of the coolant, coolant flow stagnation pockets, the oil film annulus between the valve stem and

guide, guide-to-head contact resistance and the resistance of the heat-flow path in the cylinder head. Film boiling would not only impede the heat transfer rate but could also act as a barrier setting an upper limit to the heat transfer rate out the stem.

Original project plans were to characterize sodium- and water-filled exhaust valves. Personal communication with Mr. L. Danis of the Eaton Corporation (Danis, 1977) brought to light test results using lithium as an internal valve coolant. The results have shown that lithium exhibits very favorable characteristics as an internal valve coolant; they are:

1. Absence of high pressure at operating temperatures.
2. A very high heat transfer coefficient at the lithium/cavity wall interface.
3. Suspected scavenging action by lithium keeping the cavity walls clean thereby maintaining characteristic 2 and a high wettability.
4. A higher overall heat transfer to the stem than offered by water-filled valves.

Disadvantages of lithium as an internal valve coolant are associated with its very reactive nature. Lithium has been shown to attack welds in previous lithium-valve research, and to exhibit a decarburizing action adversely affecting the valve material (Danis, 1977). A potential solution to the decarburizing problem would be to plate or coat the cavity with iron or some resistant material.

As a result of the very encouraging findings regarding the characteristics of lithium-filled exhaust valves and the relative ease of their construction, they were used instead of water-filled valves in the test program. These valves have the potential of bringing the exhaust valve head temperature down to the temperature of the intake valve.

Any reduction in exhaust-valve peak temperature with the sodium-, water- and lithium-filled exhaust valves (super-conductive valves) is at the expense of a greatly increased temperature level of the valve stem and increased heat rejection

requirements (Danis, 1973).

If higher valve stem and guide temperatures are present, valve-stem lubrication will be aggravated. Lubrication problems, however, were not expected nor experienced during the project experiments. Redesign of future cylinder heads for hydrogen-fueled engines utilizing super-conductive valves could aid in reducing the heat load on the lubricant at the guide. One potential solution would be to lengthen the valve stem and guide and bring the bearing surface higher up on the stem by increasing the valve-guide clearance at the lower portion of the guide. Modifications may also be required to permit more uniform and adequate coolant velocity around some valve guides which currently cannot manage stem head rejection rates from super-conductive valves (Danis, 1973) .

The thermal characteristics of sodium- and lithium-filled exhaust valves, the seat, stem and guide were determined by monitoring the temperature at select locations in the cylinder head near the valve. A finite element, electrical analogue model following the techniques of Hires and Pochmara (1976) was formulated. The measured temperatures were used to verify the validity of the model and to make appropriate adjustments to the approximate heat transfer coefficients given by Danis (1973), Stotter, et al (1965), and Hires and Pochmara (1976, 1977), (these are reproduced in Figure 2). The model, in conjunction with the measured temperatures, yielded approximate values of peak valve temperature and stem and seat heat rejection requirements for the various valves tested (Section 11.2.1).

In order to determine the proper locations near the seat and valve guide for the thermocouples to be used for monitoring temperatures, a Toyota cylinder head was sectioned. Figure 3 is a view of the sectioned head on which the probe locations are shown. The three holes for the guide temperature probes, Gd a, Gd b and Gd c, were drilled and Chromel-Alumel thermocouples, sheathed with 0.040 in O.D. stainless steel tubing, were inserted until they contacted the guide. Four holes (one for each cylinder) were drilled at the other probe location shown in Figure 2 into which the same type of thermocouple probe was placed in contact

with the exhaust-valve seat to yield the Figure 3 thermocouple temperature T, seat i. The method used for installing the thermocouples in the head is explained in Appendix 3.

The temperature data thus obtained will be analyzed to determine if exhaust-valve seat temperature, which in turn is directly related to exhaust-valve temperature, can be correlated with individual cylinder air-fuel ratio, detonation, flashback and abnormal combustion.

Sodium-filled exhaust valves from the following vehicles were investigated as potential candidates for the present study:

- 1970 Porsche 911S
- 1968 Porsche 911
- 1971 Mercedes 250
- 1969 Mercedes 220
- 1964 Mercedes 190SL
- 1972 Alfa Romeo 2000
- 1970 Alfa Romeo 1300
- 1969 Alfa Romeo 1600
- 1969 Dodge Truck 361 cu. in.
- 1968 Dodge Truck 251 cu. in.
- 1971 BMW 2000

Where necessary, the valve was sectioned to ascertain its relative merit. The Alfa Romeo 1300 cc valve was selected as the best candidate. It has a better cavity design near the head of the valve than does its 1600 cc counterpart, thereby making it more suitable. This improved cavity design is shown in Figure 4 where it is seen that there is a greater surface area in the cavity at the head of the valve and that there is a smaller distance, and thus less thermal resistance, between the cavity and the combustion-chamber side of the valve. The specifications of the Alfa Romeo 1300 cc exhaust valve are:

- Part Number - 105390330100
- Stem Diameter - 0.352 in.
- Head Diameter - 1.34 in.
- Length - 4.27 in.
- Seat Angle - 30°
- Lock Groove Type - Single Groove

This type valve was used to construct lithium-filled valves. The construction involved removing the valve stem tip, emptying the sodium and filling with lithium and welding a replacement

tip. (See Appendix 4)

7.6 WATER INJECTION

For the test-engine configurations used to simulate potential hydrogen-fueled engine configurations 3 and 6 (Table 1), water was injected into the runners of the intake manifold about four inches from the intake valve. The water was pressure-fed from a reservoir by way of copper tubing to the injectors at a continuous rate. The entire system was designed and adjusted to minimize cylinder-to-cylinder variation in water injection rate. A $\pm 5\%$ variation in flowrate was present in the final configuration. The leanest cylinder was fitted with the lowest-capacity water injector.

Two types of injectors were used: (1) fuel injectors from the Bosch gasoline fuel-injection system used in the Volkswagen engine; and (2) orifice injectors having a diameter of 0.016 in. The Bosch injectors yielded a very fine mist and the orifice injectors yielded a jet of water. During testing, the difference in NO_x emissions and flashback control when using the different injectors was found to be minimal. This finding and the fact that the Bosch injectors had a tendency to become clogged resulted in choosing the orifice injectors for the testing program. Photograph 1 shows the orifice injectors and Photograph 2 shows the injection system.

For all of the tests distilled water was used.

7.7 EXHAUST GAS RECIRCULATION

The method of controlling and quantifying EGR rates involved the division of the exhaust-gas flow in 19 separate and equal parts and directing anywhere from one part up to six of the parts to the EGR carburetor. To accomplish this division of the exhaust gas, the device shown schematically in Figure 5 was constructed. In Figure 5 only two of the 19 passages into which the exhaust gas flow was divided are shown. In the actual device, a portion of which is shown in Photograph 3, there are 6 streams i and 13 streams j . The i streams are directed to either the EGR carburetor or to the exhaust pipe leading outside the building. The number of i tubes used depended on the desired EGR rate. All of the i and j steam flowrates are controlled and made equal to each other by adjusting their respective flowrate control valve (FC_i or FC_j)

until the pressure P_2 is the same for each passageway.

To insure that each of the 19 streams contain the same flow-rate of exhaust gas when the pressures P_{2i} and P_{2j} are equal, the following constraints were imposed on the design:

1. A sufficient length of straight pipe upstream of Station (1) was used to insure parallel streamlines and thus equal pressure, P_1 , at the entrance to each of the 19 passageways.
2. The straight pipe upstream of Station (1) was insulated to minimize temperature and thus density variations at Station (1).
3. The Reynolds number in each of the 19 passageways was kept sufficiently high to minimize entrance effects caused by the passageway-to-passageway variation in entrance [Station (1)] velocity profiles.
4. The 19 passageways between Stations (1) and (2) were of the same dimensions thereby resulting in the same relation between pressure drop ($P_1 - P_2$) and exhaust-gas flowrate for each passageway for identical entrance [Station (1)] conditions.

In principle, for steady flow, strict adherence to the above-described design constraints will result in the same relation between exhaust-gas flowrate and pressure drop for each of the 19 tubes. The relation is of the form:

$$\dot{m} \propto \sqrt{P_1 - P_2}$$

where $\dot{m}$ is the flowrate in each passageway; P_1 is the upstream pressure for each passageway; and P_2 is the downstream pressure for each passageway. To insure steady flow, surge tanks were incorporated as a part of the EGR gas flow system.

With this EGR system the EGR rate is varied in discrete steps. Theoretically, for one of the i passageways, the EGR rate is $1/19$ or 5.26% of the exhaust gas flow; six of the i tubes yield 31.6% and so forth. Calibration of the separator showed it to operate with a maximum deviation from theory of 3.5%.

After separation of the required amount of EGR gas, it was fed to the engine by way of a gas-mixer EGR carburetor mounted

atop the hydrogen-metering carburetor. A gas-mixer carburetor, rather than direct induction of EGR gas into the intake manifold was selected because it was felt that the carburetor would give a more uniform cylinder-to-cylinder distribution of EGR gas. The EGR carburetor is identical to the Beam carburetor described in Section 7.4. The entire installed EGR system is shown in Photographs 4 - 6.

Original plans called for the use of the EGR separation and quantification method described in Appendix 5. For reasons given there, it was discarded. It is felt, however, that for some situations the Appendix 5 method is a good one and thus the presentation of it.

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8. MEASUREMENTS AND ANCILLIARY EQUIPMENT

Certain measurements and ancilliary equipment are felt to be of particular concern to the present work associated with the use of hydrogen as a fuel. In Sections 8.1 - 8.6 these items are discussed. Measurements and ancilliary equipment of a more standard nature are not discussed in detail, instead they are delineated in Section 8.7. Certificates of Calibration for some of the instruments utilized are given in Appendix 6, with clarifications where necessary.

8.1 FUEL AND AIR FLOWRATE MEASUREMENTS

Since the confidence level of measured brake thermal efficiency and fuel-air equivalence ratio are critically dependent upon the accuracy of fuel flowrate and air flowrate measurements a survey was made of the available flowrate measuring devices. With the help of Dr. David W. Baker of the National Bureau of Standards Fluid Mechanics Department, Table 5 was compiled to aid in the selection of the fuel flowrate measuring method to be used throughout the project.

8.1.1 Fuel Flowmeter

Selection - Among the criteria for the fuel flowrate meter selection are: high precision, high accuracy, availability of hydrogen calibration facilities, ease of maintenance, long-lived repeatability and accuracy, low fuel-pressure requirements and reasonable cost. It was also felt desirable to have more than one meter to bridge the entire range of fuel flowrate (140 to 1); with multiple meters the ranges can be overlapped affording the opportunity to conduct pretest calibration checks in the overlap regions. An additional criteria to be considered when rotameters are candidate flowmeter types is readability. Readability is a major contributor to total accuracy when high accuracy is desired in a rotameter; it varies with the length, dynamic range, and type of tube taper. For example, Cox gives a readability of $\pm 1\%$ for their logarithmic tube 120 model which has a large dynamic range (13:1) and $\pm 0.5\%$ for their logarithmic tube 129 model which has a small dynamic range (5:1).

Based on the above considerations, a high-precision

rotameter with a potential precision of $\pm 0.2\%$ of reading and an accuracy of $\pm 0.5\%$ of reading was selected along with a high-accuracy mercury manometer and certified standard calibrated thermocouple as the means of measuring the fuel flowrate. The meter was manufactured by Cox (Model 129-287-Range) and is composed of three logarithmic tubes each having a readability of about $\pm 0.5\%$ of reading.

Calibration - Calibration for standard application, high-accuracy flowmeters is conducted by using air as a flow medium and then modifying the calibration curves by using the appropriate theoretical correction factors for the fluid to be metered in practice. This procedure results in accuracies of $\pm 1\%$ for high-precision meters exclusive of readability inaccuracy and inaccuracies associated with measuring the properties of the metered fluid. Special applications requiring greater accuracy are handled by calibrating with either: (1) the fluid to be used in practice or; (2) a fluid that closely simulates the physical flow properties of the fluid to be used in practice (Cox Instruments calibrates with helium for hydrogen simulation). Calibration using hydrogen as a working fluid is not standard practice. The increased accuracy obtained by hydrogen calibration is not commonly required and the potential safety hazard of venting hydrogen during or after calibration is not a task looked upon favorably by those in the calibration laboratories.

Since the Cox, high-precision flowmeters potentially offered a high degree of accuracy through calibration with hydrogen, a search was made to find a facility to perform this calibration. As a result of this search the Colorado Experimental Engineering Station, Inc. (CEES), Boulder, Colorado, was contracted to calibrate each of the logarithmic tubes in the Cox fuel flowmeter. CEES calibrated and certified the meter tubes to an accuracy of $\pm 0.5\%$ of reading. The calibration was conducted at 12 points for each logarithmic tube with six of the points being repeated. Since the rotameter is a volumetric flowmeter, the pressure and temperature of the metered fluid must be measured to ascertain mass flowrates. Errors in these measurements reduce the total accuracy. The maximum error for the fuel mass-flowrate measurement, was approximately $\pm 1.33\%$ found by adding the following components

(at full scale):

Manufacture specs. meter readability	$\pm 0.5\%$
Calibration error	$\pm 0.5\%$
Temperature ($\pm 0.5^{\circ}\text{F}$) error	$\pm 0.2\%$
Barometric Pressure error	$\pm 0.07\%$
Hg Manometer error	$\pm 0.06\%$
	<hr/>
Maximum error	$\pm 1.33\%$

Note that 1.33% is the maximum error and not the probable error found by taking the square root of the sum of the squares of the above-listed errors.

8.1.2 Air Flowmeter

Air flowrate measurements were made with a laminar flow element with a certified accuracy of $\pm 0.5\%$ of reading, using for pressure-drop measurement a high-precision inclined manometer having an accuracy of $\pm 0.25\%$ of reading. The instruments selected were:

Meriam 50 MC2-2F Laminar Flow Element
Meriam 40H35, 50" Scale Manometer

Like the fuel-flow measurements it was necessary to ascertain the density of the air by measuring air pressure and temperature. In addition, it was necessary to determine the humidity of air. The maximum (not probable) error for the air mass-flowrate measurement was approximately $\pm 1.67\%$ found by adding the following components at full scale:

Calibration of Laminar Flow Element error	$\pm 0.65\%$
Calibration table readability	$\pm 0.25\%$
Readability of 50" Scale Manometer	$\pm 0.25\%$
Temperature ($\pm 0.5^{\circ}\text{F}$) error	$\pm 0.2\%$
Barometric Pressure error	$\pm 0.07\%$
Hg Manometer error	$\pm 0.25\%$
	<hr/>
Maximum error	$\pm 1.67\%$

8.2 FUEL-DELIVERY SYSTEM

The mass flowrate measurement of a gaseous fuel like hydrogen requires knowledge of the pressure and temperature at the flowmeter. Hydrogen fuel was stored as a gas in standard welding-type tanks. During the experiments, the temperature and pressure in the tanks decreases as the hydrogen was consumed. In order to minimize the temperature drop, a fuel-delivery system consisting of two sets of four tanks connected in parallel was used. During these tests, fuel was emptied simultaneously by all four tanks of one set. Upon depletion of the fuel in one set the other set could be turned on allowing for continuous operation without shut-down. In addition to the simultaneous use of four tanks, the fuel was passed through a coil of tubing immersed in a constant temperature bath to aid in maintaining a constant temperature at the flowmeter. A two-stage pressure regulator downstream of the pressure regulators of the individual tanks was used to give precise control over the pressure level of the fuel at the fuel flowmeters. This fuel pressure was measured with a precision mercury manometer having an accuracy of $\pm 0.06\%$ of reading as given in Section 8.1.1 Calibration.

8.3 OXIDES OF NITROGEN MEASUREMENTS

In order to prevent the potential loss of NO_2 in the water which is condensed from nonheated samples, a heated sample was used for oxides of nitrogen analysis. A vacuum reaction chamber type chemiluminescent NO_x analyzer manufactured by Thermo Electron (Model 10 A rack mounted) was used to make the NO and NO_x measurements. The sample was kept above the dew point up to the heated capillaries. To avoid potential H_2 interference in the NO_2 -to- NO converter (Siewart, 1975) an air bleed was factory-installed upstream of the converter.

8.4 SOLID STATE LOGIC IGNITION INTERRUPTER CIRCUIT (SSLII)

To determine the effects of flame initiation- or ignition-failure (misfire) on hydrogen-fueled engine performance, during some of the tests the ignition was intentionally interrupted every Yth time a spark plug should have been activated. The circuit of the apparatus used to interrupt the ignition is illustrated

in Figure 6. With this circuit, Y can take on values of 10, 20, 50 or 100 resulting in corresponding simulated flame initiation- or ignition-failure of 10%, 5%, 2% or 1% of the time.

8.5 HYDROGEN PEROXIDE EMISSIONS

It has been previously reported that hydrogen-fueled internal combustion engines emit hydrogen peroxide as a potentially hazardous component of their exhaust gases (Griffith, 1974). For this reason it was decided that an accurate analysis should be carried out for the detection of hydrogen peroxide in the exhaust gases. A standard KMnO_4 titration was initially used for the analysis (Koltoff, 1952). Under moderate loads no hydrogen peroxide was detected. Under moderately-heavy loads the KMnO_4 method of analysis revealed positive results. Due to the relatively high concentration of nitric oxide levels under these conditions, it was necessary to determine the validity of the method used. A perusal of the literature revealed the possibility of interference by nitric oxide (Dennis, 1913). Approximately thirty cubic feet of nitric oxide gas was bubbled through the 0.1N KMnO_4 solution used as the analytical reagent. Within seconds a positive reaction could be observed with the naked eye. It was thus decided to search for an alternate method of analysis due to the nitric oxide interference. A strong reaction between hydrogen peroxide and phenol-ferrous sulfate solution was found to be quite unreactive towards nitric oxide (Welcher, 1947, Spiro, 1915, Chemical Abstracts, 1915). Testing showed the hydrogen peroxide/phenol-ferrous sulfate (PFS) solution reaction method to be an acceptable one for hydrogen peroxide analysis.

In order to collect a sample of the exhaust gas for analysis a hole was drilled into the exhaust pipe ten feet downstream from the exhaust manifold (refer to Figure 7). A Swagelock fitting was brazed to the opening so that a six inch length of 1/4 in. O.D. stainless-steel tubing could be attached. Two and a half feet of teflon tubing (1/4 in. O.D.) was connected to the stainless-steel tubing by means of a Swagelock fitting. The teflon tubing was used to provide maximum flexibility of movement during sample collection and to provide an inert tube wall surface to guard

against reduction of any hydrogen peroxide that may be present in the exhaust gas. The instability of hydrogen peroxide at high temperatures is only too well known and the possibility of any reduction (oxidation-reduction) reactions during collection could greatly affect the validity of the results.

The teflon tubing was connected to a one-liter suction flask fitted with a rubber stopper; a hole was drilled into the stopper and the teflon tubing placed through it into the collection flask. The nipple of the suction flask was connected to a sample pump by a two-foot section of tygon tubing. Before a sample of exhaust gas was collected, a 200 ml volume of freshly-prepared phenol-ferrous sulfate reagent was placed into the flask and the stopper was replaced with its teflon tubing protruding well below the level of the reagent. When the sample pump was turned on it withdrew a sample of exhaust gases from the exhaust pipe via the stainless-steel and teflon tubing and through the reagent in the flask. This method is usually referred to as "scrubbing". The collecting flask was kept in an ice-bath to insure maximum sample collection. After the sample was scrubbed it was forced via the sample pump through a rotameter to an exhaust fan. The rotameter was used to measure the flowrate of the exhaust sample collected at each data point. A flowmeter reading of 50 corresponds to 0.605 cubic feet per minute. Calibration of the rotameter was accomplished by means of a positive displacement method using a beaker submerged in water and a stopwatch. Each sample was collected for twenty minutes making a total sample size of $12.1 \pm 1.8 \text{ ft}^3$ of gas at standard temperature and pressure. (70°F, 14.7 psi).

Samples were analyzed quantitatively by using hydrogen peroxide of known concentration and titrating a portion of the reagent to the same visually-observed color as that of the sample. Photograph 7 shows the color of the PFS solution after having scrubbed samples having the values of ϕ indicated.

8.6 IN-CYLINDER PRESSURE-TIME DIAGRAM

A Kistler 640-601B Transducer was used to measure in-cylinder, pressure-time diagrams. Ancilliary equipment include a Kistler 640-N54R spark plug adapter, a Kistler 121M15 cable and a Kistler 503D charge amplifier. The amplifier output was stored on

a dual-trace storage scope and permanently recorded on Polaroid photographs.

Although the extension passage of the spark-plug adapter serves to protect the transducer from peak combustion temperatures, the transducer was coated with a silicone protective coating (General Electric RTV 116 Silicone Sealant). This precaution was taken because of concern for the small quench distance of the hydrogen flame.

The transducer was calibrated in combination with the 503D amplifier using two separate high-precision pressure gauges as reference. The calibration of the transducer-amplifier pair was found to correspond to the factory calibration of the transducer within 0.7%.

8.7 SUMMARY OF MEASUREMENTS

Table 6 contains a listing of the measuring instrumentation and equipment used including items described above and others for measuring torque, temperature, engine speed, barometric pressure, etc. Also included in Table 6 are: the manufacturer's precision (repeatability or reproducibility of instrument output for a fixed input) claim; the estimated accuracy (difference between average instrument reading and true value of a given input) based on calibrations performed by the authors and/or other agents; scale readability of instruments having an analog output (scale readability is the observer's opinion as to how close he can read the analog scale). For some instruments readability is separated from manufacturer's precision claim because it was found that their instrument scales could not be read to the degree of precision claimed by the manufacturer. An estimate of the total instrument error is given, which is the sum of the manufacturer's precision claim, accuracy, and readability. As will be seen in later sections when the data is presented, the scatter in the data, in general, is found to fall within the estimated total instrument error. There are some exceptions where there is a much larger scatter apparent in a data plot resulting from the inability to control the dependent variables. For example, when NO_x emissions are plotted as a function of brake mean effective pressure, bmep, the scatter in the emission level is found to

fall within the estimated total NO_x instrument error. When NO_x emissions are plotted as a function of fuel-air equivalence ratio, ϕ , however, the scatter in the NO_x emission level is much greater. The reason for this large scatter is that NO_x is a strong function of ϕ as well as spark advance, SA, and slight variations in either SA or ϕ cause large variations in NO_x . The smaller scatter in the NO_x emissions when bmep is used as the dependent variable is attributed to: (1) the weaker dependence of NO_x emissions on bmep; (2) the fact that it is easier to control the bmep level than it is to control the ϕ level. Thus, slight uncertainties in ϕ result in large apparent changes in NO_x with similar uncertainty levels in bmep resulting in small apparent changes in NO_x .

9. TESTING PROGRAM

In Section 9.1 the testing program is described in terms of the various combinations of experiments that were performed. Three engine speeds were selected at which the bulk of the characterization experiments were performed: viz. 800, 1800, and 3300 RPM. The number of tested engine speeds was limited to three because engine speed is not an important variable for the characteristics being sought and the data collected at the speeds indicated yielded a sufficiently-comprehensive data set. Also, the time saved by not testing at other engine speeds was better spent establishing estimates of experimental uncertainty and investigating unexpected phenomena found during the tests (e.g. the presence of higher-than-expected concentrations of hydrogen in the exhaust, Section 11.1).

With the above considerations in mind the test program was formulated. It yielded, with a minimum of data, the information necessary to determine performance and emissions characteristics of the Table 1 potential engine configuration 1 - 6.

9.1 DESCRIPTION OF EXPERIMENTS

In Table 7 the major experiments performed using the Table 1 Pre IVC engines 1 - 6 are delineated.* There are a total of 14 basic experiments (Column 1). Testing was done at various engine speeds (Column 2) and loads (Column 3). EGR rates (Column 4) and water injection rates (Column 5) were also varied. Using the Solid State Logic Ignition Interrupter circuit described in Section 8.4, some tests (Experiment number 2, 6) were performed during which the spark was intentionally not fired at a rate ranging from 0 to 10% of the cycles.

During all the tests the measurements given in Table 8 and labeled a - l were made. In addition, during tests numbered 2 and 12, Table 8 measurements m through q were made. During essentially all of the tests, cylinder-to-cylinder fuel-air ratio distribution measurements were made for the engine configuration given in the next to the last column. The last column of Table 7 identified the engine configurations that were used for the given experiments.

* Other experiments were performed as deemed necessary during the program.

During the experiments given in Table 7, the equivalence ratio was near stoichiometric for the throttled engine (Engine Configuration No. 1). A limited but sufficient number of experiments were also performed during which the equivalence ratio was varied for the throttled mode of operation to yield a comprehensive brake thermal efficiency, BTE, and NO_x emission map at 1800 RPM.

9.2 FUELS

Welding-grade hydrogen (99.9% pure) was used for the bulk of the experiments. A few tests were carried out using ultra-high purity hydrogen (99.999 and 99.995% pure, See Section 11.1.1).

9.3 ENGINE BREAK-IN

Since breaking-in of the engine on hydrogen was considered to be too costly, original plans called for breaking-in the engine using LPG. Subsequent investigation revealed that the break-in time using LPG was too long. It was thus decided to break-in the engine on gasoline for 15 hours and then to disassemble it and to clean it before running on hydrogen. The break-in schedule using gasoline as fuel, as described in Table 9, was followed. Discussions regarding break-in of a hydrogen engine brought up the possibility of using special break-in oils if and when hydrogen engines become used on a wide-scale basis.

9.4 TEST PROCEDURES

In the following sections the test procedures for each test are outlined. For a description of the accuracies of the instruments and calibration procedures see Table 6.

9.4.1 Temperature Measurements

The instrumentation used for temperature measurements consisted of a Fluke Model 8600A Digital Multimeter (DVM), shielded chromel-alumel thermocouples, a selector switch and a certified thermometer.

The temperature of the room in which the instrumentation was housed was first checked and adjusted to insure proper temperature of the instrumentation. The water flowrate for the heat exchanger and the ice bath for the thermocouple reference junction were set one hour prior to testing to stabilize their respective

temperatures. The DVM was then checked for zero. After the ice bath stabilized all engine thermocouples were compared to each other and to the ambient temperature. Ambient air and hydrogen thermocouples were compared to the calibrated-thermometer reading. The temperature of the exhaust gas condenser was also checked for proper operation of the refrigeration system.

9.4.2 Air Flowrate Measurements

Air flowrate measurements (See Section 8.1.2) were made with a Meriam Laminar Flow Element (LFE) and a Meriam certified precision water manometer in conjunction with a calibrated barometer, thermocouple and Psychro-Dyne PPl00 humidity test kit (thermometers calibrated).

The LFE air filter dust cover was removed and its manometer zeroed. Barometric pressure and ambient temperature were read and recorded. The relative humidity was measured by the wet bulb method via the Psychro-Dyne test kit and recorded. Manufacturer-supplied dry-air flowrate calibration charts were used and appropriate corrections made to account for variations in humidity, barometric pressure and ambient temperature.

9.4.3 Fuel Flowrate Measurements

Fuel flowrate was measured with a set of three Cox 18 inch rotameters certified at Colorado Experimental Engineering Station. A precision mercury manometer calibrated with a certified standard was used to monitor the calibration pressure of 25 psia at the rotameter exit (See Section 8.1.1).

The pressures in the hydrogen manifolds were checked prior to each test. Empty tanks were replaced and the fittings leak tested with Snoop*. The mercury manometer was adjusted to account for variations in ambient temperature; a local-gravity adjustment had been made previously. The manometer was then zeroed. Subsequently, the manometer indicator was set at 25 psia minus barometric pressure. With this adjustment, the indicator was positioned such that when the hydrogen pressure at the rotameter exit was 25 psia (the calibration pressure) the indicator would be

* Liquid leak detector 1976 MarRad Service Co., Nupro Co., Willoughby, Oh. 44094.

nulled. During testing this null position was maintained by a pressure regulator. The entire fuel-delivery system was pressurized and given a final check for leakdown.

9.4.4 Measurements of NO_x Concentration in Exhaust Gas

The concentration of NO_x in the exhaust gas was measured with a Thermo Electron 10A analyzer with heated capillaries. The data was recorded on a Hewlett Packard Moseley 7100B Strip Chart Recorder.

The DC amplifier section of the analyzer was turned on at least one day before testing to minimize drift. The NO_x converter and chart recorder were turned on one-half hour before testing. A flow of gas (room air) was passed through the analyzer during this period. A clean sample gas filter was installed at this time. The analyzer was zeroed using bottled zero N_2 and calibrated using a span gas (NO in nitrogen) of known concentration, on the order of 9500 ppm. The chart recorder used to record the NO_x readings was zeroed and calibrated during this period.

During testing, the analyzer was spanned and zeroed periodically. Heated lines were used to prevent the water vapor in the sample from condensing. These lines had gas passing through them continuously. During sampling the flowrate of sample gas was set equal to the flowrate of the calibration gas used to calibrate the analyzer.

9.4.5 Measurements of O_2 Concentration in Exhaust Gas

The concentration of O_2 in the exhaust gas was measured with a Beckman, OM-11 analyzer. The data was recorded on a Hewlett Packard Moseley 7100B Strip Chart Recorder.

The DC amplifier section of the analyzer was turned on at least one day before testing to minimize drift.

The O_2 sensor and chart recorder were turned on one-half hour before testing. A flow of gas (room air) was passed through the analyzer during this period. A clean sample gas filter was installed at this time. The analyzer was zeroed using prepurified (99.998%) nitrogen and calibrated on room air as per the operating manual. The chart recorder used to record O_2 readings was zeroed and calibrated during this period. The analyzer was

zeroed and spanned periodically during testing.

Water was found to collect in the sampling lines during the interim between test days. This water was blown out before any sample gases were drawn into the instruments to prevent any water droplets from entering the O₂ analyzer. A condenser was used to remove water vapor from the samples gases before they entered the O₂ analyzer. This condenser was emptied approximately every 45 minutes. During sampling the flowrate of sample gas was set equal to the flowrate of room air used to calibrate the analyzer.

9.4.6 Torque Measurement

A Buck Engineering, Go Power Model DA 312 water-brake dynamometer was used for measuring engine torque output. The amount of torque absorbed by the dynamometer was varied by varying the flowrate of water through the dynamometer. The dynamometer output was one psi of load cell oil pressure for every one ft lb of torque. The oil pressure was measured with a Heise oil pressure gauge. The oil pressure gauge was zeroed before each testing period by completely unloading the dynamometer. After zeroing the pressure gauge, the manufacturer-supplied calibration arm and weight were attached to the dynamometer to apply a known static torque.

9.4.7 RPM Measurement

The RPM was measured by matching the engine RPM to a strobotac. The strobotac was calibrated internally using local utility power at 60 cycles/sec (3600 RPM) and 30 cycles/sec (1800 RPM). Once calibration at 1800 RPM was complete, the dial on the strobotac was not moved. Small deviations from 1800 RPM could be ascertained by counting the apparent number of crankshaft pulley rotations per minute.

9.4.8 Some Procedures During Testing

After starting the engine, sufficient time was allowed for the engine to warm up to the point where the coolant water temperature was 185°F.

When the engine reached operating temperature, the torque was adjusted by varying the water flowrate through the dynamometer.

Fuel flow was varied by varying the hydrogen delivery pressure. Air flow was varied by throttling the inlet air with the Beam or Impco gas mixer throttle plate. The spark advance mechanisms within the distributor was locked in a single position and the spark advance was varied by rotating the distributor. During testing, to isolate the effects of EGR and H₂O injection rate on engine performance at constant load, the flowrate of water through the dynamometer was left untouched.

10. EXPERIMENTAL RESULTS

During the first year of the project. Table 1 Pre IVC engine configurations 1 - 6 were characterized by performing the experiments delineated in Table 3. These results, along with the computer program which was used to reduce the raw data to convenient and/or meaningful forms are given in Appendix 7 which is described below. The reduced data, presented in tabular form in Appendix 7, is presented in graphical form, as needed in Section 11.

Appendix 7 commences with definitions of Fortran symbols (A7.1) used in the data-reduction Fortran computer program (A7.2). The program was used to facilitate reduction of the data and to put the reduced data into tabular form ready for publication. Input to the program were measured variables such as Torque, RPM, NO_x and O_2 emission levels, volumetric flowrates of air and hydrogen atmospheric temperature, pressure and so forth for each data point; output includes Φ , bmep, BTE and so forth.

In Appendix 7.3 the definitions of the symbols used in the experimental results tables (A7.4) are given including units.

The experimental results presented in Appendix 7.4 are grouped into sets of data points; each set corresponds to one of the Table 7 experiments; individual data points are assigned and identification number (TEST NO.). For example, the first set of data is made of five data points (TEST NO. 1-5); this set comprises the data taken for Table 7 Experiment Number 1 (EXP 1) with Table 1 engine configuration No. 1 (ENG 1). The second set of data (TEST NO. 6-10) comprises the data for Table 7 Experiment Number 2 (EXP 2) with Table 1 engine configuration No. 4 (ENG 4), etc. There are some data points for the ENG, EXP row that have been labeled EXTRA. These points are extra data in that they were not originally planned to be taken and are therefore not included in Table 7. The need for taking these points was discovered during the first year testing to help explain unexpected observations.

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11. DISCUSSION OF RESULTS

In the following sections the results of the experiments are discussed, some comparisons with theory are made, the results are compared with those obtained by other hydrogen-fueled engine researchers, and some comparisons are made with gasoline-fueled operation.

11.1 HYDROGEN IN THE EXHAUST AND H_2 UTILIZATION COEFFICIENT

During the experiments, when the Section 8.4 SSLII was used to cause intentional misfiring of the engine (Table 7 Experiments Number 2 and 6) excess oxygen was observed in the exhaust and, as is discussed below, there was excellent agreement between the increased oxygen concentration measured and the concentration calculated assuming that none of the hydrogen fuel in the intentionally-misfired cylinder(s) was oxidized. This agreement indicates that there was no significant hydrogen oxidation interference at the silver anode of the polarographic oxygen analyzer sensor. The ability of the hydrogen fuel to survive a none-spark-ignited engine cycle and flow through the exhaust system while mixing with exhaust gases from other fired cylinders and not oxidize, suggests that unburned hydrogen might survive a fired engine cycle. This survival during normal firing is not expected by some hydrogen engine researchers: for example, when some researchers calculate fuel-air ratios for normal-operation tests (viz: BEC, 1977; Strelau, 1976) they assume that there is no hydrogen in the exhaust. During the present study, however, unburned hydrogen was found to exist in the exhaust during normal operation. The following discussion concerns the methods used to determine the hydrogen concentration in the exhaust and the results obtained during normal operation and during SSLII-induced misfire operation.

11.1.1 Detection of Hydrogen in the Exhaust

The concentration of unburned hydrogen in the exhaust was determined by a mass balance using as known quantities the hydrogen and air (including humidity) mass flowrates as determined by their respective metering devices (Section 8.1) and the measured oxygen concentration in the exhaust gas (See Appendix 9 for the calculation procedure). An exhaust-gas sample for the unthrottled

engine at $\phi^* = 0.27$ and one at $\phi = 0.33$ were also analyzed by a mass spectrometer to confirm the presence of hydrogen as determined from the mass balance. The semi-quantitative results of the mass spectrometer analysis were in good agreement with those determined from the mass balance. This agreement, plus the excellent correlation between the exhaust-gas hydrogen concentration predicted for the SSLII-induced misfire tests and the concentration determined from the mass-balance calculation, give confidence to the conclusion that hydrogen is present in the exhaust of a normally-fired hydrogen-fueled engine. If this conclusion is true then fuel-air ratio determinations made by researchers who rely solely on measurements of oxygen in the exhaust and not on measurement of fuel and air flowrates are in error. This error causes them to report a value of ϕ smaller (leaner) than the actual value.

Since this conclusion will affect the results and conclusions drawn by others, a search for potential errors that could have occurred during the present study was made. One potential error was that of hydrogen oxidation in the polarographic oxygen analyzer sensor, but, as discussed above, this error did not exist. Another potential, but unlikely, source of error could have been the presence of oxygen in the hydrogen fuel. If oxygen were present, the results pertaining to the existence of hydrogen in the exhaust gas would have been in error, but unburned fuel level trends would not have exhibited the observed characteristics. For completeness, however, tests were conducted using two samples of high-purity hydrogen. The manufacturer of the first sample claimed a purity of 99.995% and the second a purity of 99.999%. The results of the tests showed no measurable difference of the engine performance or emissions characteristics when the high-purity hydrogen fuel was used as compared to the 99.9% purity welding-grade hydrogen used for the bulk of the testing.

11.1.2 Experimental Results

The results obtained during the present study relating to the presence of hydrogen in the exhaust are presented in Table 10

* ϕ = Fuel-Air Equivalence Ratio ($\phi < 1$ is lean); See Section 16.

in terms of H_2 Utilization Coefficient (HU) which is defined by the following equation:

$$HU = \frac{\dot{m}_{H_2, \text{ Fuel}} - \dot{m}_{H_2, \text{ Exhaust}}}{\dot{m}_{H_2, \text{ Fuel}}} 100 \quad (11.1)$$

In Equation (11.1), $\dot{m}_{H_2, \text{ Fuel}}$ is the mass flowrate of hydrogen fuel supplied to the engine, and $\dot{m}_{H_2, \text{ Exhaust}}$ is the mass flowrate of hydrogen in the exhaust gas. The quantity $100 - HU$ is a measure of the percentage of the hydrogen fuel which ends up in the exhaust*.

In Table 10, the HU is given for both the throttled (Table 1 Engine Configuration 1) and unthrottled (Configuration 4) mode of operation for various degrees of SSLII-induced misfire (% Misfire). For the throttled engine, with 0% misfire, the fuel-air equivalence ratio is near stoichiometric ($\phi \sim 1.04$ at 1800 RPM and 1.05 at 820 RPM). Even in the case of 0% SSLII-induced misfire, the HU is less than 100% with substantial amounts of unburned hydrogen appear in the exhaust of the unthrottled engine when a lean mixture ($\phi = 0.18$) is used at the idle-condition engine speed of 820 RPM.

The HU in Table 10 decreases proportionately as the SSLII-induced misfire rate is increased. In order to exemplify this proportionality, Table 11 is presented where $\overline{HU}$ MEAS is given in place of HU. $\overline{HU}$ MEAS is a measured non-dimensional H_2 Utilization Coefficient defined by Equation (11.2):

$$\overline{HU} \text{ MEAS} = \frac{\text{Measured HU with SSLII-Induced Misfire}}{\text{Measured HU with Zero SSLII-Induced Misfire}} 100 \quad (11.2)$$

Also included in Table 11 is:

$$\overline{HU} \text{ PRED} = 100 - \% \text{ Misfire} \quad (11.3)$$

$\overline{HU}$ PRED is a non-dimensional H_2 Utilization Coefficient predicted for SSLII-induced misfire tests by assigning a value of 100 to the H_2 Utilization Coefficient with 0% SSLII-induced misfire (per Equation (11.2)) and assuming that the H_2 Utilization Coefficient decreases proportionately with increasing SSLII-induced misfire rate. As seen in Table 11 there is excellent agreement between the measured H_2 Utilization Coefficient during SSLII-induced misfire

*When $\phi > 1$ values of $HU < 1$ are due to deficiency of oxygen in the rich fuel-air mixture as well as to combustion inefficiency. (See Section 16)

($\overline{HU}$ MEAS) and that predicted ($\overline{HU}$ PRED). The difference between $\overline{HU}$ MEAS and $\overline{HU}$ PRED is well within experimental uncertainty except for the unthrottled engine at the operating condition of 820 RPM and a ϕ of 0.18. This operating condition, however, was found to be quite unstable. The instability is attributed to the extremely lean value of ϕ (close to the lean flammability limit), and is reflected by the very low values of HU, as shown in Table 10, as well as by the observation of incipient misfire during testing.

Two basic causes of lean-limit misfire in a gasoline-fueled engine have been delineated (Quader, 1976): (1) Ignition-failure induced misfire where the spark plug fails to ignite the lean mixture and the resulting cylinder-pressure/time diagram is equal to that of a motored cycle and (2) Partial-burn induced misfire where there is ignition of the charge but the flame speed is so low that there is insufficient time for it to propagate across the combustion chamber prior to the opening of the exhaust valve. Both of these causes of lean-limit misfire are reflected by unburned hydrocarbon fuel in the exhaust. During the testing of the unthrottled hydrogen-fueled engine at 820 RPM and a lean mixture ($\phi = 0.18$), insufficient motoring cycles were observed to account for the majority of the unburned hydrogen fuel in the exhaust. It is suspected that, for these tests, the unburned hydrogen appearing in the exhaust results from the low flame speed at the lean value of $\phi = 0.18$. It thus appears that hydrogen can appear in the exhaust as a result of a lean-limit misfire (motoring cycle) and/or the occurrence of low flame speeds at lean-mixture operation. (See also Section 11.2.3.)

It is of interest to note that the critical value of ϕ , operation below which will result in a significant decrease in HU, is around 0.26 for the present-study test engine. This value is consistent with the results of others who find it necessary to enrich the mixture and introduce partial throttling to maintain good efficiency at light loads as will be discussed in Section 11.4.

It is not believed that the excess oxygen in the exhaust is caused by poor trapping with the combustion-chamber/camshaft combination used in the present-study test engine because: (1) The excess oxygen was found to be unaffected by exhaust-system

back pressure up to 3 psi, and (2) The excess oxygen was present at high manifold vacuum.

The results of other investigators give support to the conclusion that higher-than-expected concentrations of hydrogen are present in the exhaust during normal operation of hydrogen-fueled engines.

1. In Figure 8 the oxygen concentration levels in the exhaust, as measured during the present study and during the work of Watson and Milkins (1975), are compared with the theoretical level calculated by a mass balance assuming complete combustion (no dissociation) with the measured hydrogen- and oxygen-mass flowrates being used as known quantities. The measured oxygen levels by both research groups are in agreement with both being in excess of the level predicted by the complete-combustion (no dissociation) theory. Therefore, the combustion is not complete, $HU < 100\%$, and there was free hydrogen in the exhaust.

2. King and Rand (1955) found it necessary to operate their CFR hydrogen engine with up to a 7%-rich mixture ($\phi = 1.07$) to achieve maximum power. This behavior strongly suggests the presence of free oxygen and therefore hydrogen in the exhaust at $\phi = 1$ and thus incomplete combustion ($HU < 100\%$) due to dissociation.

3. The unusual BTE versus ϕ characteristics presented by Strelau (1976) can also be explained if incomplete combustion at lean mixtures is assumed. Strelau's result showed an unexpected rise in efficiency for ϕ less than around 0.38. Strelau, however, used an orifice meter to measure hydrogen flowrates that had been calibrated by measuring oxygen in the exhaust and assuming complete combustion (no dissociation). If during his tests the HU was less than 100%, as found during the present study (Table 10), then the hydrogen flowrates he used to calculate BTE were too small, especially for lean mixtures where HU decreases significantly (Table 10). The unexpected non-linearity in Strelau's

orifice meter calibration might also be explained by allowing for the presence of hydrogen in the exhaust.

4. Homan (1977) reported excess oxygen in the exhaust of a premixed, spark-ignited, hydrogen-fueled CFR engine. In his work he assumed a HU of 98.5% in his calculations.

11.1.3 Engine Speed, Combustion-Chamber Shape and H_2 Utilization Coefficient

In Table 12 the effect of engine speed on HU is illustrated. The velocity of the gas in the combustion chamber near the spark plug, as measured by Noguchi et al (1976), is included. (See Nomenclature (Section 16) for definition of VCF used in Table 12.) It is observed that engine speed has a negligible effect on HU. This small effect is not unexpected for the throttled mode of operation because at the instant of ignition, which occurs near top dead center (See Section 11.3: Best Efficiency Spark Advance), the gas velocity in the combustion chamber is very small and does not vary significantly with engine speed (Noguchi et al, 1976). The engine speed is also found to have a negligible effect on HU for the unthrottled engine even though, as a result of the best efficiency spark advance crank angle changing (See Section 11.3), the gas velocity changes substantially at the instant of ignition.

The data in Table 12 show that fuel-air equivalence ratio and not engine speed (or combustion-gas velocity) is the dominant factor influencing HU. Therefore, if large gains are to be made in the HU of unthrottled hydrogen-fueled engines, it is important to first investigate methods such as charge stratification rather than combustion-chamber shape.

11.1.4 Charge Stratification and H_2 Utilization Coefficient

Charge stratification can lead to large gains in H_2 Utilization Coefficient (HU) whereas, based on the Table 12 effects of gas velocity on HU, much smaller gains will be realized from combustion-chamber redesign. The potential for increasing HU by charge stratification is illustrated in Table 13. The data for the

present-study Toyota engine with a homogeneous (carburetted) fresh charge is compared with the data for a Toyota engine utilizing a stratified charge (Swain and Adt, 1972; Swain, 1973). Both engines were operated in the unthrottled mode. The Table shows that the leanest mixture that could be utilized by the present-study engine was 7.2% by volume hydrogen ($\phi = 0.18$) and that the corresponding no-load (idle) engine speed was 820 RPM. For this condition the HU was low, having a value of 56.9%. The HU for the other Toyota engine was estimated by assuming the various values of volume cylinder filling VCF* shown in the Table (VCF was not measured for the testing of the other Toyota engine). The actual VCF was probably about 85% and the corresponding HU is 96.9%. To show the effect of changing the assumed value of VCF on the HU calculations, calculated HUs are shown at VCF=93.6% and VCF=75%. Even with the very low VCF of 75%, the HU is significantly greater than that found with the present-study, homogeneous-charge engine.

It is also of interest to note that the engine with the stratified charge was able to operate at a much lower engine speed than that of the homogeneous-charge engine. This ability is due to the fact that it could operate at a much leaner mixture.

From this section and the previous one, it appears that larger gains in HU will be realized if research efforts are channeled to developing stratified-charge hydrogen engines in lieu of attempting to develop better combustion-chamber shapes for uniform-charge engines. (The development of a stratified-charge hydrogen-fueled engine is felt to be an easier task than the development of one fueled by gasoline.)

11.1.5 Potential Need for H₂ Utilization Coefficient Less Than 100%

It should be pointed out that with regard to HU, it may be desirable to have some unburned hydrogen present in the exhaust acting as an oxygen consumer if reduction catalysts are to be used for NO_x control at high loads. (Tests are needed.)

* See Section 16 for definition of VCF.

11.2 FLASHBACK

In Section 3.3 five suspected causes of flashback were discussed:

1. Hot spot(s) (spark plug, exhaust valve, deposit).
2. Communication of fresh charge with exhaust gas during the valve-overlap period when engine operation is normal (no misfire).
3. Misfired cycles.
4. Still-burning gases from the previous cycle present in the combustion period when the intake valve opens at very lean mixture operation.
5. Oil in residual gas.

As described below, suspected causes 1-4 were investigated during the first year and preliminary conclusions regarding the existence of cause 5 were made.

11.2.1 Hot Spots

Engine Behavior - Hot spots have been known to be a cause of flashback for some time (King, 1955). The chain of events leading up to hot spot-induced flashback became a topic of investigation during the present study. Flashback caused by hot spots is a pre-ignition-related phenomenon. This type of flashback is preceded by progressively-earlier preignition of the charge as illustrated qualitatively in Figure 9. This figure illustrates the chain of events occurring before hot spot-induced flashback in terms of typical pressure-rise curves on a cylinder-pressure/time diagram during the period before the onset of audible flashback. During this period preceding hot spot-induced flashback (Period PHIF), ignition of the charge results from the high temperature of the hot spot rather than from the electrical discharge of the spark plug. Curve 1 is a normal cycle with the charge being ignited by the spark plug (after the intake valve closes). The presence of a hot spot results in the charge being ignited earlier, before spark ignition (curve 2). This earlier ignition causes the hot spot to increase in temperature resulting in earlier and earlier ignition (curves 3 and 4). Finally the hot spot causes the charge to ignite before the intake valve closes (curve 5) resulting in

propagation of the flame into the intake manifold with subsequent flashback. The above-described process of progressively-earlier ignition is evidently an unstable one because early ignition by the hot spot increases the heat load (combustion gas heat rejection rate) which raises the temperature of the hot spot thereby leading to earlier hot spot-induced ignition in the next cycle as illustrated in Figure 9.

The duration of the Period PHIF is very short when water injection is not used; for an engine speed of 1800 RPM the duration was about 5 cycles or one-third of a second. Once the Period PHIF begins in an engine operating without water injection, it is impossible for the operator to react quickly enough to prevent flashback.

When water injection is used, the duration of the Period PHIF is extended. Water injection rates of 5.1 pounds of water per pound of hydrogen fuel extended the Period PHIF considerably: it was possible to operate the engine for several seconds while the Period PHIF existed in the cylinder with a rich ϕ ; there was sufficient time for the operator to reduce ϕ and eliminate flashback.

Engine behavior during the Period PHIF, as is illustrated qualitatively in Figure 9, was found to be very repeatable: the NO_x emissions level was found to progressively increase at a rapid rate and the engine speed was found to decrease. The increase in NO_x emission level is attributed to the progressively earlier ignition by the hot spot. The drop in engine speed is attributed to the progressive deviation of the ignition point from the best efficiency spark advance (BESA).

Hot Spot Types - Two types of hot spots were found during the first year work:

1. Oil deposits on the electrode of high operating temperature (high heat range) spark plugs.
2. Oil deposits on the exhaust valve.

During the tests, spark plugs of varying heat ranges were used and they developed varying degrees of deposit formation. As stated above, the behavior of the engine was very repeatable when hot spot-induced flashback occurred. The behavior was so repeat-

able that it was possible to select a spark plug of a given heat range and deposit level and have the flashback limit be repeatable within ± 1 ft lb of torque during the unthrottled mode of operation.

It was also found that there was no significant difference in the flashback-limited load level when plugs of different heat ranges were used provided they all had the standard grounded electrode design (Obert, 1973). The flashback-limited load level was increased, however, when retracted plugs (Obert, 1973) were used. Changing the heat range of the retracted plug, however, did not significantly change the flashback-limited load level. It was also found that when retracted plugs were used the flashback-limited load was about 45 ft lb torque (70 psi bmep) regardless of the deposits.

Loads above 45 ft lb torque could not be achieved (without EGR or water injection) because the oil deposits on the sodium-filled exhaust valves became hot spots which induced flashback. For results comparing sodium- and lithium-filled valves see Appendix 10.

11.2.2 Valve-Overlap

Flashback behavior of the engine with zero-overlap Camshaft C4 was found to be worse than that with Camshaft C1; the tests showed that this zero-overlap camshaft reduced the peak load at which both the throttled and unthrottled engine could operate without flashback. This result strongly suggests that the hypothesis that flashback can be caused by communication of the fresh charge with still-burning exhaust gases in the exhaust port during valve overlap is not a valid one.

11.2.3 Misfire

To study the effect of nonignition on flashback, tests were conducted during which the SSLII (Section 8.4) was used to induce 1, 2, 5 and 10% nonignition with the engine operating at loads up to those near the flashback limit. It was found that nonignition does not cause flashback. The engine could be operated for hours with intentional nonignition and no flashback occurred. It is thus concluded that nonignition is not a source of flashback. (It is

possible, however, that long ignition delays could be a source of flashback and this characteristic will be studied at a future date.)

11.2.4 Lean Mixture

Original plans called for the gathering of no load (idle) characteristics at an engine speed of 800 RPM. It was found, however, that the unthrottled-engine (Table 1, Engine Configuration No. 4) with the carburetted, and therefore pre-mixed, hydrogen-air charge could not operate at lower than 820 RPM. At 820 RPM the ϕ was about 0.18. To lower the engine speed below 820 RPM in the unthrottled mode it would have been necessary to decrease ϕ below 0.18. When this enleanment was attempted, however, flashback commenced. This flashback is attributed to the presence of still-burning gases from the previous cycle when the intake valve opens resulting from the very low flame speeds at lean-mixture operation. The very low flame speeds are reflected by the very large values of best efficiency spark advance (BESA) for lean mixtures (See Section 11.3.2). For example, at 820 RPM and $\phi = 0.18$ the BESA was 94° before top dead center. Also, the cylinder-pressure time oscilloscope traces during lean operation ($\phi = 0.18$) were very unstable.

The behavior of the engine during lean operation-induced flashback differed from the behavior during hot spot-induced flashback (Section 11.7.1). The engine did not exhibit a Period PHIF: there was not progressively earlier ignition and the NO_x emission level did not exhibit an increase.

11.2.5 Oil in Residual Gas

The influence of oil in residual gases on flashback will be eventually studied in detail. Some preliminary observations concerning residuals and the presence of oil in residuals, however, have been made as discussed below.

The role played by residual gases in causing early ignition was demonstrated during the testing described by Swain (1973): He found that with a compression ratio of 9.0:1, short duration, full-load runs could be made without flashback if extreme care was taken to clean the oil deposits from the exhaust valves when

properly-designed and -seated oil rings were used. When the compression ratio was lowered to 7.2:1, the short duration tests at full load without flashback could not be performed. It is well known that lowering compression ratio increases exhaust-gas temperature and thus residual-gas temperature (Taylor and Taylor, 1966). The observed inability to operate without flashback at the lower compression ratio is attributed to the increased residual-gas temperature, even though it is known that exhaust-valve temperatures increased.

Another observation made during those tests (Swain, 1973) was that the presence of oil in the residual gases enhanced the onset of early ignition. This enhancement was demonstrated by the fact that one of the cylinders commenced to flashback before the others. The piston for this cylinder was modified in such a way as to increase the oil content of the residuals. The piston modification, which involved removal of a portion of the land above the top compression ring, is shown in Figure 10 where it is seen that there is probably a greater chance for the vortex formed by the piston motion (Tabaczynski, et al, 1970) to entrain oil and distribute it into the residual gas of the cylinder with the modified piston. The fact that this cylinder commenced to flashback first lends support to the proposition that oil in the residuals enhances flashback.

There are two potential ways that the presence of oil in the residuals can enhance flashback: (1) the oil particles serve as nucleation sites; (2) the oil reacts with the oxygen in the combustion gases causing local high temperatures in the residuals. The oil serving as nucleation sites is consistent with other experiments during which City-of-Miami water ingested with intake air was found to produce severe flashback. When evaporated the water was found to leave a solid residue which was assumed to have formed nucleation sites for early ignition and flashback.

From the above, it is suspected that to avoid flashback, hydrogen engines must be designed to minimize the presence of oil in the combustion chamber. In addition, it may be necessary to utilize break-in schedules with new or rebuilt hydrogen engines that will minimize the presence of oil in residual gases.

Semi-quantitative insight into the effect of break-in and oil control on flashback has been obtained from the present program: In Table 14 three hydrogen engines tested by the authors are described. From the viewpoint of flashback, Engine III operated with the least-favorable camshaft design, coolant temperature, exhaust-valve type, and spark-plug type. In spite of these unfavorable conditions, Engine III was best able of the three to sustain operation without flashback. This ability is reflected by the fact that Engine III, which is the one constructed for the present project, was able to operate at a load of 60 psi brake mean effective pressure (bmep) for 15 minutes without flashback. Engine I could be operated without flashback at a bmep of only 50 psi. Engine II would not operate at all without occasional flashback. It should be noted that all of the engines could operate at higher than 60 psi bmep but with varying degrees of flashback.

The above-described relative flashback behavior of the three engines is consistent with the hypothesis that flashback can be caused by the presence of oil droplets in the residual gases. Engine I was broken-in for about 3 hours with gasoline; Engine II had no gasoline-fueled break-in period; and Engine III was broken-in for 15 hours with gasoline. The ability of Engine III to operate best without flashback is attributed to careful break-in and the subsequent improvement in oil control. In short: dependent on ring design and extent of ring seating, oil droplets are contained in the residuals (their deposits can be found on the spark-plug porcelin and on exhaust valves); these oil droplets can cause flashback if they are of sufficient size as to not be completely oxidized or cooled during the exhaust stroke. Additional testing will be carried out to give a further understanding of the effect of residuals and oil entrainment on flashback.

The behavior of the engine when flashback occurred apparently due to oil in the residual gas differed from the behavior observed during hot spot-induced flashback (Section 11.2.1) in that there was no sudden increase in the NO_x emission level before flashback. In addition, the onset of flashback suspected of being caused by oil in the residual gas was not predictable, it would occur in an

apparently random manner. More testing will have to be performed to gain a better understanding of this apparent source of flashback.

11.3 SPARK ADVANCE

The bulk of the testing was performed at best efficiency spark advance (BESA). In this section the method used to determine BESA is described. It is important to know the significance of the term BESA and how the BESA was determined because spark advance (SA) can have a strong influence on NO_x emissions as will be seen in Section 11.5.

11.3.1 BESA Determination

The method used to determine BESA was to set the SA to near the BESA value and adjust the engine speed, torque and ϕ to the appropriate values. The SA was then varied and the engine speed was monitored with the stroboscope and BESA was defined as the SA which yielded the highest speed. Table 15 shows the results obtained from a typical BESA-determination test. There it is seen that the engine speed is a very sensitive indicator of the spark advance at which the best efficiency occurs: The maximum RPM occurs at about 14.5 deg BTDC; at this crank angle the highest of best brake thermal efficiency also occurs, being approximately equal to 26.3%; thus, the BESA is about 14.5 deg BTDC. This method of using RPM to ascertain BESA is convenient and it is estimated that the thus-determined BESA value is within ± 2 deg of the true value. It should be pointed out that the stroboscope could measure changes in engine speed by as small as ± 1 RPM. It is of interest to note in Table 15 the large change in NO_x emissions as the SA is varied; a characteristic to be considered again in Section 11.5.

11.3.2 BESA Results - Throttled vs. Unthrottled Operation

Figure 11 shows the BESA as a function of ϕ for the unthrottled engine* and as a function of non-dimensional VCF for the throttled engine* both at 1800 RPM. In the Figure,

$$\overline{\text{VCF}} = \frac{\text{VCF}}{(\text{VCF})_{\text{max}}} \quad (11.4)$$

*In this and the following sections the term unthrottled engine refers to Table 1 Engine Configuration No. 1 and the throttled engine to Table 1 Engine Configuration No. 4. There is no EGR or water injection unless otherwise specified.

where $(VCF)_{\max}$ is the maximum VCF** obtained with the engine operated at 1800 RPM and fitted with the Beam gas-mixer (carburetor) described in Section 7.4; in other words $(VCF)_{\max}$ is the wide-open-throttled VCF at 1800 RPM with the Section 7.4 Beam carburetor.

For the throttled engine the BESA=0 deg (i.e. top dead center) over the entire range of VCF tested. This result is attributed to the very high flame speeds in the throttled engine which was operated at $\phi \approx 1$.

For the unthrottled engine the BESA is seen to vary over a wide range. As the mixture is made leaner the flame speeds decrease and greater SA is needed to attain best efficiency. Two data points are shown giving the BESA for the unthrottled engine at $\phi = 0.290$ and $\phi = 0.435$ at 3300 RPM. The former point shows a decreasing BESA with increasing engine speed; the latter point shows the opposite engine speed effect. The reason for this opposite behavior is not known for certain. Perhaps: (1) the turbulence level is the dominant mechanism governing combustion chamber flame speed for the $\phi = 0.29$ point with the flame speed increasing at a greater rate than the engine speed and; (2) the laminar flame speed of the mixture is the dominant mechanism governing combustion chamber flame speed for the other point. If this supposition is true then: as the engine speed, and therefore the turbulence level, increases for the $\phi = 0.29$ point, the flame speed will increase thereby resulting in lower values of BESA; for the $\phi = 0.44$ point, the increasing engine speed and turbulence level will not significantly increase the ϕ -controlled combustion-chamber flame speed and thus more advance is needed to achieve BESA.

11.3.3 Factors Controlling BESA

The BESA is a strong function of ϕ and a weak function of throttle setting (i.e. VCF or manifold vacuum). These relative functional dependencies are shown in Figure 12. There the BESA is shown as a function of load (bmep) at 1800 RPM for unthrottled operation at the values of ϕ indicated. In addition, there are

** VCF is defined in Section 16.

four data points at zero bmep. These four points are connected to the unthrottled data points having the same value of ϕ . Movement along these lines from the zero bmep condition to the right approximately corresponds to opening the throttle at constant ϕ . As observed, the BESA does not change significantly as the throttle position is changed at constant ϕ . Contrarily, the BESA changes are quite large as ϕ is changed at constant throttle setting.

11.4 BRAKE THERMAL EFFICIENCY

Figure 13 illustrates the brake thermal efficiency (BTE) data obtained from the throttled engine and the unthrottled engine as a function of load (brake mean effective pressure, bmep). The throttled engine was operated at $\phi \approx 1.0$; for the unthrottled engine ϕ was varied and the values can be found in the Appendix 7 tables. The spark advance was BESA and, except for the four data points indicated, the engine speed was 1800 RPM. For both engines the load was limited by the flashback limits shown in the Figure. There are ten data points shown for the unthrottled engine at a bmep of 51.5 psi. These points were collected on different days and their close proximity to each other illustrates the high precision of the present-study BTE measurements.

As expected from previous studies (Escher, 1975), the unthrottled mode of operation yields higher efficiencies due to the use of lean mixtures and the minimization of pumping losses. The throttled engine, which was operated at $\phi \approx 1.0$ for the Figure 13 data, can be operated at leaner mixtures and yield higher efficiencies than those given in the Figure as will be discussed in Section 11.6.

At a ϕ of around 0.26 the BTE of the unthrottled engine is seen to decrease rapidly as the mixture is made leaner. A higher BTE can be maintained if, instead of reducing ϕ below 0.26, the load is decreased by employing partial throttling. This effect, which was predicted by the authors (Swain, Adt, 1974), and demonstrated recently (Pischinger and Schaffrath, 1976) is illustrated in Figure 13 where the dashed curve at $\phi < 0.26$ represents data taken with partial throttling. The limiting value of ϕ of 0.26, operation below which necessitated partial throttling to increase

BTE at constant bmep, corresponds to the ϕ where operation below which was found to drastically reduce HU. Pischinger and Schaffrath found the limiting value of ϕ to be about 0.33. They also observed a bigger improvement when partial throttling was used at light loads than found in the present study. This observation suggests that perhaps their HU was lower than that of the present-study test engine in the range $0.26 < \phi < 0.33$. This lower HU, for $\phi < 0.33$, is suggested by the fact that their limiting ϕ was 0.33 compared to the present-study limiting value of 0.26.

Figure 13 also shows the effect of engine speed on BTE. The four data points at 3300 RPM all exhibit lower BTE than those taken at the same bmep and 1800 RPM. This decrease in BTE with increasing engine speed suggests that the negative effects on BTE caused by increasing bearing friction with increasing engine speed predominate over the positive effects on BTE caused by the reduction in heat transfer with increasing engine speed.

In Figure 14 the BTE is shown as a function of $\overline{VCF}$ for the throttled engine and as a function of ϕ for the unthrottled engine. In addition, the BTE that would be obtained if the HU were maintained at the value of 85% found at $\phi = 0.26$ (a possibility if the unthrottled engine is made to operate at $\phi < 0.26$ by incorporating some means of charge stratification (e.g. Swain, 1973)) is also shown.

11.5 OXIDES OF NITROGEN

Oxides of Nitrogen emission levels are shown in Figures 15a and 15b as a function of bmep for both the throttled engine (at $\phi \approx 1.0$) and the unthrottled engine. The engine speed was 1800 RPM except as indicated and BESA was used. Figures 16a and 16b show NO_x emissions as a function of ϕ for the unthrottled and $\overline{VCF}$ for the throttled engine. Consistent with past results (Escher, 1975), the emission level from the unthrottled engine is much lower than that from the throttled engine. This lower emission level is attributed to the lean mixture with the unthrottled engine and the consequent low, peak combustion-chamber temperatures. For the unthrottled engine the emission level remains very low (less

than 10 ppm) up to a bmep of about 55 psi ($\phi \approx 0.5$). The NO_x level increases approximately linearly with increasing bmep for the throttled engine.

In Figure 17 the NO_x emission level is shown as a function of ϕ for constant bmep of 51.4 psi and 0 psi (idle); the spark was maintained at the BESA valve and the engine speed at 1800 RPM. The NO_x emission level is seen to exhibit a maximum value at $\phi \approx 0.95$. This maximum is at a higher value of ϕ than found by others using single-cylinder CFR engines. They find peak NO_x levels at $\phi \approx 0.8$ (Parks and Stebar, 1976; Watson and Milkins, 1975; Strelau, 1976; Pischinger and Schaffrath, 1977). The higher value of ϕ for maximum NO_x emissions with the present-study test engine is attributed to the greater amount of internal EGR in the present-study engine. There was more internal EGR for the present-study engine because the peak-torque engine speed of the present-study engine is 3800 RPM; the corresponding speed for the CFR engine is around 1500 RPM. Thus the present-study test engine was operated for the Figure 17 data at an advanced valve timing resulting in more internal EGR than was present during the CFR tests. Also, the CFR engines were operated at wide-open throttle and the present-study engine at part throttle for the Figure 17 data thereby resulting in more internal EGR. The additional internal EGR occurring during the testing of the Figure 17 data causes the ϕ at which peak NO_x emission occur to be high because "EGR is most effective in lean mixtures" (DeBoer, 1974). In other words if an engine is operated without EGR, a typical mountain-shaped NO_x vs. ϕ curve will be found. If EGR is introduced the left ($\phi < 1$) side of the mountain will be lowered more than the right ($\phi > 1$) side thereby shifting the peak to the right, to a greater value of ϕ . An additional affect that could cause the NO_x emissions to peak at a ϕ larger than 0.8 is intermittent preignition in one of the three cylinders not monitored for pressure/time diagrams.

As mentioned in Section 11.3, SA has a very strong effect on NO_x emissions. This effect is shown in Figure 17 where NO_x emissions are shown for SA slightly different than BESA. For example, at $\phi = 0.86$, advancing the spark 4 deg from the BESA value results in a 15% or a 660 ppm increase in NO_x emissions.

BTE is much less influenced by SA. Thus, significant control over NO_x emissions without sacrificing BTE is possible by SA retardation from BESA.

11.6 PERFORMANCE MAP & INDICATED QUANTITIES

Observation of the data revealed that for constant engine speed (N) and ϕ , the brake mean effective pressure (bmep) varied approximately linearly with volume cylinder filling (VCF) which in turn was found to be a linear function of manifold vacuum. This observation is consistent with theory because, as was seen in Section 11.3, the best efficiency spark advance was not a strong function of VCF. In addition, when the lines of bmep versus VCF at constant N and ϕ were extended at zero VCF, they were found to converge to approximately the same point on the bmep axis. These observations, which are further described in Section 11.6.2, led to the use of the method described in Section 11.6.1 for performance mapping the hydrogen-engine data and for estimating indicated mean effective pressure (imep) and indicated thermal efficiency (ITE) from the bmep and BTE measurements.

11.6.1 Method (Performance Mapping and Indicated Quantities from Brake Measurements)

The solid line labeled bmep in Figure 18 is a representative linear approximation of the bmep versus VCF data for constant N and ϕ (See Figure 19a). The dashed extrapolation-to-zero VCF line intercepts the bmep axis at a value of -A. At this interception the imep is zero and

$$-A = (\text{bmep})_{\text{VCF}=0} = (\text{pmep})_{\text{VCF}=0} + (\text{fmep})_{\text{VCF}=0} \quad , \quad (11.5)$$

where $(\text{pmep})_{\text{VCF}=0}$ and $(\text{fmep})_{\text{VCF}=0}$ are the pumping and frictional mean effective pressures respectively at zero VCF (i.e. at zero imep).

To obtain indicated quantities the analysis described below was used. By definition (Obert, 1973)

$$\text{imep} = \text{bmep} + \text{pmep} + \text{fmep} \quad . \quad (11.6)$$

Differentiation with respect to VCF, while holding N and Φ constant yields:

$$\left. \frac{\partial(\text{imep})}{\partial(\text{VCF})} \right|_{N, \Phi} = \left. \frac{\partial(\text{bmep})}{\partial(\text{VCF})} \right|_{N, \Phi} + \frac{\partial(\text{pmep})}{\partial(\text{MV})} \left. \frac{\partial(\text{MV})}{\partial(\text{VCF})} \right|_{N, \Phi} + \left. \frac{\partial(\text{fmep})}{\partial(\text{VCF})} \right|_{N, \Phi}, \quad (11.7)$$

where MV is manifold vacuum. As illustrated in Figure 18, bmep is approximately a linear function of VCF for constant N and Φ , therefore

$$\left. \frac{\partial(\text{bmep})}{\partial(\text{VCF})} \right|_{N, \Phi} = \alpha, \quad (11.8)$$

where α is the slope of the Figure 18 bmep versus VCF line.

The rate of decrease of pmep with decreasing manifold vacuum is approximately constant:

$$\frac{\partial(\text{pmep})}{\partial(\text{MV})} = -\beta, \quad (11.9)$$

where, according to Kerely and Thurston (1961), $\beta \approx 0.45$ psi/in. Hg. Measurements made with the present-study test engine showed the manifold vacuum to be a linear function of VCF for constant Φ and N:

$$\left. \frac{\partial(\text{MV})}{\partial(\text{VCF})} \right|_{N, \Phi} = \gamma, \quad (11.10)$$

where, from present-study data, $\gamma \approx 0.0033$ in. Hg/% VCF.

To account for the increase in fmep at constant N due to increasing ring, piston and bearing friction as the imep increases, the following equation is used:

$$\text{fmep} = (\text{fmep})_{\text{imep} = 0} + \delta(\text{bmep} + A). \quad (11.11)$$

Equation (11.11) is a conservation accounting of the change in fmep with imep; Gish and McCullough (1958) found $\delta \approx 0.05$. Integration of Equation 11.11 and combination with Equation (11.8) yields:

$$\frac{\partial(\text{fmep})}{\partial(\text{VCF})} = \delta \cdot \gamma. \quad (11.12)$$

When Equations (11.7 - 10 and 12) are combined the following results:

$$\left. \frac{\partial(\text{imep})}{\partial(\text{VCF})} \right|_{N, \phi} = (1 + \delta) \alpha - \beta \cdot \gamma \quad (11.13)$$

Integration of Equation (11.13) while holding N and ϕ constant yields:

$$\text{imep} = [(1 + \delta) \alpha - \beta \cdot \gamma] \text{VCF} \quad (11.14)$$

Since

$$\alpha(\text{VCF}) = (\text{bmep} + A), \quad (11.15)$$

Equation (11.14) can be rewritten as:

$$\text{imep} = \text{bmep} + A + \alpha(\text{bmep} + A) - \beta \cdot \gamma(\text{VCF}), \quad (11.16)$$

which in turn can be combined with Equation (11.5) to yield:

$$\text{imep} = \text{bmep} + (\text{pmep})_{\text{VCF}=0} - \beta \cdot \gamma(\text{VCF}) + (\text{fmep})_{\text{VCF}=0} + \delta(\text{bmep} + A). \quad (11.17)$$

Equation (11.17) was used to calculate imep from bmep and VCF data as presented in a performance map like Figure 18 where the various terms in Equation (11.17) are illustrated. The term $\beta \cdot \gamma(\text{VCF})$ accounts for the decrease in pmep from its maximum value of $(\text{pmep})_{\text{VCF}=0}$ where no fresh charge is admitted to the cylinder to its value of $(\text{pmep})_{\text{VCF}=0} - \beta \cdot \gamma(\text{VCF})$ at an operating condition where the throttle valve is open thereby allowing fresh charge to enter the cylinder ($\text{VCF} > 0$). The term $\delta(\text{bmep} + A)$ accounts for the increase in ring, piston and bearing friction as the imep increases.

11.6.2 Results

Figure 19a illustrates the bmep versus VCF data taken at 1800 RPM and BESA. As discussed in Section 11.6.1 the bmep is seen to be approximately a linear function of VCF for constant ϕ . (Figure 19b shows the bmep as a function of ϕ for the unthrottled and $\overline{\text{VCF}}$ for the throttled engine.) In Figure 20 the portions of the bmep versus VCF lines which have been extrapolated to zero VCF are shown. These lines correspond to the dashed line in Figure 18 and are seen to converge at the point $A \approx 30$. Using this value of A and the values of $\beta = 0.45$ psi/in. Hg, $\gamma = 0.0033$ in. Hg/% VCF,

and $\phi = 0.05$, as given in Section 11.6.1, Equations (11.5) and (11.17) become

$$\text{imep} = 1.05(\text{bmep} + 30) - 0.152 \text{ VCF} \quad (11.18)$$

Equation 11.18 was used in conjunction with Figure 19 and the following equation to generate the indicated thermal efficiency values given in Table 16:

$$\text{ITE} = \text{BTE} \frac{\text{imep}}{\text{bmep}} \quad (11.19)$$

The Figure 19 performance map was also used to generate Figure 21 where BTE is plotted as a function of bmep for various values of ϕ (and throttle setting). It is observed that for bmep greater than about 6 psi, the unthrottled mode of operation yields the higher BTE. For lower bmep, partial throttling yields a higher BTE. This change in the effect of throttling on BTE is attributed to the very large burning-time and heat losses associated with the large spark advance required during operation at the extremely lean mixtures needed to obtain bmep less than around 6 psi in the unthrottled mode. By closing the throttle at such light loads, the increased pumping losses are more than compensated for by the decrease in the aforementioned losses. For constant loads above 6 psi bmep the BTE is approximately a linear function of ϕ : See for example Figure 22 where the load is 51.4 psi, the engine speed is 1800 RPM and BESA is used.

11.6.3 Discussion

The peak values of ITE given in Table 16 are in good agreement with those of King and Rand (1955). This agreement supports the validity of the Section 11.6.1 method of calculating indicated quantities from brake quantities. The advantage of this method is that it avoids the problems associated with the motoring method of determining indicated quantities (Lancaster, 1975). It is less complex than the indicator-diagram measurement method which also has errors in it (Evers, 1977). Since the bmep is very close to being a linear function of VCF, relatively few data points are required to generate a performance map of the type presented in Figure 19 from which can readily be calculated indicated quantities. The method, however, is not exact. For example, the bmep versus VCF lines probably do not remain straight all the way to the bmep axis (point A, Figure 18). Instead, as a result of increasing

burning-time and heat losses at low values of VCF due to large spark advance requirements, the bmep probably decreases rapidly near $VCF = 0$ resulting in an interception at a point lower than A. While not an exact method, it must be pointed out that neither are the motoring and indicator diagram measuring methods. Tests should be carried out to ascertain the relative merit of the Section 11.6.1 method.

11.7 PEAK POWER OUTPUT

During the first-year work the power output for the Pre IVC engine was limited by the occurrence of flashback emanating from preignition at the exhaust valve. The peak power levels attained have been illustrated in Figure 13 for both the throttled and unthrottled mode of operation without EGR or water injection. The effects of EGR and water injection on peak power output are shown in the data considered in Section 11.13 and 11.14 respectively.

11.8 HETEROGENEOUS MIXTURES - COMBUSTION CHAMBER

The effects of heterogeneous mixtures in the combustion chamber have been considered in Section 11.1.4 where the effect of charge stratification was discussed and shown to increase the H_2 Utilization Coefficient, HU. Whether charge stratification increases HU depends on the characteristics of the stratification. The higher HU observed by the authors during their testing of a 1972 Toyota engine (Section 11.1.4; Swain and Adt, 1972; Swain, 1973) is attributed to the induction of hydrogen near the center of a swirling fresh-air charge. The swirl model given in Appendix 12 shows that if hydrogen is introduced at the center of a swirling-air charge it will remain concentrated there. The diffusion rate of hydrogen, while large, is not sufficient to cause destratification of the charge once it is formed. Only turbulence will destroy the stratification. The spark plug of the 1972 Toyota engine, being at the center of the combustion chamber, was able to ignite the stratified charge with its hydrogen-rich core even though the overall ϕ was very low. Watkins and Milkins(1976) data indicate that they may also have had a stratified charge and that their stratification resulted in a leaner mixture near the plug. This suspected stratification resulted in their spark advance being much greater than that found by other researchers using a similar type engine

The cause of the stratification in their engine is attributed to their introducing the hydrogen by way of a valve located close to the intake valve and by retarding the introduction of hydrogen relative to the induction of air.

While there is only a small amount of data available, it is felt that nonuniformities in the combustion-chamber gases can be used to advantage in Pre IVC engines.

11.9 HETEROGENEOUS MIXTURES - INTAKE MANIFOLD

The results of the present study (Section 7.4) and others (Billings, 1977) show that heterogeneous mixtures, as reflected by cylinder-to-cylinder maldistribution, can exist in carburetted, hydrogen-fueled engines and that the extent of the maldistribution is affected by carburetor design (Section 7.4). Consideration of the diffusion rate of hydrogen shows that it is not sufficient to make uniform mixtures out of nonuniform mixtures in intake systems. Thus, it is important to use a carburetor or gas-mixer that yields a homogeneous hydrogen-air mixture. Since maldistribution prevents all of the cylinders from operating at best efficiency, it results in lower BTE. The effects of maldistribution on BTE are similar to those found with gasoline-fueled engines (See for example Yu, 1963).

Maldistribution can also increase NO_x emissions. Its effects on the emissions from a hydrogen-fueled engine are quite different from those found with gasoline-fueled engines because the former engines typically operate on different portions of the NO_x versus ϕ curve than the latter.

11.10 FLAME INITIATION FAILURE

Flame initiation failure, induced by the Section 8.4 SSLII, did not result in any unexpected behavior. It was of interest to find that the hydrogen fuel survived the misfired cycles and did not burn in the exhaust system as discussed in Section 11.2.3. No safety-related problems, however, were found.

11.11 HYDROGEN CONCENTRATION IN CRANKCASE (BLOWBY)

Blowby rates and the resulting concentration of hydrogen in the crankcase of the various potential hydrogen engine configurations is a function of hours of engine use, ring design, etc. and its detailed assessment is considered beyond the scope of the present study. Tests were performed, however, to measure the blowby rate for the present-study test engine (Section 9.3) and to determine the

flammability of the crankcase gases. These tests were performed mainly as a safety precaution to determine whether the rate at which shop air flowed into the crankcase during testing for ventilation purposes was sufficient to prevent the build-up of a combustible mixture therein. With a shop-air flow rate of 8.0 cu ft/min, crankcase gases were collected in balloons into which a spark plug was placed. Under all operating conditions, ignition of the plug did not result in combustion of the sampled gases.

11.12 MISCELLANEOUS EMISSIONS

11.12.1 CO, CO₂, HC

Table 17 shows the CO, CO₂ and flame ionization detector (FID) HC emission data obtained at 1800 RPM and 0 psi bmep for various values of ϕ and VCF. For stoichiometric and leaner mixtures ($\phi < 1$) the CO emission level was below the 0.002 volume % sensitivity level of the analyzer. At a ϕ of 1.1 the emission level was about 100 ppm. Strelau (1976) found similar trends in CO emission level with his CFR engine, reporting a CO concentration of 700 ppm at $\phi \approx 1.3$ (excluding an unusual data point at $\phi \approx 0.7$ with CO $\approx$ 1000 ppm).

The CO₂ emission level was below the approximately 0.05 volume % sensitivity level of the CO₂ analyzer for all ϕ .

FID HC emissions were found to be a function of manifold vacuum as is illustrated in Table 17 where the HC emission level is seen to increase with decreasing VCF (increasing manifold vacuum). The HC levels presented are somewhat lower than the 650 ppm level reported by Strelau (1976). This difference is probably due to engine-to-engine differences in the condition of the rings, cylinder walls and valve guides.

11.12.2 H₂O₂

An analysis using the phenol-ferrous sulfate solution (PFS) analysis method (Section 8.5) under moderately-heavy loads gave no indication of the presence of H₂O₂ in the hydrogen-fueled engine exhaust. Subsequent PFS analysis of the exhaust-gas samples taken during light loads, however, showed H₂O₂ to be present and that the level was a function of ϕ . To maximize the H₂O₂ emission level, tests were conducted at zero bmep. The results of these tests are shown in Figure 23. There it is seen that low H₂O₂ emission levels are maintained except for lean mixtures having ϕ less than about 0.5.

It is not known for certain if the influence of ϕ on H_2O_2 emissions is a result of changing exhaust gas temperatures and corresponding H_2O_2 decomposition reactions in the exhaust system, or a result of the excess oxygen present during combustion, or the result of poor HU with quenching and nonuniform burning at lean mixtures. It is of interest to observe, however, that the increase in H_2O_2 emission correlates with the deterioration in HU as seen in Figure 23. This observation and the results of tests with ethane (Trumpy et al, 1969), which show H_2O_2 to be formed during preflame reactions, suggests that perhaps the high H_2O_2 levels at lean mixtures in the hydrogen-fueled engine results from the quenching of preflame reactions. This quenching is reflected by the poor HU and the excess oxygen in the exhaust during operation when H_2O_2 emissions are high.

11.13 EFFECTS OF EGR ON PERFORMANCE AND EMISSIONS

In the following sections the effect of EGR on performance and emissions characteristics are discussed. In Section 11.13.1 the unthrottled mode of operation with EGR (Table 1, Engine Configuration No. 6) is considered; in Section 11.13.2 the throttled mode (Configuration No. 3) is considered. In Section 11.13.3 the effects of EGR on both engines are compared.

11.13.1 Unthrottled Engine

The use of EGR while maintaining constant load, constant speed and BESA did not affect the NO_x emission level of the unthrottled engine (Appendix 7, TEST NO. 74-80). This result is suspected of being caused by the necessity of increasing ϕ while increasing the EGR rate at constant bmep: the increase in NO_x emissions resulting from the needed ϕ increase negates the tendency to have reduced NO_x emissions with increasing EGR rates.

It is of interest to compare the effect of increasing EGR rate at constant ϕ and VCF (thus decreasing bmep) on the BESA with the effect at constant VCF and bmep. TEST NO. 36 and 78 (See Appendix 7) show that, as the EGR rate was increased for constant ϕ and VCF, it was necessary to advance the spark to maintain BESA conditions; TEST NO. 74-80 show that it was necessary to retard the spark to maintain BESA conditions. The latter effect is unexpected until it is realized that in order to maintain constant bmep at

constant VCF with increasing EGR rates it was necessary to enrich the mixture; the BESA-advancing effects of increasing EGR rates are more than compensated for by the BESA-retarding effects of increasing ϕ .

The Appendix 7 TEST NO. 74-80 show that at BESA there is a small, but approximately linear, decrease in BTE from 28.4% to 27.6% as the EGR rate is increased from 0 to 29.2%.

11.13.2 Throttled Engine

With the throttled mode-of-operation EGR tests, ϕ was kept constant ($\approx 1.03 - 1.04$) and the VCF was increased (throttle was opened) to maintain constant bmep (≈ 51 to 52 psi). As shown in Figure 24 there was a continual decrease in the NO_x emission level as the EGR rate was increased while maintaining constant load at BESA and 1800 RPM. With zero EGR, the BESA value was 0 deg. Data taken with the SA kept constant at this value are also shown. Since this 0 deg SA condition is retarded from BESA (Figure 24c), NO_x emissions are lower.

Figure 24b shows the BTE as a function of % EGR for BESA and constant SA. The BTE is seen to increase for operation at both BESA and constant 0%-EGR BESA up to an EGR rate of about 19%. This increase is suspected of being a result of a reduction in pumping mean effective pressure (pmep) resulting from the increased VCF (more open throttle) needed to maintain constant load with increasing EGR. There are other factors that can affect BTE as EGR is changed such as: (1) Change in the pressure-time diagram as reflected by the change in BESA with increasing EGR; (2) Change in the heat transfer from the working fluid; (3) Change in the specific heat of the working fluid. The influence of these and other factors on BTE apparently cancel each other. Consider, for example, Appendix 7 TEST NO. 27 and 91. These points have the same BTE, yet the EGR rate is different. They do, however, have the same pmep (as reflected by equal values of VCF). Thus, the effects of EGR gas on BTE are apparently negligible when constant pmep (VCF) is maintained.

For EGR rates greater than around 19%, Figure 24b shows the BTE to decrease for the constant SA case. This decrease is suspected of being caused by excessive losses, accruing due to operation at progressively more retarded SA (as the EGR rate is

increased; Figure 24c). These losses at 19% EGR become greater than the gains accruing due to operation with progressively lower pmep. For the BESA-operation case, the lowering of pmep as the EGR rate increases is suspected of causing the continual increase in BTE with increasing EGR rate.

11.13.3 Unthrottled vs. Throttled Engines

In this section throttled engine characteristics are compared with those of the unthrottled engine with a constant EGR rate of 14%, $N = 1800$ RPM, and BESA.

Figure 25a shows the BTE as a function of $\overline{VCF}$ for the throttled engine and as a function of ϕ for the unthrottled engine. The loads are relatively high, commencing at near the flashback-limited ϕ or $\overline{VCF}$ for the non-EGR mode of operation (Section 11.7). There is seen to be a very slight decrease in BTE as ϕ is increased for the unthrottled engine and a slight increase in BTE as $\overline{VCF}$ is increased for the throttled engine. Figure 25b shows the corresponding BESA and Figure 26 the bmep, while Figure 27 shows the NO_x emissions. In Figures 28 and 29 the BTE, BESA and NO_x emissions are shown as a function of bmep.

It is observed that the unthrottled engine exhibits higher efficiency and lower emissions than the throttled engine for all conditions tested.

11.13.4 EGR and Flashback

At 1800 RPM the engine would not operate without flashback at wide open throttle, $\phi \approx 1.05$ and 13.9% EGR. Increasing the EGR rate to 18.7% (for the same other conditions) allowed for operation without severe flashback but not with 100% dependability; occasional flashback occurred. Increasing the EGR rate to 23.6% (for the same other conditions) allowed for continuous flashback-free operation.

11.14 EFFECTS OF WATER INJECTION

The effects of water injection on performance and emissions characteristics for the unthrottled (Table 1 Configuration No. 6) and the throttled (Table 1 Configuration No. 3) mode of operation are discussed in Sections 11.14.1 and 11.14.2 respectively. In Section 11.14.3 these characteristics are compared for the two modes of operation at variable load (ϕ or $\overline{VCF}$) with a constant water injection rate of 5 lbm H_2O /lbm H_2 .

11.14.1 Unthrottled Engine

In Figure 30a the NO_x emission level and BTE are shown as a function of water injection rate at 1800 RPM for the unthrottled mode of operation at BESA and 30 deg SA (the BESA at zero-water injection, Figure 30c). The NO_x emissions are seen to be low with zero water injection as a result of unthrottled operation, and become even lower as the water injection rate is increased. After an injection rate of about 7 - 8 $\text{lbm H}_2\text{O}/\text{lbm H}_2$ is reached, increasing the injection rate has a small effect on further reduction in NO_x emissions. The BTE is also seen (Figure 30b) to decrease with increasing injection rates up to 7 - 8 $\text{lbm H}_2\text{O}/\text{lbm H}_2$. Similarly, the BESA value increases and then remains constant with increasing injection rates when 7 - 8 $\text{lbm H}_2\text{O}/\text{lbm H}_2$ is reached. This diminishing effect of water injection on the engine performance occurs at about the same diluent heat-capacity injection rate where the effects of EGR were found to diminish (to be discussed in Section 11.15).

In Section 11.13.1 increasing EGR was found to have no measurable effect on the NO_x emission level for the unthrottled engine because with increasing EGR it was necessary to increase ϕ to maintain constant bmep due to the effective reduction in VCF caused by the presence of EGR gas in the fresh charge. This effective reduction in VCF does not occur with water injection and it was therefore not necessary to increase ϕ to maintain constant bmep. Thus, the injected water resulted in a reduction in NO_x ; there was no NO_x increase due to ϕ enrichment as found with EGR to counteract the NO_x -reducing effects due to water injection.

11.14.2 Throttled Engine

With the throttled engine the water injection also decreased NO_x emissions (Figure 31a). The reduction, however, is more significant since the zero-water-injection NO_x level is much higher than that for the unthrottled engine (3600 ppm vs. 5 ppm). The effectiveness of increasing injection rates tends to diminish at about 7 $\text{lbm H}_2\text{O}/\text{lbm H}_2$, similar to the characteristics observed for the unthrottled engine. The BTE at BESA (Figure 31b) at first increases with increasing water injection rate and then reduces back to the BTE at zero water injection. It is of interest to observe that the BESA increases up to an injection

rate of about 6 - 7 lbm H₂O/lbm H₂; further increase in injection rate results in no change in BESA. This observation suggests that, for greater than 6 - 7 lbm H₂O/lbm H₂, an increase in water injection rate does not affect flame initiation or flame speed. This result was not expected and is an area which will be addressed in the future.

11.14.3 Unthrottled vs. Throttled Mode Of Operation

Similar to the information given for EGR in Section 11.13.3, Figures 32 - 34 show the effect of ϕ or VCF on the BTE, BESA, bemp, and NO_x emissions for the unthrottled and throttled mode of operation at a constant water injection rate of 5 lbm H₂O/lbm H₂. The unthrottled mode of operation yields superior performance and emissions characteristics. Similar characteristics are shown in Figures 35 and 36 at the same water injection rate but this time as a function of bmep.

11.14.4 Water Injection and Flashback

In order to eliminate flashback at 1800 RPM, $\phi \approx 1.05$ and wide open throttle, it was necessary to use water injection at a rate of approximately 6 lbm H₂O/lbm H₂.

11.15 EGR, WATER INJECTION AND NO_x EMISSIONS

When a diluent is used to reduce NO_x emissions it is found that there is a good correlation between the reduction in NO_x emissions and the diluent heat capacity flowrate; the diluent heat capacity being defined as the mass flowrate of the diluent times the specific heat of the diluent (Quader, 1971). In Figure 37 the percentage reduction in the NO_x mass emission level is shown as a function of the diluent heat capacity flowrate as found during the EGR and water-injection tests. It is seen that there is a very good correlation between NO_x reduction and diluent heat capacity flowrate. The EGR data exhibits a much smoother correlation which is attributed to the good performance of the EGR separator (Section 7.7) over the entire range of EGR testing. The poorer correlation of the water injection data is attributed to the imprecision of the rotameter used to measure injected water flowrate at low diluent heat capacity flowrate (less than

around 0.1 BTU/°R min). The rotameter used was one that gave high precision at the higher values of heat capacity flowrate. This meter was chosen because it was felt more important to gather precise data at the high water injection rates where it is needed for flashback and NO_x control.

It is observed from Figure 37 that: (1) Water injection and EGR are equally effective in reducing NO_x emissions when equal diluent heat capacity flowrates are used and that; (2) There is a diminishing effect of diluent heat capacity increase on NO_x reduction at a diluent heat capacity of about 0.2 BTU/°R min for both EGR and H₂O injection.

11.16 HUMIDITY

Robinson (1970) found an increase in humidity to result in a decrease in NO_x emissions. Similar results were anticipated for the hydrogen-fueled engine. Test results, however, did not show the anticipated results. In the following table the hydrogen engine data is compared with those of Robinson's (1970) gasoline-fueled engine:

Appendix 7 TEST NO.	$\left(\frac{\text{H}_2\text{O Partial Pressure}}{\text{Atmospheric Pressure}} \right)^*$	NO _x , ppm	
		Present Study	Robinson Data (1970) **
26 (95.8) ⁺	0.398/14.74	4380	2600
81 (59.0) ⁺	0.262/14.25	4670	
88 (47.9) ⁺	0.157/14.82	4500	3100

* Present Study

** At similar (H₂O Partial Pressure/Atmospheric Pressure) and ϕ as used in the present study.

+ Relative Humidity, %, present study.

The Robinson data show a 500 ppm decrease in NO_x emissions as the partial pressure of water (humidity) increases. The hydrogen-engine data, however, does not exhibit the same behavior. The test-to-test (26-81-88) variation in NO_x emissions with changing humidity is due to experimental uncertainty.

While NO_x variation with humidity was anticipated, the observed lack of variation is not unexpected. The equilibrium exhaust-gas composition of a gasoline-fueled engine operated at $\phi = 1$ is 14% by volume water; for a hydrogen-fueled engine the corresponding concentration is 35%. For an engine with a compression ratio of 9 to 1, approximately one-ninth of the charge at the end of the intake stroke is residual gas. Thus, for 50% relative humidity, approximately one-half of the water in the combustion chamber at the end of the intake stroke of a gasoline-fueled engine is from the residual gas, while five-sevenths of the water in a hydrogen-fueled engine is from the residuals. Therefore, for a hydrogen-fueled engine the influence of water in the atmosphere on the water concentration at the end of the intake stroke is small compared to the gasoline case. Consequently, humidity affects on NO_x will be smaller for the hydrogen-fueled engine.

12. COMPARISON WITH RESULTS OF OTHERS

In previous sections the results of others were compared with those of the present study when pertinent. Other comparisons have also been made and they are presented in the following.

12.1 THERMAL EFFICIENCY

12.1.1 Brake vs. Indicated Thermal Efficiency

BTE is not a good parameter for comparing the energy conversion efficiency of hydrogen as a fuel when it is used in different-type engines because: (1) compression ratio differences are not readily accounted for, and (2) there can be marked differences in engine-to-engine frictional losses. See, for example, Figure 38 in which the BTE is shown as a function of ϕ for the unthrottled mode of operation as found by various researchers. The single cylinder CFR-engine BTE data of King (1955), and Watson and Milkins (1975), and the single cylinder (compression-ignition type) diesel engine of Lenertz (1974) are seen to be lower than the BTE data of the multicylinder engine BTE data of Finegold (1974), Thomas and Sorenson (1976), Billings Energy Corporation (1977) and the present study. This difference in BTE is due to the high frictional losses of the single cylinder engines.

Because of engine-to-engine variation in friction and compression ratio, ITE is preferred over BTE as a parameter when results of various investigators are compared.

12.1.2 Method of Determination of ITE From BTE Data

While ITE is a more appropriate parameter to use when comparing the energy conversion efficiency of different engines, the ITE, unfortunately, is not always measured. It was thus decided to use the method presented by Bishop (1964) to estimate ITE from hydrogen-fueled engine BTE data. Bishop's method accounts for engine-to-engine differences in friction. To account for the effect of oil viscosity, the method of Taylor (1976) was employed. Resulting ITE values were corrected to a compression ratio, r , of 9 to 1 using the air-cycle equation (Taylor, 1976):

$$ITE = 1 - \left(\frac{1}{r} \right)^{k-1} , \quad (12.1)$$

where k is the ratio of specific heats. As will be seen, this method of estimating ITE from BTE data appears to yield good results

which are in close agreement with the ITE estimated by the method given in Section 11.6 for the present-study test engine.

Finegold (1974) did not publish BTE vs. ϕ data for his engine. Instead he presented BTE vs. bmep. In order to calculate ITE from his data, it was assumed that his engine was operated with constant VCF, and using his published value of $\phi = 0.6$ at maximum bmep, the bmep at $\phi = 0.3$ was calculated. Using this calculated value, the BTE was then found from his published BTE vs. bmep curve.

12.1.3 Comparison of ITE With Others

In Figure 39 the calculated values of ITE for engine speeds ranging between 1500 and 2040 RPM are shown. The Table in Appendix 12 contains the detailed information used to generate the points plotted in Figure 39 as well as in Figure 40 where higher engine-speed data are presented.

With a few exceptions, there appears to be good agreement between the ITE values for the various investigators in Figure 39. One of the exceptions is the data of Billings Energy Corporation (BEC, 1977). It is suspected that the reason for the apparent discrepancy of their Appendix 12.3.3 DATA NO. 7 and 8 data is due to problems experienced by BEC in measuring hydrogen fuel mass flowrates. Consider, for example, Appendix 12.3.3 DATA NO. 7 and 11. For these two points, the bmep differs by 2.4% reflecting good reproducibility of bmep (torque) measurement. The efficiencies (BTE), however, differ by 8% reflecting the suspected problems of measuring fuel mass flowrates.

A study of the BEC method of measuring hydrogen mass flowrates was made (BEC, 1977-78). Their fuel-flow delivery system was as follows: The hydrogen was stored outside the building in 96 high pressure tanks at a temperature of between 20°F and 45°F. A regulator reduced the pressure to 500 psi and the hydrogen traveled through 3/8 in. OD copper tubing through the building, which was kept at 68°F, until it reached the test cell which was maintained at 85°F. In the test cell a regulator dropped the pressure to 100 psi after which it traveled through the flowmeter and on to the engine. Since the hydrogen flow was measured as a volume flowrate in the fuel flowmeter, the density of the hydrogen was needed to calculate mass flowrate. A pressure of 100 psi and a temperature of 85°F was assumed by BEC in calcu-

lating density.

From the present-study authors' experience, the natural convection heat transfer rate from environmental air to hydrogen flowing in a 3/8 in. OD copper tube is not always adequate to raise the temperature of the hydrogen to that of the surrounding air. If the heat transfer rate is not adequate then, in effect, the hydrogen will be at a lower temperature and therefore denser than assumed. In the limiting case, this increased density effect could cause an artificial increase in BTE of 13%. This effect could explain the 8% difference in BTE between Appendix 12.3.3 DATA NO. 7 and 8 and the apparent Figures 39 and 40 ITE discrepancy.

The other BEC data, which do not appear to be consistent with the results of others, are Appendix 12.3.3 DATA NO. 13 and 19. These data points were taken with a second engine of 454 CID, the same size as the engine used to generate Appendix 12.3.3 DATA NO. 7 and 8. Discussions with BEC revealed that though both engines were tested with the same rotameter, the same calibration curve was not used to reduce both sets of data. The calibration curve used for the data taken with the second 454 CID engine, reduced the volumetric flowrate of hydrogen by 7% for the same meter reading. If the second calibration curve is assumed to be the one in error, this assumption explains the low values of volumetric efficiency one calculates when using the published BEC data.

Another exception is the data of Thomas and Sorenson (1976) which shows low values of ITE. The reason for this exception is not known. Also included in Figure 39 are the ITE values with water injection. It is observed that water injection increases the ITE. This increase is suspected of being caused by a decrease in the specific heat of the working fluid with water injection.

12.1.4 Comparison of ITE Determination Methods

For the present-study test engine, the ITE was estimated by two methods: (1) The Bishop/Taylor method of Section 12.1.2; and (2) The performance map plot method of Section 11.6. There is excellent agreement between the two methods. Consider Figure 41

where the ITE for the unthrottled mode of operation as determined by both methods is presented. The difference in the ITE predicted by the methods is on the order of the present-study BTE experimental uncertainty.

13. SUMMARY AND CONCLUSIONS

Baseline data for throttled and unthrottled, carburetted, hydrogen engine configurations with and without EGR and water injection have been presented. In addition to basic data gathering concerning performance and emissions, the test program conducted was formulated to address in detail the two major problems that must be overcome if hydrogen-fueled engines are to become viable: (1) flashback and (2) comparatively high NO_x emissions at high loads. In addition, the results of other hydrogen engine investigators were adjusted, using accepted methods, in order to make comparisons with the results of the present study. The comparisons revealed no major conflicts. In fact, with a few exceptions, there was found to be very good agreement between the results of the various studies.

Some of the results found during the present study were:

13.1 PERFORMANCE AND DESIGN

(1) Unburned hydrogen was found in the exhaust during normal operation. The level of unburned hydrogen was quite large during lean-operation.

(2) Consistent with the results of others, the unthrottled engine, vis à vis the throttled engine, exhibited higher brake thermal efficiency except at very light loads where some throttling is beneficial.

(3) Peak power output calculations show that the potential, non-flashback-limited power output of Pre IVC engines is larger than usually reported if advantage is taken of the properties of hydrogen as a fuel.

(4) Power was limited by flashback for all engine configurations tested without EGR and water injection. The use of EGR and water injection, however, allowed for wide-open throttle operation at near peak-power ϕ .

(5) Comparison of the results of the present study with those performed by the authors in the past indicates that charge stratification can extend the lean limit of operation for the unthrottled engine.

(6) Proper carburetor (gas-mixer) selection can reduce cylinder-to-cylinder variations in fuel-air ratio with subsequent improvement in performance.

(7) For residence times in normal-operating engines, the diffusion rate of hydrogen is not sufficiently high to make homogeneous mixtures from initially heterogeneous mixtures.

(8) Two independent methods were used to generate indicated quantities from brake measurement: (a) Performance-Map Method; (b) Bishop/Taylor Method. The methods gave results consistent with each other when applied to the present-study data. The Bishop/Taylor method was used to make the comparison of the present-study data with those of others and good agreement was found eliminating much of the apparent conflict of results as reported by others.

(9) The lithium-filled exhaust valves did not attain their performance potential, a phenomena recently found by industry.

13.2 FLASHBACK

(1) Hot spot-induced flashback was found to be predictable and therefore subject to characterization. There is a period preceding hot spot-induced flashback (Period PHIF) during which the charge is ignited by the hot spot at progressively earlier crank angles until it occurs prior to closing of the intake valve resulting in flashback. During the Period PHIF NO_x emissions rise very rapidly and the RPM drops.

(2) Large valve overlap was not found to increase flashback tendencies, negating the hypothesis that flashback can be caused by communication of the fresh charge with still-burning exhaust gases during the valve overlap period.

(3) Lean mixtures operation can cause flashback, raising the lower engine speed limit of the carburetted (uniform charge) unthrottled engine.

(4) Evidence was found to substantiate the hypothesis that oil in the residual gas can cause flashback.

(5) Partial ignition failure (misfire, flame initiation failure) does not result in flashback.

13.3 OXIDES OF NITROGEN EMISSIONS

(1) Consistent with the results of others, NO_x emissions were found to be a strong function of ϕ .

(2) The use of EGR while maintaining constant load, constant engine speed and BESA did not affect NO_x emissions for the unthrottled engine as a result of mixture enrichment with increasing EGR rates.

(3) The use of EGR while maintaining constant load, constant engine speed and BESA resulted in a decrease in NO_x emissions for the throttled engine operated at constant $\phi \approx 1.0$.

(4) The use of water injection while maintaining constant load, constant engine speed and BESA was found to decrease the initially low NO_x emission level at low ϕ .

(5) The use of water injection while maintaining constant load, constant engine speed and BESA was found to decrease the NO_x emission level of the throttled engine operated at $\phi \approx 1.0$.

(6) For the same diluent heat capacity flowrate, EGR and water injection were found to decrease NO_x emissions by a similar amount.

(7) Compared to gasoline-fueled operation, humidity has a negligible effect on NO_x emissions.

13.4 OTHER EMISSIONS

(1) FID HC emissions were found to be a function of manifold vacuum (VCF) with increasing levels at higher vacuum.

(2) The previously-used KMnO_4 titration method of measuring H_2O_2 emissions was found to give erroneous results due to interference with NO. The phenol-ferrous sulfate solution method was subsequently employed for the present study.

(3) H_2O_2 emissions were found to increase with decreasing ϕ . There was good correlation between the drop-off in the Hydrogen Utilization Coefficient, and the rise in H_2O_2 emissions during lean-mixture operation.

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16. NOMENCLATURE

In this section symbols or terms used in the text of the report and in Tables and Figures are defined. Excluded are symbols used exclusively in certain sections or Appendices when they are therein defined.

<u>Symbol</u>	<u>Definition</u>	<u>Units</u>
A	See discussion above Equation 11.5, Section 11.6.1	psi
BESA	Best Efficiency Spark Advance (before top dead center), See Section 11.3.1	deg
bmep	Brake Mean Effective Pressure	psi
BSNO _x	Brake Specific (Mass) NO _x Emissions	g/hp hr
BTE	Brake Thermal Efficiency	%
EGR	Exhaust Gas Recirculation	---
% EGR	(mole _{EGR gas} /mole _{fresh charge}) 100	%
Flashback	Combustion of the hydrogen-air mixture in the induction system. Flashback in the induction system occurs when the hydrogen-air fresh charge ignites prematurely prior to the closing of the intake valve. The flame thus generated, propagates through the fuel-air delivery system	---
fmep	Frictional Mean Effective Pressure	psi
HU	Hydrogen Utilitization Coefficient. Defined by Equation 11.1. For $\phi \leq 1$ HU is a measure of the hydrogen fuel which ends up in the exhaust as a result of incomplete oxidation even though available oxygen is present for the oxidation. For $\phi > 1$ HU is a measure of the hydrogen fuel which ends up in the exhaust gas as a result of incomplete combination of hydrogen with available oxygen, as well as, a result of the deficiency of available oxygen resulting from the rich mixture. For $\phi \leq 1$ HU is an indicator of combustion inefficiency	---
HU MEAS	As given by Equation 11.2, $\overline{HU}$ is a normalized hydrogen utilization coefficient. It is normalized by the HU measured during operation without intentional flame-initiation failure (i.e. with zero SSLII - induced misfire)	---
$\overline{HU}$ PRED	See Equation 11.3 and the note below it	---
H ₂	Hydrogen	---

16. NOMENCLATURE (Cont.)

<u>Symbol</u>	<u>Definition</u>	<u>Units</u>
H ₂ O injection	Water Injection. Injection of water into the intake manifold. Water injection rates are expressed in terms of lbm H ₂ O per lbm H ₂	---
imep	Indicated Mean Effective Pressure	psi
ITE	Indicated Thermal Efficiency	%
MV	Manifold Vacuum	in. Hg
N	Engine Speed (RPM)	rev/min
Period PHIF	The Period Preceding Hot-Spot-Induced Flashback. See Section 11.2.1	---
Post IVC	Post Intake Valve Closing. See Section 3.1	---
Pre IVC	Pre Intake Valve Closing. See Section 3.1	---
pmep	Pumping Mean Effective Pressure	psi
SA	Spark Advance (usually bt/dc)	deg
SAE	Society of Automotive Engineers	---
SSLII	Solid State Logic Ignition Interrupter. See Section 8.4	---
Throttled Operation	Operation during which the load is changed by varying the density of the fresh charge with a conventional throttle plate	---
Unthrottled Operation	Operation during which the throttle plate is left in a wide-open position and the load is controlled by changing the flowrate of hydrogen. The total volume flowrate of fresh charge is essentially constant for a fixed RPM. Thus, reducing the hydrogen flow at constant RPM results in a leaner mixture with less BTU per ft ³ of charge and therefore less power output	---
VCF	Volume Cylinder Filling, VCF, is the actual mass of hydrogen plus air, plus EGR gas present in the cylinder when the intake valve closes, divided by the mass of the same proportions of air plus hydrogen plus EGR gas that would be ingested during the intake stroke if they were at a temperature of 70°F and a total pressure of 14.7 psi. (times 100 to yield %). The term is similar to the more familiar term volumetric efficiency, VE. Volumetric efficiency, however, is usually used to measure the breathing capacity of engines at wide-open-throttle conditions. In the present work, VCF is used as a load-control parameter for the throttled hydrogen engines. When the throttle is moved to a closed position the amount of fresh mixture (air plus hydrogen	%

16. NOMENCLATURE (Cont.)

<u>Symbol</u>	<u>Definition</u>	<u>Units</u>
	plus EGR gas) is reduced and in turn the power output is directly reduced. It was decided to use VCF rather than manifold vacuum as the throttled engine load-control parameter because manifold vacuum is not a direct measurement of the mass of fresh charge as is VCF.	
VE	Volumetric Efficiency as defined in Obert (1973) is the actual mass of air* inducted per intake stroke divided by the theoretical mass of air* to fill the piston-displacement volume under atmospheric conditions	
α	See Equation 11.8	psi/%VCF
β	See Equation 11.9	psi/in.Hg
	See Equation 11.10	
	See Equation 11.11	---
ϕ	Fuel-Air Equivalence Ratio: $\phi < 1$ is lean, $\phi > 1$ is rich	---
% H ₂	Percent by Volume of Hydrogen in the Fresh Charge	%
% Misfire	Percentage of the cycles during which there is intentional flame initiation failure accomplished by nonignition of spark plugs as controlled by the Section 8.4 SSLII	---

* Air or Mixture

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TABLE 1 POTENTIAL HYDROGEN-FUELED ENGINE CONFIGURATIONS

TOTAL	Engine Con- figuration No.	Pre IVC	Post IVC	Throttled	Un- Throttled	EGR	H ₂ O Injection	Single Chamber	Pre- Chamber	Burn During Injection	Burn After Injection
	1	X*		X							
	2	X*		X		X					
	3	X*		X			X				
	4	X*			X						
	5	X*			X	X					
	6	X*			X		X				
	7	X**			X						
	8	X**			X	X					
	9	X**			X		X				
	10	X**		X	X						
	11		X					X		X	
	12		X					X			X
	13		X						X	X	
	14		X						X		X

* Carburetted (Gas Mixer)

** Timed Injection at the Intake Valve.

TABLE 2 EVALUATION OF HYDROGEN ENGINE CONFIGURATIONS⁺

PRE INTAKE VALVE CLOSING FUEL INGESTION (PRE IVC)												POST IVC			
CARBURETTED												SINGLE CHAMBER		DUAL CHAMBER	
THROTTLED			UNTHROTTLED			TIMED INJECTION			Throt. & unthrot.			BAI++	BDI++	BAI++	BDI++
Engine Configuration No.	1	EGR	H ₂ O**	2	EGR	H ₂ O**	3	EGR	H ₂ O**	4	Throt. & unthrot.	11	12	13	14
NO _x	High Load	HIGH	LOW	LOW	HIGH	LOW	LOW	HIGH	low	low	low	high	medium	medium	medium
	Low Load	LOW	LOW	LOW	LOW	LOW	LOW	LOW	low	low	low	low	LOW	LOW	LOW
Thermal Efficiency	High Load	HIGH	HIGH	HIGH	HIGH	HIGH	HIGH	HIGH	high	high	high	high	LOW hi*	MED hi*	LOW hi*
	Low Load	LOW	LOW	LOW	HIGH	HIGH	HIGH	HIGH	high	high	high	high	LOW hi*	MED hi*	LOW hi*
Maximum Power**	@Max HP RPM	92%	64%	92%	92%	64%	92%	96%	64%	96%	96%	121%	121%	121%	121%
	@Max Torque RPM	83%	59%	83%	83%	59%	83%	85%	59%	85%	85%	139%	139%	139%	139%
Flash Back	High Load	YES	NO	NO	YES	NO	NO	YES	no	no	no	NOT POSSIBLE			
	Low Load	NO	NO	NO	NO	NO	NO	NO	no	no	no	NOT POSSIBLE			
Uncontrolled Combustion	High Load	YES	NO	NO	YES	NO	NO	YES	no	no	no	yes	YES no*	YES no*	YES no*
	Low Load	NO	NO	NO	NO	NO	NO	NO	no	no	no	no	NO	NO	NO
Inhomogeneity in Combustion Chamber		slight problem	worse problem	worse problem	slight problem	worse problem	worse problem	slight problem	worse problem	worse problem	slight problem	serious problem	SERIOUS PROBLEM	SERIOUS PROBLEM	SERIOUS PROBLEM
Inhomogeneity in Intake Manifold		small problem	small problem	small problem	small problem	small problem	SMALL PROBLEM	no problem	no problem	no problem	no problem	no problem	no problem	no problem	no problem
H ₂ in Crankcase		may occur	may occur	may occur	may occur	may occur	may occur	may occur	may occur	may occur	may occur	less likely	not likely	less likely	not likely
Failure of Charge to Ignite	This problem has not been studied but presents a potential safety problem.														
Hydrogen Peroxide Emissions	This problem has not been studied in depth but is suspected of being of minor importance.														
Initial Costs		low	medium	medium	low	medium	medium	low	medium	medium	low	high	high	high	high
Operating Costs Exclusive of Fuel		low	low	medium	low	low	medium	low	low	medium	low	high	high	high	high
Complexity of Design		good	fair	fair	good	fair	fair	good	fair	fair	fair	poor	poor	poor	poor
Safety		average	average	average	average	average	average	average	average	average	average	below average	below average	below average	below average
Fuel System/Engine Compatibility		fair	fair	fair	fair	fair	fair	fair	fair	fair	fair	poor	poor	poor	poor

⁺ CAPITALIZED entries are determined from experimental results, lower case entries are estimates.

⁺⁺ BAI = Burn After Injection; BDI = Burn During Injection.

^{*} The CAPITALIZED entry has been determined experimentally, the lower case entry is predicted assuming proper engine design.

^{**} Values given are estimates with 100% being the power produced during gasoline-fueled operation at maximum horsepower RPM. Refer to Appendix 1 for the underlying assumptions made in generating these values.

^{**} H₂O = water injection.

TABLE 3 - TOYOTA 2T-C FOUR CYLINDER HEMISPHERICAL
COMBUSTION CHAMBER ENGINE

Displacement:	1588 cc or 96.9 cubic inches
Bore Diameter:	80 mm or 3.346 inches
Stroke Length:	70 mm or 2.756 inches
Compression Ratio:	9.0 to 1
Compression Pressure at 250 RPM:	12.0 Kg/cm ² or 170.6 psi
Maximum Horsepower (SAE Net):	88 horsepower at 6000 RPM
Maximum Torque (SAE Net):	91.3 ft lb _f (12.6 kg·m) at 3800 RPM
Camshafts:	(See Table 4)
Valve Clearance, Hot Intake:	0.20 mm or 0.007 inch
Valve Clearance, Hot Exhaust:	0.33 mm or 0.013 inch
Firing Order:	1342
Engine Oil Capacity:	3.7 liters or 3.9 U.S. quarts

TABLE 4 - CAMSHAFT SPECIFICATIONS
(SPECS GIVEN AT 0.050 IN. LOBE LIFT)

CAMSHAFT TYPE *	INTAKE VALVE OPENS DEG.	INTAKE VALVE CLOSES DEG.	EXHAUST VALVE OPENS DEG.	EXHAUST VALVE CLOSES DEG.	INTAKE LOBE LIFT IN.	EXHAUST LOBE LIFT IN.	INTAKE LOBE DURATION DEG.	EXHAUST LOBE DURATION DEG.	OVERLAP DEG.
C ₁	12 atdc**	21 abdc+	29 bbdc++	11 btdc ^o	0.252	0.253	189	198	-23
C ₂	12 atdc	21 abdc	31 bbdc	19 btdc	0.252	0.257	189	192	-31
C ₃	8 atdc	18 abdc	28 bbdc	18 btdc	0.238	0.238	190	190	-26
C ₄	24 atdc	17 abdc	18 bbdc	25 btdc	0.220	0.220	173	173	-49

C₁: Stock, 1976 2TC Toyota camshaft
C₂ & C₃: Low-residual camshaft
C₄: Low Duration, zero-overlap camshaft

**atdc = after top dead center
+abdc = after bottom dead center
++bbdc = before bottom dead center
^obtdc = before top dead center

TABLE 5
FUEL FLOWMETER CHARACTERISTICS

<u>MEASURING METHOD</u>	<u>ACCURACY*</u>	<u>CALIBRATION GAS</u>	<u>PRECISION**</u>	<u>RANGE</u>	<u>REMARKS</u>
Anemometer Mass Flow Transducer (Temp. Compensated)	$\pm 2\%R^+$ plus $\pm 0.4\%R$ per $10^\circ F$ $\pm 0.5\% FS^{++}$ plus $\pm 0.4\% FS$ per $10^\circ F$	Air	$\pm 1\%R$	100 to 1	$\pm 0.5\% FS$ accuracy yields greater than $\pm 2\%$ accuracy at lower end of meter range. Consecutive sized meters do not overlap in the $\pm 2\%$ accuracy range.
High Accuracy Rotameters	$\pm 1\%R$	Air	$\pm 0.2\%R$	13 to 1	
Square Edge Orifice	$\pm 1\%R$	Air	$\pm 0.2\%R$	3 to 1	ΔP measurement with precision inclined monometer accuracy ± 0.25 to $\pm 0.5\%R$.
Critical Flow Nozzle	$\pm 1\%R$	Air	$\pm 0.1\%R$	10 to 1	Requires relatively high upstream pressure which is undesirable. Expensive
Turbine Meter	$\pm 1\%R$	Air	$\pm 0.25\%R$	10 to 1	Additional electronics accuracy of up to $\pm 0.1\%R$. Expensive, bearing maintenance a consideration.

- * Accuracy is defined as the closeness to the true flow rate.
- ** Precision is defined as the repeatability of the measurement for a fixed flow rate.
- + %R = percentage of reading.
- ++ %FS = percentage of full scale.

TABLE 6 - INSTRUMENTATION (cont.)

<u>VARIABLE</u>	<u>METHOD</u>	<u>MANUFACTURER, MODEL</u>	<u>CALIBRATION</u>	<u>MANUFACTURER PRECISION CLAIM</u>	<u>ACCURACY</u>	<u>READABILITY</u>	<u>ESTIMATE OF ERROR</u>
CO	Nondispersive Infrared	Beckman, 315B	Calibrated Against Certified Span Gases (CO + CO ₂ in N ₂) and Nitrogen	±1% FS	±1.5%FS	±1%FS	±2.5%FS
O ₂	Polarographic	Beckman, OM-11	Calibrated Against Certified Nitrogen Span Gas and Room Air		±1% FS		±1% O ₂ ***
Total HC	Flame Ionization Detector	Beckman, 400	Calibrated Against Certified Span Gas (CH ₄ in N ₂ and Nitrogen) ²	1% FS	±1.5%FS	±1%FS	±2.5%FS
Blowby Rate	Displacement of H ₂ O in a Graduated Beaker	N/A	N/A				±10%R
Geometric Distribution	Individual-Cylinder O ₂ Emissions	Beckman, OM-11	Calibrated Against Certified Nitrogen Span Gas and Room Air		±1%FS		±1% O ₂ ***
Hydrogen Delivery Pressure @ Flow- meter	Precision Mercury Manometer	Wallace & Tierwan FA 187	Calibrated Against Certified Calibration Standard	0.05%FS		±0.005psi	±0.015psi

*R = Reading

+SCFM = Standard (147 psi, 70°F) cubic feet per minute

#FS = Full Scale

**N/A = Not Applicable

++% of Absolute Temperature

*** ±1% O₂ Concentration

TABLE 6 - INSTRUMENTATION (See Section 8.7)

<u>VARIABLE</u>	<u>METHOD</u>	<u>MANUFACTURER, MODEL</u>	<u>CALIBRATION</u>	<u>MANUFACTURER PRECISION CLAIM</u>	<u>ACCURACY</u>	<u>READABILITY</u>	<u>ESTIMATE OF ERROR</u>
Fuel Flowrate	Rotameters	Cox, 129-287	Certified by Colo. Exp. Engr. Station (See Section 8.1.1)	$\pm 0.2\%R^*$	$\pm 0.5\%R$	$\pm 0.5\%R$	$\pm 1\%R$
Air Flowrate	Laminar Flow Element Inclined Manometer Readout	Meriam 50 MC2-2F Meriam 40 HE35 WM	Certified by Mgf. Certified by Mgf.		$\pm 0.65\%R$ $\pm 0.25\%R$	$\pm 0.005 \text{ in. H}_2\text{O}$	$\pm 0.9\%R$ $\pm 0.06 \text{ SCFM}^+$
H ₂ O Injection Flowrate	Rotameters	Model 120-204	Certified by Mgf.	$\pm 0.2\%R$	$\pm 1\%R$	$\pm 0.5\%R$	$\pm 1.5\%R$
NO (and NO _x)	Chemiluminescent	Thermo Electron 10A with Heated Capil- laries and Airbleed (See Section 8.3)	Calibrated against Certified Span Gases NC in N ₂ and Nitrogen		$\pm 2\%FS\#$	$\pm 0.5\%FS$	$\pm 2.5\%FS$
Hydrogen Peroxide	Phenol-Ferrous Sulfate Solution	N/A **	N/A				$\pm 30\%R$
Spark Advance	Timing Light	Sears #244.2115	N/A			$\pm 1 \text{ deg}$	$\pm 1 \text{ deg}$
Engine Speed	Strobotac	Type 1538-A General Radio Co.	Calibrated against 60 Hertz line frequency				$\pm 4 \text{ rpm}$
Torque	Water-Brake Dyna- mometer Load Cells	Buck Engineering D-312 Absorption Unit	System Calibrated against known static loads	$\pm 0.2\%R$	$\pm 5\%FS$		
	Pressure Gauge	Heise H9793			$\pm 0.3\%FS$	$\pm 0.1 \text{ ft lbf}$	$\pm 0.4 \text{ ft lbf}$
Wet and Dry Bulb Temperatures	Temperature Measure- ments Wet and Dry	Psychro-Dyne Model PP100	Certified Calibration Thermometer		$\pm 0.25^\circ F$	$\pm 0.5^\circ F$	$\pm 0.75^\circ F$
Thermocouple Temp- eratures	Chromel-Alumel Thermo- couples	Thermal Corporation Models 2025-04,05,06	Calibrated against Ice Point and Boiling Water	N/A	$\pm 0.5^\circ F$ @32-212 $^\circ F$	$\pm 0.5^\circ F$	
	Digital Voltmeter	Fluke Company 8600A DMM	Certified by Mgf.	N/A	90 Days - 0.04% of input, $\pm 0.01\%$ of range (@200 MV)	$\pm 0.5^\circ F$	$\pm 0.4\%^{++}$
Barometric Pressure	Mercury Barometer	Arthur Thomas Co. 724	Calibrated against National Hurricane Center Certified Barometer	N/A	$\pm 0.02\%R$	$\pm 0.09\%R$	$\pm 0.11\%R$

TABLE 7 - DESCRIBES THE EXPERIMENTS FOR CHARACTERIZATION OF THE PRE IVC ENGINES

Experiment Number	RPM	Load*	+++ EGR	+ Water Inj.	** Misfire Rate, %	++ Extra Measurements	Distribution Measured and Engine used	Table 1 Engine Config. No.
1	800	IDLE	0	0	0	None	1, 4, 7, 10	1-6
2	800	IDLE	0	0	0-10	m-q		1, 4
3	1800	0-100%	0	0	0	None	10 only	1, 4, 7, 10
4	1800	0-100%	25	0	0	None		2, 5, 8
5	1800	0-100%	0	10	0	None		3, 6, 9
6	1800	100%	0	0	0-10	None		1, 4
7	1800	cruise +15%***	5,15,25 35,45	0	0	None		2, 5
8	1800	cruise +15%***	0	6,8,10, 12,14	0	None		3, 6
9	3300	cruise	0	0	0	None	1, 4, 7, 10	1, 4, 10
10	3300	cruise	25	0	0	None		2, 5
11	3300	cruise	0	10	0	None		3, 6
12	3300	100%	0	0	0	m-q	1, 4, 7, 10	1, 4, 10
13	3300	100%	25	0	0	None		2, 5
14	3300	100%	0	10	0	None		3, 6

* 100% is defined as the maximum load obtained with a given engine configuration.

+ $\text{lb}_{\text{H}_2\text{O}} / \text{lb}_{\text{H}_2}$

** For some tests intentional misfire will be imposed.

++ For all experiments the Table 8 measurements labeled a through l will be made. For the experiments so indicated in this table the measurements m through q will also be made.

*** The load produced by a mixture whose strength is defined as the percentage of hydrogen at cruise load with Test Engine 3 plus 15% more hydrogen.

+++ $\left(\text{lb}_{\text{EGR gas}} / \text{lb}_{\text{fuel-air mixture}} \right) \times 100$

TABLE 8 - MEASUREMENTS

Identification Letter	Variable
a	NO, NO ₂
b	O ₂
c	Fuel Flow
d	Air Flow
e	Distribution
f	Coolant Temperature
g	Individual Cylinder Exhaust Gas Temp.
h	Exhaust Valve Seat Temperature
i	Mixed Exhaust Gas Temperature
j	Torque
k	PV Diagrams
l	Engine Speed
m	Blowby Rate
n	CO
o	CO ₂
p	HC
q	H ₂ O ₂

TABLE 9 - BREAK-IN SCHEDULE (GASOLINE-FUELED)

ENGINE SPEED RPM	LOAD (TORQUE) ft lb _f	DURATION OF LOAD (HOURS)	BLOWBY RATE* (LITERS/SEC)
2000 to 3500	6 to 10	0.25	0.20
2000 to 3500	9 to 15	0.50	
2000 to 4000	15 to 27	0.25	
3000	20	2.00	0.12
3000	20	1.00	
3000	32	1.00	
3400	32	1.00	0.12
2800	27	0.25	
3300	36	0.75	
3300	44	2.00	0.12
2600	31	1.00	
3600	36	2.00	0.12
3600	36	3.00	0.12

* All Blowby flow rates measured at 3000 RPM and 20.5 ft lb_f torque during break-in period at the times indicated. For example, the blowby rate was 0.20 liters/sec after 0.25 hr of break-in operation; 0.12 liters/sec after 3.00 hr; 0.12 liters/sec after 6 hr, etc. The method used to measure blowby rates was displacement of water from an inverted graduated beaker.

TABLE 10 - H₂ UTILIZATION COEFFICIENT (HU) LESS THAN 100% INDICATES THE PRESENCE OF UNBURNED HYDROGEN IN THE EXHAUST

Throttled					
RPM	820	815	810	800	760
ϕ	1.05	1.05	1.06	1.08	1.13
% Misfire	0	1	2	5	10
HU, %	83.2	83.2	81.5	78.9	72.6

Throttled					
RPM	1800	1800	1780	1750	1660
ϕ	1.04	1.04	1.05	1.06	1.09
% Misfire	0	1	2	5	10
HU, %	92.4	92.6	91.7	89.0	84.1

Unthrottled					
RPM	820	820	820	820	810
ϕ	0.18	0.18	0.18	0.18	0.19
% Misfire	0	1	2	5	10
HU, %	56.9	50.9	45.5	45.8	43.9

Unthrottled					
RPM	1800	1780	1765	1740	1710
ϕ	0.41	0.41	0.42	0.43	0.45
% Misfire	0	1	2	5	10
HU, %	93.4	91.1	90.0	86.5	81.1

TABLE 11 - SHOWS THE GOOD AGREEMENT BETWEEN PREDICTED AND MEASURED NONDIMENSIONAL H_2 UTILIZATION COEFFICIENTS

Throttled: RPM=820 $\phi=1.05$					
% Misfire*	0	1	2	5	10
$\overline{HU}$ PRED**	100	99	98	95	90
$\overline{HU}$ MEAS**	100	100	98	95	87

Throttled: RPM=1800 $\phi=1.04$					
% Misfire*	0	1	2	5	10
$\overline{HU}$ PRED**	100	99	98	95	90
$\overline{HU}$ MEAS**	100	100	99	96	91

Unthrottled: RPM=820 $\phi=0.18$					
% Misfire*	0	1	2	5	10
$\overline{HU}$ PRED**	100	99	98	95	90
$\overline{HU}$ MEAS**	100	89	80	80	77

Unthrottled: RPM=1800 $\phi=0.41$					
% Misfire*	0	1	2	5	10
$\overline{HU}$ PRED**	100	99	98	95	90
$\overline{HU}$ MEAS**	100	98	96	93	87

* SSLII-Induced Misfire (Nonignition; flame initiation failure).
 ** See Section 11.1.2.

TABLE 12 - SHOWS THAT ϕ AND NOT ENGINE SPEED IS THE DOMINANT FACTOR INFLUENCING H_2 UTILIZATION COEFFICIENT

Unthrottled			
ϕ	RPM	GAS VELOCITY (M/SEC) AT IGNITION*	HU, %
0.410	1800	25	93.4
0.434	3300	44	94.8
0.184	820	8	56.9
0.190	1800	15	52.7

Throttled				
ϕ	VCF **	RPM	GAS VELOCITY (M/SEC) AT IGNITION*	HU, %
1.05	0.19	820	less than 5	83.2
1.09	0.19	1800	"	85.0
1.05	0.39	1800	"	90.7
1.05	0.37	3300	"	92.5

* Noguchi, et al, 1976

** See Nomenclature, Section 16.

TABLE 13 - ILLUSTRATES THE GAINS IN H_2 UTILIZATION COEFFICIENT
MADE POSSIBLE BY CHARGE STRATIFICATION

CHARGE	MINIMUM RPM	VCF	% H_2 +++	HU, %	BESA ***
HOMOGENEOUS ⁺	820	85	7.2	56.9	94
STRATIFIED ⁺⁺	400	93.6*	3.7**	100.0**	90
STRATIFIED ⁺⁺	400	85*	4.1**	96.9**	90
STRATIFIED ⁺⁺	400	75*	4.4**	91.2**	90

+ Present-Study Test Engine

++ Swain, Adt 1972; Swain 1973.

* Assumed Value (85% is probably closest to the actual VCF value)

** Calculated Based on Assumed Value of VCF

***deg BTDC

+++ Minimum % H_2 with Stable Operation.

TABLE 14 - DESCRIPTIONS OF HYDROGEN-FUELED ENGINES
TESTED BY THE AUTHORS (SPARK SET AT BESA+)

	<u>ENGINE</u>	<u>ENGINE</u>	<u>ENGINE</u>
Total duration of break in	3 hr	0 hr	15 hr **
Camshaft Type*	C ₂	C ₂ and C ₃	C ₁
Cooling System	Stock radiator no thermostat	Stock radiator no thermostat; once through city water	Stock radiator with a calibrated 195°F thermostat
Exhaust valve type	Sodium cooled 1.44 in. head diameter	Sodium cooled, 1.44 in. head diameter	Solid stem, 1.41 in. head diameter
Method of mixture preparation	Manifold induction unthrottled	Manifold induction unthrottled	Carburetor induction unthrottled
Spark plug type (larger numbers hotter)	N-19 V surface gap N-54R retracted	N-19V surface gap N-54R retracted	N-88 extended electrode
Compression ratio tested	9.0:1	9.0:1 and 7.2:1	9.0:1
Ignition type	Standard Toyota	Standard Toyota	Capacitive Discharge
Oil ring type	One piece	One piece; three piece	Three piece
Maximum BMEP at which engine would operate without flashback for a long duration (15 min)	50 psi	0 psi	60 psi

+ BESA = Best Efficiency Spark Advance

* See Table 4

** See Table 9 for break-in schedule

TABLE 15 - TYPICAL BEST EFFICIENCY SPARK ADVANCE-DETERMINATION TEST

TEST NO.*	138	139	140	141	142	143
RPM	1775	1790	1800	1800	1790	1780
SA, deg**	21	18	16	13	11	9
ϕ	0.65	0.64	0.64	0.64	0.64	0.65
Vol. % H ₂	21.4	21.2	21.2	21.2	21.3	21.4
BTE, %	25.8	26.1	26.3	26.3	26.1	25.9
NO _x , ppm	850	470	350	250	180	160
bmepp, psi	51.2	51.4	51.4	51.4	51.4	51.2

* TEST NO. corresponds to the number assigned to the data points in Appendix 7.

** deg BTDC.

TABLE 16 - ITE VALUES AS CALCULATED IN SECTION 12.1.2

<u>bmp, psi</u>	<u>VCF, %</u>	<u>ϕ</u>	<u>BTE, %</u>	<u>ITE, %</u>
12.7	78	0.24	12.5	31.3
20.5	78	0.26	18.8	36.7
31.4	78	0.32	24.0	39.3
39.3	78	0.37	26.4	40.2
44.8	78	0.41	27.6	40.4
52.0	78	0.48	28.0	39.4
68.0	78	0.62	29.7	29.3
8.5	70	0.24	9.4	31.6
15.3	70	0.26	15.7	36.6
25.0	70	0.32	21.3	39.1
32.1	70	0.37	24.0	39.9
37.2	70	0.41	25.5	40.2
43.8	70	0.48	26.3	39.4
58.1	70	0.62	28.3	39.3
69.0	70	0.79	27.8	37.1
4.4	62	0.24	5.5	31.8
10.1	62	0.26	11.7	36.4
18.7	62	0.32	18.0	38.9
25.0	62	0.37	21.1	39.7
29.4	62	0.41	22.8	40.1
35.4	62	0.48	24.0	39.3
48.2	62	0.62	26.5	39.3
57.6	62	0.79	26.2	37.0
0.3	54	0.24	0.4	29.8
4.9	54	0.26	6.5	36.0
12.4	54	0.32	13.7	38.7
17.8	54	0.37	17.3	39.6
21.7	54	0.41	19.3	39.8
26.9	54	0.48	20.9	39.0
38.3	54	0.62	24.2	39.3
46.3	54	0.79	24.2	36.9
57.3	54	1.05	24.3	34.8
6.1	46	0.32	7.9	38.5
10.7	46	0.37	12.2	39.4
14.0	46	0.41	14.6	39.6
18.5	46	0.48	16.9	39.0
28.4	46	0.62	21.0	39.3
34.9	46	0.79	21.4	36.8
44.0	46	1.05	21.7	34.6
3.5	38	0.37	4.8	38.6
6.3	38	0.41	8.0	39.5
10.1	38	0.48	11.2	38.9
18.4	38	0.62	16.5	39.3
23.5	38	0.79	17.4	36.4
30.8	38	1.05	18.6	34.3
1.6	30	0.48	2.2	37.7
8.5	30	0.62	9.7	39.5
12.1	30	0.79	11.4	36.2
17.6	30	1.05	13.5	33.9
0.8	22	0.79	1.0	34.8
4.6	22	1.05	4.8	33.2

TABLE 17 - MISCELLANEOUS EMISSIONS CO. CO₂, FID HC
(1800 RPM bmep = 0 psi)

Φ	0.19	0.36	0.54	0.67	0.75	0.85	0.82	1.02	1.1
CO ppm	0	0	0	0	0	0	0	0	100
CO ₂ , ppm	0	0	0	0	0	0	0	0	0
HC, ppm	0	1	3	5	11	15	19	33	51
VCF, %	82	54	26	22	21	20	19	18	18*

* This test point corresponds to a very high manifold vacuum, much higher than would be used in practice.

Photograph 1 - Shows the Orifice-Type Water Injectors Used

Photograph 2 - Shows the Water-Injection System
Tubes connected to Intake Manifold

Photograph 3 - Shows the Figure 5 EGR-Gas Separator

Photograph 4 - Shows the Installed Figure 5
EGR-Gas Separator

Photograph 5 - Another view of the Installed Figure 5
EGR-Gas Separator

Photograph 6 - Shows the Manometer Bank Used to Insure that
the Figure 5 Pressure Drop $P_1 - P_{2i} = P_1 - P_{2j}$
for all i and j
(See Section 7.7)

Photograph 7 - Shows the Color of the Hydrogen/Peroxide Phenol/Ferrous Sulfate Solution After Scrubbing Exhaust-Gas Samples having ϕ 's as Indicated at Zero bmep

Figure 1 - Illustrates Typical Trends in Oxides of Nitrogen Emissions as a Function of Fuel-Air Equivalence Ratio for Hydrogen-Fueled Engines (deBoer, July 1975)

Figure 2 - Approximate Local Heat Transfer Coefficients at Significant Operating Valve Interfaces and Masses in BTU/HR in.² °F (w/cm² °C), (Taken from Danis, 1973).

Figure 3 - Illustrates Locations of Temperature Probes in the Cylinder Head.

Alfa Romeo 1600 cc

Alfa Romeo 1300 cc

Figure 4 - Illustrates Improved Cavity Design of the Alfa Romeo 1300 cc Exhaust Valve.

Figure 5 - Schematic Representation of the Improved EGR-Gas Separation and Quantification System (See Section 7.7)

1. Exhaust pipe proximal to manifold; 2. Brazed fitting; 3. Stainless steel tubing; 4. Stainless steel union; 5. Teflon tubing; 6. Collection flask; 7. Ice-bath; 8. Tygon tubing; 9. Rotometer @ 100 = 1.21 Cubic feet/minute; 10. Collection pump; 11. Exhaust fan.

Figure 7 - Schematic of the Experimental Apparatus used for the Phenol-Ferrous Sulfate Solution Analysis of Hydrogen Peroxide Emissions.

Figure 8 - Illustrates the Oxygen Levels in Hydrogen-Fueled Engine Exhaust: Zero-Dissociation Theory; Present-Study Results; Watson and Milkins (1975) Results.

Shows Typical Pressure-rise Curves Moving to Earlier Portions of Cycle During the Time Period of About 5 Cycles* Before Flashback

Figure 9 - Illustrates Cylinder Pressure, NO_x Emissions and Engine Speed Characteristics at Near Flashback Operation (*Period PHIF ≈ 5 cycles at 1800 RPM, No EGR, No Water Injection; Section 11.2.1).

Figure 10 - Illustrates the Modification to the Piston Used to Increase Oil in the Residuals.

Figure 11 - Illustrates the BESA as a Function of ϕ for the Unthrottled and $\overline{VCF}$ for the Throttled Mode of Operation.

Figure 12 - Illustrates the Small Effect of Manifold Vacuum or Load (bmep) on BES and the Large Effect of ϕ .

Figure 13 - Shows the BTE as a Function of bmep for the Throttled and Unthrottled Engine Including Engine Speed Effects.

Figure 14 - Shows the BTE as a Function of ϕ for the Unthrottled and $\overline{VCF}$ for the Throttled Engine Including Engine Speed Effects.

Figure 15a- Shows Oxides of Nitrogen Concentration Emission Level as a Function of bmep for the Throttled and Unthrottled Engine Including Engine Speed Effects.

Figure 15b - Shows Oxides of Nitrogen Brake Specific Emission Level as a Function of bmeep for the Throttled and Unthrottled Engine Including Engine Speed Effects.

Figure 16a - Shows the Oxides of Nitrogen Concentration Emission Level as a Function of ϕ for the Unthrottled and $\overline{\text{VCF}}$ for the Throttled Engine Including Engine Speed Effects.

Figure 16b - Shows the Nitric Oxide Brake Specific Emission Level as a Function of ϕ for the Unthrottled and $\overline{VCF}$ for the Throttled Engine Including Engine Speed Effects.

Figure 17 - Shows the Effect of Φ on NO_x Emissions at Two Load Levels (bmep = 0 and bmep = 51.4 psi) Including the Effect of Spark Advance.

Figure 18 - Illustrates the Section 11.6 Method for Determining Indicated Quantities From Brake Measurements.

Figure 19a - Section 11.6 Performance Map of the Present-Study Data.

Figure 19b - Shows the bmep as a Function of ϕ for the Unthrottled and $\overline{VCF}$ for the Throttled Engine Including Engine Speed Effects.

Figure 20 - Extrapolation of the Present-Study Data to Zero VCF (See Section 11.6).

Figure 21 - Shows the BTE as a Function of bmep for Various Values of ϕ as determined from the Figure 20 Performance Map.

Figure 22 - Shows the BTE as a Function of ϕ at Constant Load ($b_{mep}=51.4$ psi) and Engine Speed (1800 RPM) with BESA.

Figure 23 - Shows the H_2O_2 Emission Level as a Function of ϕ at Zero bmep.

Figure 24 - Shows the Effect of EGR on NO_x Emissions, BTE, and BESA for the Throttled Engine at Constant Engine Speed (1800 RPM) and Load (≈ 52 psi bmep).

Figure 25 - Shows the Effect of ϕ for the Unthrottled and $\overline{VCF}$ for the Throttled Engine on BTE and BESA with 14% EGR.

Figure 26 - Shows the Effect of Φ for the Unthrottled and $\overline{VCF}$ for the Throttled Engine on bmep with 14% EGR.

Figure 27 - Shows the Effect of ϕ for the Throttled and $\overline{VCF}$ for the Unthrottled Engine on NO_x Emissions with 14% EGR.

Figure 28 - Shows how the DTE and BESA vary with bmep for both the Throttled and Unthrottled Engine with 14% EGR.

Figure 29 - Shows how the NO_x Emissions Vary with bmep for Both the Throttled and Unthrottled Engine with 14% EGR.

Figure 30 - Shows the Effect of Water Injection on NO_x Emissions, BTE and BESA for the Unthrottled Engine at Constant Engine Speed (1800 RPM) and Load (≈ 52 psi bmep)

Figure 31 - Shows the Effect of Water Injection on NO_x Emissions, BTE and BESA for the Throttled Engine at Constant Engine Speed (1800 RPM) and Load (≈ 52 psi bmep)

Figure 32 - Shows the Effect of ϕ for the Unthrottled and $\overline{VCF}$ for the Throttled Engine on BTE and BSFC with 5 lbm H_2O /lbm H_2 Water Injection.

Figure 33 - Shows the Effect of ϕ for the Unthrottled and $\overline{VCF}$ for the Throttled Engine on BTE and BESA with 5 lbm H_2O /lbm H_2 Water Injection.

Figure 34 - Shows the Effect of ϕ for the Throttled and $\overline{\text{VCF}}$ for the Unthrottled Engine on NO_x Emissions with 5 lbm $\text{H}_2\text{O}/\text{lbm H}_2$ Water Injection.

Figure 35 - Shows how the BTE and BESA vary with bmep for both the Throttled and Unthrottled Engine with 5 lbm H_2O /lbm H_2 Water Injection.

Figure 36 - Shows How the NO_x Emissions Vary with bmep for Both the Throttled and Unthrottled Engine with 5 lbm H_2O /lbm H_2 Water Injection.

Figure 37 - Shows that for the Same Diluent Heat Capacity (Quader, 1971), EGR and Water Injection Have Essentially the Same Effect on NO_x Emissions.

Figure 38 - Shows the BTE as a Function of ϕ For Pre IVC Unthrottled Engines as Obtained By Various Research Efforts.

Figure 39 - Compares the ITE Calculated from the BTE Data of Various Pre IVC Unthrottled Engines by the Methods of Section 12.1.2 (BESA; $1500 \leq \text{RPM} \leq 2040$; See Appendix 12 for Data).

Figure 40 - Compares the ITE Calculated from the BTE Data of Various Unthrottled Engines by the Methods of Section 12.1.2 (BESA; $2800 \leq \text{RPM} \leq 3500$; See Appendix 12 for Data).

Figure 41 - Compares the ITE Predicted by the Methods of Section 11.6 and that of Section 12.1.2.

APPENDIX 1 MAXIMUM POWER ESTIMATES FOR HYDROGEN-FUELED ENGINE CONFIGURATIONS

In this Appendix the details of the calculations performed to make the Table 2 estimates of potential maximum power levels of hydrogen-fueled engines are presented. To make these estimates, it was assumed that the volumetric efficiency of a gasoline-fueled engine is 75% at maximum horsepower (max hp) RPM and 95% at maximum torque (max T) RPM (Campbell, 1964). To account for the gains in volumetric efficiency accrued by proper design of hydrogen-fueled engine intake manifolds, cylinder heads and cams (Section 4.3.3), the gasoline engine volumetric efficiency is increased by 10% and 3% at the peak torque and peak horsepower RPMs respectively for engines 1 through 6 (Table A1). Since engines 7 through 10 induce air only during the beginning of the intake stroke, greater scavenging of residuals from the cylinder can be accomplished without loss of fuel out of the exhaust valve during valve overlap. Thus for engines 7 through 10 the volumetric efficiency is increased by an additional 5% at max hp RPM and an additional 2% at max T RPM. The 100% volumetric efficiency assumed for max T RPM is not an unreasonable one, in fact it is conservative; with gasoline-fueled engines, volumetric efficiencies as high as 116% can be achieved as found by Engleman (1973) and Vorum (1976). The same increase in volumetric efficiency has been assumed for engines 11 through 14. Since there are more stringent design constraints imposed on these engines, this assumption is felt to be an upper bound on volumetric efficiency for them.

For all of the Pre IVC engines there is a displacement of air due to the large volume* occupied by the gaseous hydrogen. For the stoichiometric mixture considered in Table 7, about 30%

* The total number of moles (volume) of air in a hydrogen-air mixture is equal to the total number of moles (air + hydrogen) times the partial pressure of the air. When hydrogen is present the partial pressure of the air is lowered thereby resulting in a reduction in the total number of moles of air.

of the air is displaced, resulting in a 70% reduction in effective volumetric efficiency. This reduction is reflected by the value of 0.7 assigned to the air displacement factor given in Table A1. For the Post IVC injection type engines 11, 12, 13 and 14 there is no air displacement by hydrogen.

Since there is 1500 BTU per lbm of air in a stoichiometric hydrogen-air mixture and 1300 BTU per lbm of air in a stoichiometric gasoline-air mixture the heating value per pound of air for a stoichiometric hydrogen-air mixture is 15% greater than that of a gasoline-air mixture. This 15% increase is accounted for by the heating value correction factor of 1.15 in Table A1.

EGR will reduce the effective volumetric efficiency. Assuming a 30% EGR rate results in the EGR factor of 0.70 shown in the table for engine configurations 2 and 5.

It should be pointed out that since (1) stoichiometric mixtures were assumed for all of the engines in Table A1 and (2) Post IVC engines, as a result of rough combustion, will probably not be able to operate at such mixtures: the peak power estimates presented are upper bounds that will probably not be attained with Post IVC engines.

The effective volumetric efficiency for each hydrogen engine is found by multiplying its hydrogen engine volumetric efficiency values by the factors. The ratio of the effective hydrogen engine volumetric efficiency to the gasoline engine volumetric efficiency times 100 then yields the estimated maximum powers presented in Table A1 and in Table 2.

TABLE A1 HYDROGEN ENGINE PEAK POWER ESTIMATE CALCULATIONS

Table 1 Engine Configuration No.	1, 3, 4, 6		2, 5		7, 8, 9, 10		11,12,13,14	
Operating Condition	Max T	Max hp	Max T	Max hp	Max T	Max hp	Max T	Max hp
Gasoline Engine Volumetric Efficiency	95%	75%	95%	75%	95%	75%	95%	75%
H ₂ Engine Volumetric Efficiency	98%	85%	98%	85%	100%	90%	100%	90%
Air Displacement by Hydrogen Factor	0.70	0.70	0.70	0.70	0.70	0.70	1.00	1.00
Heating Value Per lb of Air Factor	1.15	1.15	1.15	1.15	1.15	1.15	1.15	1.15
EGR Factor	1.00	1.00	0.70	0.70	1.00	1.00	1.00	1.00
Effective H ₂ Engine Volumetric Efficiency	79%	68%	55%	48%	81%	73%	115%	104%
<div>Maximum Power Hydrogen</div> <div> <div> <div></div> <div>x 100% =</div> </div> <div>Maximum Power Gasoline</div> </div>	83%	91%*	58%	64%	85%	97%	121%	138%

* Values given in Table 2 due to rounding errors

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As discussed in Section 3.2, one of the two major problems found with Pre IVC type hydrogen engines is NO_x emissions at high loads. During the planning stages of the present work an engine configuration was formulated which should allow for substantial control of NO_x emissions at high loads while maintaining the good thermal efficiency of an unthrottled engine at light load. This engine, termed the Discontinuity Engine, uses both throttling and equivalence ratio (quality) control as a means of varying the engine power output. The rationale behind the design of the Discontinuity Engine and a description of its operating characteristics follows.

In Figure A2a the NO_x emissions from both a throttled and an unthrottled engine are shown qualitatively as a function of brake mean effective pressure (bmep). Figure A2b shows the brake thermal efficiency as a function of bmep for the same engines. It is observed that the unthrottled engine exhibits high efficiency and low NO_x emissions at light loads. As the load is increased, however, serious problems arise due to the substantial increases observed in NO_x emissions for $\text{bmep} > X$. The operating characteristic of the Discontinuity Engine incorporates a combined method of load control: unthrottled control at light loads and throttling control at high loads. This dual mode of load control is accomplished by incorporating an automatically controlled device which causes a sudden decrease in manifold pressure, P_m , and a sudden increase in equivalence ratio, ϕ , at a value of bmep equal to X . The sudden decrease in P_m and sudden increase in ϕ , illustrated in Figures A2c and A2e, are such as to result in no net change in bmep and an acceptable level of NO_x at high loads. In order to accomplish this no net change, the hydrogen flow rate must be suddenly increased when the pressure is lowered as is illustrated in Figure A2d. The sudden reduction in P_m will be accomplished by a sudden change in position of the throttle plate located in the intake manifold. The magnitude of the required change in position will be a function of engine speed. The required change in position as well as the corresponding change in fuel flow will be determined

during the proposed work. This information could then be used to design the cams, linkages and pressure sensing venturis to make the sudden changes in Pm and fuel flowrate automatic.

With regard to the driveability of a Discontinuity Engine-powered automobile, it will only be affected for operation during passage through the bmep value of X. This passage is expected to reflect itself in a response no more serious than that experienced when conventional-fueled vehicles of circa 1965 and earlier vintage change from operation on primary to both primary and secondary carburetor circuits. Limited testing of the Discontinuity Engine at the discontinuity point will be performed during the present program.

Figure A2 - Illustrates Qualitatively the Characteristics of the Discontinuity Engine.

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APPENDIX 3 . THERMOCOUPLE INSTALLATION

Among the design constraints of thermocouple probe installation in the engine were: first that the probe be stationary and not vibrate loose during engine operation, and second that there be no leakage of oil out the rocker cover through which some of the probes were passed. These constraints were met by the method described in this Appendix and shown in Photograph A3.

1. A hole having the same diameter as the OD of the stainless-steel sheathed probes was drilled into the cylinder head to the point where it was desired to locate the tip of the probe.

2. The hole was enlarged (to $3/8$ in. diameter for the 0.040 in. OD probes) for a distance of approximately $1/2$ in. measured from the exterior surface of the engine.

3. The wire from the center of a $1/2$ in. length of Yellow Jacket high temperature ignition wire was removed and the outside layer of insulation was peeled away.

4. The remaining $1/2$ in. long tube or plug of inner insulation was worked into the enlarged portion of the hole.

5. Thermocouples were then installed by pushing them through the hole in the plug which originally housed the ignition wire.

This method satisfied the design constraints for no oil was ever found to leak past the plugs and thermocouples were kept in place during operation. In addition, the method was found to be very convenient for it allowed for easy removal and replacement of thermocouples.

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Photograph A3 - Illustrates the Method Used to Install Thermocouples

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APPENDIX 4 CONSTRUCTION OF LITHIUM-FILLED VALVES

The procedure used for converting sodium-filled Alfa Romeo exhaust valves (part no. 105390330100) to lithium-filled valves is outlined below:

1. The stem is drilled with a tungsten-carbide center drill and a 1/8 in. cobalt drill bit from the rocker end of the valve to the sodium-filled cavity.
2. A 2/0 taper pin reamer is then used to ream the drilled passage. Reamer compositions and hardnesses were unknown as purchased. Thus a Rockwell hardness tester was used to select a reamer of sufficient hardness (the required hardness was RC 64).
3. After reaming, the sodium is removed with a water bath and a 10% hydrochloric acid cleaning solution.
4. The stem is subsequently filled half-way with lithium. The lithium was purchased from Foote Mineral Company and is in the form of 1/8 in. diameter wire. The volume of the stem was measured with a buret and the lithium was weighed for proper filling.
5. 416-2/0 x 3/4 in. long hardened tapered pins (Almetal Screw Co.) are then brazed into the reamed hole in the valve stem with Eutectic 1020 XFC brazing rod. This pin was chosen from available pins based on hardness, thermal expansion coefficient compared to that of the exhaust valve (as measured at 2°F and 212°F), and its chromium content (wear resistance). The Eutectic 1020 XFC brazing alloy was chosen on the basis of its high wettability, high strength and low brazing temperature.
6. The final step is to grind the rocker-arm end of the valve to the proper finish.

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Original plans called for the use of the EGR separation and quantification method described below. For reasons given there, it was discarded. It is felt, however, that for some situations the method is a good one and thus the presentation of it.

It is difficult to make accurate measurements of EGR flow rate with conventional flowmeters. The gas to be measured is exhaust gas which changes in temperature, pressure and composition, as the engine is varied over the operating range of concern. For example: exhaust gas temperatures at the exhaust port vary from 300°F to 1050°F ; the exhaust gas pressure is dependent on EGR passage design, and for a given design, pressure variations will occur at the flowmeter with changes in load, fuel-air ratio and EGR flowrate; the constituents of the exhaust gas can vary from approximately 65% N_2 and 35% H_2O at the stoichiometric air-fuel ratio to 78% N_2 , 18% O_2 and 4% H_2O at the lean limit of combustion.

The use of conventional flowmeters will also result in substantial changes in the condition of the gas used for EGR. For example, the gas would have to be cooled which may also result in condensation of water vapor, which in turn will change the NO_x -reducing characteristics of the EGR gas.

Because of the above-described problems associated with the use of conventional flowmeters to measure EGR flowrate, alternate methods of EGR measurement were evolved: (1) one involved the separation of the exhaust gas into separate streams of equal flowrate (Section 7.7) and was the one used in the present study, (2) the other method involved the measurement of oxygen concentration in the EGR exhaust gas, $[\text{O}_2]_{\text{EGR}}$, and in the fresh charge composed of hydrogen, fuel and EGR gas, $[\text{O}_2]_{\text{mix}}$, as well as the measurement of air volume flowrate, $\dot{V}_a$, and hydrogen volume flowrate, $\dot{V}_h$. The following equation was derived and can be used to calculate the % EGR from these measurements.

$$\% \text{ EGR} = 100 \frac{F_a[O_2]_a - F_h[O_2]_m - F_a[O_2]_m}{F_a[O_2]_a - F_h[O_2]_{\text{EGR}} - F_a[O_2]_{\text{EGR}}} \quad (1)$$

In Equation (1) the F_i 's are nondimensional volume flows defined by

$$F_i = \frac{\dot{V}_i}{\dot{V}_m} \quad (2)$$

where $\dot{V}_m$ is the fresh charge volume flowrate (air plus hydrogen plus EGR gas). The fresh charge volume flowrate is given in terms of measured parameters by the following equation:

$$\dot{V}_m = \frac{[O_2]_{\text{EGR}} (\dot{V}_a + \dot{V}_h) - \dot{V}_a [O_2]_a}{[O_2]_{\text{EGR}} - [O_2]_m} \quad (3)$$

In deriving Equations (1) and (3), it was assumed that the air, hydrogen and EGR gas are ideal gases at equal temperature and pressure before mixing.

An analysis was performed to determine the maximum experimental uncertainties expected using the above-described $[O_2]$ method of % EGR determination. In the analysis the following uncertainties were assumed:

$\dot{V}_a$	:	± 0.0075 reading
$\dot{V}_h$	:	± 0.0050 reading
$[O_2]_{\text{mix}}$	:	± 0.0005 oxygen fraction
$[O_2]_{\text{EGR}}$	:	± 0.0005 oxygen fraction.

Room air was assumed to be composed of 21% oxygen by volume. Table A5 contains the result of the analysis. For each of the methods ($[O_2]$ method or $\dot{V}_{\text{EGR}}$ method) two numbers are given for each combination of Y and % EGR. These numbers give the upper and lower bounds on the measured % EGR. For example, at $Y=0.85$ and with 25% EGR, the value of % EGR: determined by the $[O_2]$ method will lie between 24.2% and 25.8%; determined by the $\dot{V}_{\text{EGR}}$ method will lie between 23.3% and 26.7%. In estimating the uncertainty for the $\dot{V}_{\text{EGR}}$ method the uncertainty in the measurement of EGR flow rate was assumed to be 6% or the reading.

The uncertainty given is labeled maximum because the calculations were performed assuming all of the measured parameters to be at their extreme values to yield either the maximum or minimum value of % EGR. The results in the table show initially that the above-described method of determining % EGR should have been a good one. Three problems arose casting doubt on the feasibility of the originally-proposed method:

1. Measurements made with the test engine showed mixture nonuniformity in the intake manifold even with the Beam gas-mixer carburetor (See Section 7.4). Thus a representative sample would be difficult to obtain for the $[O_2]_m$ measurement.
2. The fuel and air flow meters selected were supposed to have an accuracy of $\pm 0.5\%$. The final calibration, however, showed an accuracy of around $\pm 1.3\%$ for both the fuel flow and air flow meters thus increasing the uncertainty in the fuel and air flow measurements needed for the method.
3. Reverse flow out of the cylinder and into the intake manifold when the intake valve opens can increase the apparent rate of EGR by giving a non-representative sample for the $[O_2]_m$ measurement.

These three problems, when taken together, were considered to be sufficiently severe to warrant a search for a better means of controlling and quantifying EGR rates. This search resulted in the Section 7.7 method.

TABLE A5

Illustrates the Maximum Uncertainty Expected in
 % EGR Determination by Oxygen Concentration
 Measuring Method, $[O_2]$, and Direct EGR Flow Rate
 Measuring Method $\dot{V}_{EGR}$.

Y^*	1.00		0.85		0.60	
% EGR	$[O_2]$	$\dot{V}_{EGR}$	$[O_2]$	$\dot{V}_{EGR}$	$[O_2]$	$\dot{V}_{EGR}$
5	4.3	4.7	4.2	4.6	4.0	4.6
	5.7	5.3	5.8	5.4	6.0	5.4
15	14.3	14.0	14.2	14.0	14.1	13.5
	15.7	16.0	15.8	16.0	15.9	16.5
25	24.3	23.4	24.2	23.3	24.1	22.5
	25.7	26.7	25.8	26.7	25.9	27.5
35	34.3	32.7	34.3	32.6	34.1	31.5
	35.7	37.3	35.7	37.4	35.9	38.5
45	44.3	42.0	44.3	41.9	44.1	40.5
	45.7	48.0	45.7	48.1	45.9	49.5

$$\begin{aligned}
 Y^* &= \frac{\text{Volume Hydrogen/Volume Oxygen}}{(\text{Volume Hydrogen/Volume Oxygen})_{\text{Stoichiometric}}} \\
 &= 0.5 \frac{(\text{Volume Hydrogen})}{(\text{Volume Oxygen})}
 \end{aligned}$$

APPENDIX 6 CERTIFICATES OF INSTRUMENT CALIBRATION

Instruments obtained for the required measurements made during this project were calibrated at the factory traceable to the National Bureau of Standard whenever possible. Certified standards were obtained for some of the instruments calibrated at the University of Miami. The following are copies of Calibration for the instruments and standards.

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A6.6 - Certificate of Calibration For The Certified Thermometer; See Section 9.4.1.	192
A6.7 - Certificate of Calibration For The Rotameters Used For Fuel Flowrate Measurements; See Section 8.1.1.	193

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A6.1

MERIAM INSTRUMENT 10920 Madison Avenue • Cleveland, Ohio 44102 U.S.A. • 216/281-1100

LETTER OF CERTIFICATION
OF
TRACEABILITY TO NATIONAL BUREAU OF STANDARDS

50MC2-2F
Serial No. L-16791
Date: 11-2-77

This is to certify that the Flow of LFE was within 1/2% of NDS Master

on Meriam Model 50MC2-2F

Furnished under customer purchase order No. 51592

is traceable to National Bureau of Standards,

TEST NUMBERS 213-31/190522

MERIAM INSTRUMENT
Div. of The Scott & Fetzer Company

Signed [Signature]

Date 11-2-77

Flow Curve CU-9853-C

11-2-77

University of Miami

Certificate of Calibration For The Laminar Flow Element Used To
Measure Engine Air Flow; See Section 8.1.2.

A6.2

CERTIFICATE OF ACCURACY

This is to certify that the following Certified Flowmeter Model
274-274 Serial 274-3 has been individually
 calibrated on a COX Primary Standard Flowmeter Calibrator
 traceable to the U.S. National Bureau of Standards, and is
 hereby certified to be accurate to 1 % of reading of
100 gpm when used with 1000 specific gravity
1000 viscosity 200 cSt at 77 °F.

Date 4/29/77 Henry R. R.
 Manager—Quality Control

COX INSTRUMENT DIVISION LYON CORP.
 15700 Redwood Avenue, Dearborn, Michigan 48127
 Phone (313) 830-5700 Telex 250430

Certificate of Calibration For The Rotameter Used To Measure
 Water Flowrates Through the Water Injectors; See Section 7.6.

A6.3

CERTIFICATION

CUSTOMER: University of Miami
 ORDER NO: PD-07334
 MERIAM JOB NO: L-1679
 SUBJECT: Certificate of Traceability to NBS

WE CERTIFY THAT PARTS COVERED BY THIS ORDER HAVE BEEN CONTROLLED AND PROCESSED IN ACCORDANCE WITH MIL-Q-9858 QUALITY CONTROL REQUIREMENTS.

WE CERTIFY THAT EACH SCALE WAS REPRODUCED FROM A GLASS MASTER, TRACEABLE TO THE U. S. BUREAU OF STANDARDS, REPORT NO. TML 48686 & 24/169573.

FOR "RC" TYPE SCALE

Scale tolerance for each one inch of scale length $\pm .002"$.
 Cumulative tolerance for 100" scale of scale length $\pm .015 - .035$.

FOR "SC" TYPE SCALE

Scale tolerance for each one inch of scale length $\pm .002$.
 Cumulative tolerance for 100" scale of scale length $\pm .001 - .005$.

1. when observed at 22°C (71.6°F)
2. all well type scales are foreshortened to compensate for manometer well drop
3. all "U" type scale are non compensated for well drop.

8" Range, 50" Scale length
 Model 40HE35WM
 scale graduated inches and
 hundredths
 Serial No. L16792

THE MERIAM INSTRUMENT COMPANY
 SCALE DIVISION
 CLEVELAND, OHIO 44102

BY: M. E. Saltgaver
 DIVISION SUPERVISOR

DATE: 5-25-77

Certificate of Calibration For The Water Manometer Used In
 Conjunction With The Laminar Flow Element; See Section 8.1.2.

E-729

2

THE MERIAM INSTRUMENT Co.
 CLEVELAND, OHIO, U. S. A.

DATE RECEIVED BY
 P. H. C. F.

DATE RECEIVED BY
 P. H. C. F.

DATE RECEIVED BY
 P. H. C. F.

DATE RECEIVED BY
 P. H. C. F.

A6.4

WALLACE & TIERNAN DIVISION
MEASUREMENT & CONTROL

25 MAIN STREET, BELLEVILLE, NEW JERSEY 07109 • (201) 759-8000

REPLY TO: P. O. BOX 178
NEWARK, N. J. 07101

DATE 11/2/77

CERTIFICATE OF COMPLIANCE

CUSTOMER University of Miami at Coral Gables

P.O. NO. 62837

SHIPPING ORDER J 20695 Shipped 3/7/67

STANDARD BAR FU 4320 LENGTH WAS CALIBRATED WITH WALLACE & TIERNAN
STANDARD #1956

This is to certify that materials and processes involved in the manufacture and verification of the product(s) included in this shipment comply with the catalog, drawing or specification referenced in the order.

It is certified further that the calibration is traceable to the National Bureau of Standards #232.12/212296.

Wallace & Tiernan Division
PENNWALT Corporation

A. S. Gaffney
A. Gaffney, Manager
Test & Inspection

Certificate of Calibration For The Standard Bar Used To
Calibrate The Mercury Manometer; See Section 8.1.1.

CERTIFICATE of CALIBRATION

The John Fluke Mfg. Co., Inc. hereby certifies that this instrument meets or exceeds all published specifications and has been calibrated using standards whose accuracies are traceable to the National Bureau of Standards within the limitations of the Bureau's calibration services, or have been derived from accepted values of natural physical constants, or have been derived by the ratio type of self-calibration techniques.

Applicable NBS Test Reports:

DC Voltage — 213285

AC Voltage — 212126 & 807675

Resistance — 210508

John F. Hollman
STANDARDS ENGINEER
A. J. Vitale
MANAGER QUALITY ASSURANCE

COMPLETE AND RETURN THIS CARD TO ASSURE YOUR WARRANTY

Certificate of Calibration For Fluke 8600A Digital Multimeter;
See Section 9.4.1.

PRINCO INSTRUMENTS INC.

SOUTHAMPTON INDUSTRIAL PARK • SOUTHAMPTON, PA. 18966

FACTORY CERTIFICATE

This is to certify that the Instrument herein described has been tested, by comparison with standards certified by the Bureau of Standards, by approved methods, and that the following corrections to observed readings have been determined on this date.

INSTRUMENT		SERIAL NO.	DATE
77-1		356118	11/9/77
SCALE RANGE		DIVISIONS	IMMERSION
25 to 140°F		1/5	Total
POINT OF TEST	OBSERVED READING	CORRECTION	
32	+32.20	-.20	
50	50.20	-.20	
68	68.00	0.00	
86	86.00	0.00	
104	104.00	0.00	
122	122.00	0.00	
140	140.00	0.00	
NBS 204612		OBSERVER	
		JC	

Certificate of Calibration For The Certified Thermometer
See Section 9.4.1.

CEESI

A6.7

COLORADO
ENGINEERING EXPERIMENT STATION
INC.

OFFICE:
P. O. Box 344
Boulder, Colo. 80302
Phone: 303-443-1344

LABORATORY:
P. O. Box 41
Nunn, Colo. 80648
Phone: 303-897-2340

Date: May 31, 1977

This is to certify that the accuracy of the calibration
on Cox Variable Area Flowmeter Set

Model No. 129-287 Serial No. 16332-92,94,96

is within plus or minus one-half percent of reading.

Walter F. Seidel

Acting President

Certificate of Calibration For The Rotameters Used For Fuel
Flowrate Measurements; See Section 8.1.1.

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APPENDIX 7 DATA-REDUCTION COMPUTER PROGRAM AND EXPERIMENTAL
RESULTS*

See Section 10 for a Description of this Appendix.

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A7.4 - Experimental Results Table: Experimental Results For Each Of The Data Points (TEST NO.) Are Presented Along With The Table 1 Engine Configuration No. (ENG) And The Table 7 Experiment Number (EXP) See Section 10 For Detailed Description.	213

* See Appendix 8 For Exhaust-Gas Thermocouple Temperature
Corrections.

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A7.1 DEFINITIONS OF FORTRAN SYMBOLS USED

<u>Fortran Symbol</u>	<u>Description</u>	<u>Units</u>
A	Computational Symbol	---
AIRTMP	Temperature of Room Air	$^{\circ}\text{F}$
ALINE	Computational Symbol	---
ATMPRS	Atmospheric Pressure	psi
bmep	Brake Mean Effective Pressure	psi
BSNO	Brake Specific NO	g/hp hr
BSNO ₂	Brake Specific NO ₂	g/hp hr
BSNO _x	Brake Specific NO _x	g/hp hr
BTE	Brake Thermal Efficiency	%
COMB EFF	H ₂ Utilization Coefficient	%
E	Flowrate H ₂ in Exhaust Gases in Exhaust pipe at 70°F and 14.7psi	ft ³ /min
ENG, EXP	Table 1 Engine Configuration No., Table 7 Experiment Number	---
EM	Emissivity of Thermocouple	---
F	Flowrate O ₂ in Exhaust Gases in Exhaust pipe at 70°F and 14.7psi.	ft ³ /min
FRADRY	Volumetric flowrate of dry air at 70°F and 14.7psi.	ft ³ /min
FRAWET	Volumetric flowrate of wet air at 70°F and 14.7psi.	ft ³ /min
FREHP	Flowrate exhaust gases at exhaust pipe in 70°F and 14.7psi.	ft ³ /min
FREXPT	Flowrate Exhaust at Port	ft ³ /min
G	Flowrate N ₂ in Exhaust Gases in Exhaust Pipe at 70°F and 14.7psi.	ft ³ /min
H	Flowrate H ₂ O in Exhaust Gases in Exhaust Pipe at 70°F and 14.7psi.	ft ³ /min
H2OH2	Mass ratio of Water Injected to Hydrogen	---
H2O PCT	Percent Water Vapor in Exhaust Gases	%
HIPO	Heat Transfer Coefficient at Inside Surface of Exhaust Manifold	$\frac{\text{Btu/hr}}{\text{ft}^2 \text{ } ^{\circ}\text{R}}$
HO	Heat Transfer Coefficient at Outer Surface of Exhaust Manifold	$\frac{\text{Btu/hr}}{\text{ft}^2 \text{ } ^{\circ}\text{R}}$
HPCT	Percent Hydrogen in Exhaust Gases	%

A7.1 (Cont.) DEFINITIONS OF FORTRAN SYMBOLS USED

<u>Fortran Symbol</u>	<u>Description</u>	<u>Units</u>
HTHO	Heat Transfer Coefficient of Thermo- couple	$\frac{\text{Btu/hr}}{\text{ft}^2 \text{ } ^\circ\text{R}}$
KH	Thermal conductivity of Hydrogen at conditions in Exhaust Port	$\frac{\text{Btu/hr}}{\text{ft } ^\circ\text{R}}$
KH2O	Thermal conductivity of water vapor at conditions in Exhaust Port	$\frac{\text{Btu/hr}}{\text{ft } ^\circ\text{R}}$
KN	Thermal conductivity of Nitrogen at conditions in Exhaust Port	$\frac{\text{Btu/hr}}{\text{ft } ^\circ\text{R}}$
KO	Thermal conductivity of Oxygen at conditions in Exhaust Port	$\frac{\text{Btu/hr}}{\text{ft } ^\circ\text{R}}$
KTOT	Thermal Conductivity of gas mixture in Exhaust Port	$\frac{\text{Btu/hr}}{\text{ft } ^\circ\text{R}}$
LOOPS	Computational Symbol	---
MDOTCP	Diluent Heat Capacity	$\frac{\text{Btu}}{^\circ\text{R min}}$
MI	Computational Symbol	---
MISFIR	Percent of Cycles which have SSLII-Induced Misfire	%
MUTOT	Density of gas mixture in exhaust port	lbm/ft^3
NPCT	Percent Nitrogen in Exhaust gases	%
NUP	Nusselt number of flow in exhaust port	---
O2 CLY	Fraction of Oxygen in dry exhaust gas from given cylinder	---
O2 DRY	Fraction of O ₂ in dry exhaust gas	---
O2 WET	Fraction of Oxygen in wet exhaust gas	---
OPCT	Percent Oxygen in exhaust gases	%
PCEGR	Percentage of gases which are EGR-re- cycled exhaust gases at carburetor	%
PCH2	Percent H ₂ in the dry fuel-air mixture	%
PHI	Fuel-Air Equivalence Ratio	---
PPNO	Parts per million Nitric Oxide	ppm
PPNOX	Parts per million NO _x	ppm
REP	Reynolds number in Exhaust Port	---
ROH	Density of Hydrogen at conditions in exhaust port	lbm/ft^3
ROH2O	Density of water vapor at conditions in exhaust port	lbm/ft^3

A7.1 (Cont.) DEFINITIONS OF FORTRAN SYMBOLS USED

<u>Fortran Symbol</u>	<u>Description</u>	<u>Units</u>
ROO	Density of Oxygen at conditions in in exhaust port	lbm/ft ³
ROTOT	Density of gas mixture in exhaust port	lbm/ft ³
SA	Spark Advance before top dead center	degrees
SFH	Viscosity of Hydrogen Gas at conditions in exhaust port	lbm/sec ft
SFH2O	Viscosity of water vapor at conditions in exhaust port	lbm/sec ft
SFN	Viscosity of Nitrogen Gas at conditions in exhaust port	lbm/sec ft
SFO	Viscosity of Nitrogen Gas at conditions in exhaust port	lbm/sec ft
T	Measured thermocouple temperature	°F
TEST NO.	Number of Particular Test	---
TDEL	Computed temperature difference between thermocouple and exhaust gas	°F
TEXHI	The corrected exhaust-gas temperature	°F
TGP	Temperature guide	°F
TH2O	Temperature of the cooling water as it leaves the block	°F
TORK	Torque	ft lb _f
TS	Temperature of exhaust manifold	°F
TSEAT	Temperature exhaust seat (T, Seat j; Fig 3)	°F
TUBES	Number of tubes i in the EGR Separator which pass exhaust gas up to the carburetor (See Figure 5)	---
UPSILON	Hydrogen-Oxygen Equivalence Ratio	---
VAPPRS	Water-Vapor pressure in Atmospheric Air	psi
VCFWET	Percentage of the cylinder that would be filled with the mixture and recycled exhaust gases at standard temperature and pressure- See Nomenclature (Section 16) for Definition of VCF.	%

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A7.2

COMPUTER PROGRAM

See Section 10 for Description.

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A7.2

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G35322086*TPFS(0).FINAL/PRGM SYM CREATED ON 27 JUL 78 AT 13:54:06
1  DIMENSION TESTNO(10),TORK(10),RPM(10),PPNOX(10),PPNO(10),O2DRY(10),
2      1 ,FRH2(10),FRAWET(10),FRADRY(10),TUBES(10),H2OH2(10),ENGEXP(96)
3      2 ,RMEP(10),VCFWET(10),PCH2(10),PHI(10),UPSIL(10),BTE(10)
4      3 ,RSNOX(10),O2WET(10),E(10),F(10),H(10),G(10),FREXHP(10)
5      4 ,COMBEF(10),BSNO(10),BSNO2(10),PCEGR(10),SA(10)
6      5 ,HPCT(10),OPCT(10),H2OPCT(10),ALINE(96)
7      6 ,TSEAT(4,10),TH2O(10),TGD(3,10),AIRTMP(10)
8      7 ,ATMPRS(10),VAPPRS(10),MISFIR(10),A(3),O2CLY(4,10),M1(4)
9      REAL NPCT(10),MDOTCP(10)
10     DIMENSION TTH(10),T(4,10),TEST(10)
11     * ,SFH(10),SFN(10),SFO(10),SFH2O(10),
12     * ROH(10),RON(10),ROO(10),ROH2O(10),ROTOT(10),REP(10)
13     * ,FREXPT(10),HIPO(10),
14     * TS(10),HTHO(10),
15     * TDEL(10),HO(10),TEXH1(4,10),EM(10),LOOPS(10)
16     REAL KH(10),KN(10),KO(10),KH2O(10),NUP(10),MUTOT(10),
17     * KTOT(10)
18     DATA A /'A','B','C'/
19     DATA M1 /1,2,3,4/
20     ISTOP=0
21     10 CONTINUE
22     IEGR=0
23     IH2OH2=0
24     IMISFR=0
25     DO 60 I=1,10
26     -N=I
27     IE=(I-1)*7+1
28     IE=IB+6
29     READ(5,90) TESTNO(I),(ENGEXP(K),K=IB,IE)
30     90 FORMAT(F8.0,7A1)
31     IF(TESTNO(I).EQ.-1.) GO TO 70
32     IF(TESTNO(I).EQ.-2.) GO TO 20
33     READ(5,100) TESTNX,TORK(I),RPM(I),PPNOX(I),PPNO(I),O2DRY(I),
34     1 FRH2(I),FRAWET(I),FRADRY(I),TUBES(I),H2OH2(I),SA(I)
35     IF(TESTNO(I).NE.TESTNX) WRITE(6,1130) M1(1),TESTNO(I)
36     IF(H2OH2(I).GT.0.0) IH2OH2=1
37     READ(5,1010)TESTNX,(T(J,I),J=1,4),(O2CLY(K,I),K=1,4)
38     IF(TESTNO(I).NE.TESTNX) WRITE(6,1130) M1(2),TESTNO(I)
39     1010 FORMAT(F5.0,4(F5.0,1X),4F7.0)
40     READ(5,137) TESTNX,(TSEAT(K,I),K=1,4),TH2O(I)
41     1 (TGD(J,I),J=1,3)
42     IF(TESTNO(I).NE.TESTNX) WRITE(6,1130) M1(3),TESTNO(I)
43     1130 FORMAT(10X,'DATA ERROR *****',I3,F5.0)
44     READ(5,137) TESTNX,AIRTMP(I),ATMPRS(I),VAPPRS(I)
45     1 ,MISFIR(I)
46     IF(TESTNO(I).NE.TESTNX) WRITE(6,1130) M1(4),TESTNO(I)
47     137 FORMAT(10F8.0)
48     IF(MISFIR(I).GE.0.0) IMISFR=1
49     IF(PPNOX(I).EQ.-1.) PPNOX(I)=1.0E10
50     IF(O2DRY(I).EQ.-1.) O2DRY(I)=1.0E10
51     IF(PPNO(I).EQ.-1.) PPNO(I)=1.0E10
52     100 FORMAT(2F5.0,7F8.0,4F4.0,2F5.0)
53     F(I)=(FRH2(I)+(FRADRY(I)*1.76/4.76))/((1.0/(O2DRY(I)+1.0E-8))-3.)
54     IF(F(I).LT.0.0) F(I)=1.0E10
55     H(I)=((2.0/4.76)-1.0)*FRADRY(I)+FRAWET(I) - 2.0*F(I)
56     E(I) = 2.0*F(I) + FRH2(I) - FRADRY(I)*2.0/4.76

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A7.2 Cont.

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57      G(I)=(FRADRY(I)*3.76)/4.76
58      FREXHP(I) = E(I) + F(I) + H(I) + FRADRY(I)*3.76/4.76
59      1      +H2OH2(I)*FRH2(I)/9.0
60      O2WET(I) = F(I)/FREXHP(I)
61      IF(O2DRY(I).GT.100.) O2WET(I)=1.0E10
62      COMBEF(I) = (FRH2(I) - E(I))/FRH2(I)
63      BSNO(I)=(FREXHP(I)*PPNO(I)*35.168*60.*5252.)/(TORK(I)* RPM(I)
64      *1000000.)
65      BSNO2(I)=(FREXHP(I)*(PPNOX(I)-PPNO(I))*53.92*60.*5252.)/
66      1      (TORK(I)*RPM(I)*1000000.)
67      BSNOX(I)=BSNO(I)+BSNO2(I)
68      IF(PPNOX(I).GT.1.0E6) BSNO2(I)=1.0E10
69      PCH2(I)=FRH2(I)/(FRH2(I)+FRADRY(I))*100.
70      PHI(I)=FRH2(I)/(FRADRY(I)*.4202)
71      VCFWET(I)=(FRH2(I)+FAWET(I)+(FREXHP(I)
72      1      *(TUBES(I)/(19.0-TUBES(I)))))/(RPM(I)*.028038)
73      BTE(I)= (TORK(I)*RPM(I)*.000030056)/FRH2(I)
74      BMEP(I)=TORK(I)*1.5562
75      UPSIL(I)=(FRH2(I))/((O2WET(I)*FPEXHP(I)*TUBES(I)/(19.0-TUBES(I))+
76      1      (FRADRY(I)/4.76))*2.)
77      PCEGR(I)=(FREXHP(I)*(TUBES(I)/(19.-TUBES(I))))/
78      1      (FRH2(I)+FAWET(I)+FREXHP(I)*TUBES(I)/(19.-TUBES(I)))
79      IF(TUBES(I).GT.0.0) IEGR=1
80      HPCT(I)=E(I)/FREXHP(I)
81      NPCT(I)=G(I)/FREXHP(I)
82      OPCT(I)=F(I)/FREXHP(I)
83      H2OPCT(I)=(H(I)+H2OH2(I)*FRH2(I)/9.0)/FPEXHP(I)
84      MDGTCPI(I)=(FRH2(I)*(2.761/540.)*H2OH2(I)*0.682)+
85      1      (FREXHP(I)*TUBES(I)/(19.-TUBES(I)))*
86      2      ((HPCT(I)*(2.761/540.)*4.16)+(NPCT(I)*(38.37/
87      3      540.)*0.311)+(OPCT(I)*(43.83/540.)*0.207)+
88      4      (H2OPCT(I)*(24.68/540.)*0.682))
89      TTH(I)=((T(1,I)+T(2,I)+T(3,I)+T(4,I))/4)+460.
90      TDELCR=0.0
91      TEST(I)=TTH(I)
92      LOOPNO=1
93      1001  EM(I)=.85
94      SIGMA=0.1714E-08
95      IF((RPM(I).GE.700.).AND.(RPM(I).LE.900.)) H0(I)=10.0
96      IF((RPM(I).GE.1600.).AND.(RPM(I).LE.1900.)) H0(I)=12.0
97      IF((RPM(I).GE.3200.).AND.(RPM(I).LE.3400.)) H0(I)=16.0
98      FREXPT(I)=(FREXHP(I)*(19./(19.-TUBES(I))))/4.0
99      SFH(I)=((3.250E-07*(TEST(I)**1.5))/(TEST(I)+127.))
100      SFN(I)=((6.950E-07*(TEST(I)**1.5))/(TEST(I)+184.))
101      SF0(I)=((8.260E-07*(TEST(I)**1.5))/(TEST(I)+190.))
102      SFH20(I)=((9.170E-07*(TEST(I)**1.5))/(TEST(I)+1185.))
103      ROH(I)=(2.761/TEST(I))
104      RON(I)=(38.37/TEST(I))
105      ROG(I)=(43.83/TEST(I))
106      ROH20(I)=(24.68/TEST(I))
107      KH(I)=(0.09945*((TEST(I)/492.)**0.78))
108      KN(I)=(0.01398*((TEST(I)/492.)**0.80))
109      KO(I)=(0.01418*((TEST(I)/492.)**0.87))
110      KH20(I)=(0.008736*((TEST(I)/492.)**1.48))
111      MUTOT(I)=SFH(I)*HPCT(I)+SFN(I)*NPCT(I)+SF0(I)*OPCT(I)
112      1      +SFH20(I)*H2OPCT(I)
113      ROTOT(I)=ROH(I)*HPCT(I)+RON(I)*NPCT(I)+ROG(I)*OPCT(I)

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A7.2 Cont.

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114      1      +ROH2O(I)*H2OPCT(I)
115      KTOT(I)=KH(I)*HPCT(I)+KN(I)*NPCT(I)+KO(I)*OPCT(I)
116      1      +KH2O(I)*H2OPCT(I)
117      REP(I)=(FREXPT(I)*TEST(I)*ROTOT(I))/(995.*MUTOT(I))
118      NUP(I)=(1.2+((REP(I)-3000.)/57000.)*.3)*(0.0243*(REP(I)**0.78))
119      1      *(1.7)
120      HIPO(I)=(NUP(I)*KTOT(I))/0.0994
121      TS(I)=((TEST(I)*HIPO(I)*.35)+(580.0*H0(I)))/((HIPO(I)*.35)+H0(I))
122      TENV=(.95*TS(I))+(.05*TTH(I))
123      HTHO(I)=KTOT(I)*(118.0*((0.0524*REP(I))**0.466))*1.5
124      TDEL(I)=(SIGMA*EM(I))*(TTH(I)**4-TENV**4)/(HTHO(I)*.35)
125      TDIFF2=ABS(TDEL(I)-TDELCR)
126      IF(TDIFF2.LE.1.0) GO TO 1000
127      TDELCR=((TDEL(I)-TDELCR)/2.0)+TDELCR
128      TEST(I)=TTH(I)+TDELCR
129      LOOPNO=LOOPNO+1
130      IF(LOOPNO.GT.25) GO TO 999
131      GO TO 1001
132      999      TDEL(I)=1.0E10
133      1000      CONTINUE
134      LOOPS(I)=LOOPNO
135      DO 1111 J=1,4
136      1111      TEXH1(J,I)=T(J,I)+TDEL(I)
137      BTE(I)=BTE(I)*100.
138      O2DRY(I)=O2DRY(I)*100.
139      O2WET(I)=O2WET(I)*100.
140      VCFWET(I)=VCFWET(I)*100.
141      PCEGR(I)=PCEGR(I)*100.
142      60      CONTINUE
143      GO TO 80
144      20      N=N-1
145      GO TO 80
146      70      N=N-1
147      ISTOP=1
148      80      WRITE(6,101) (TESTNO(K),K=1,N)
149      NCOL=N*7
150      WRITE(6,138) (ENGEXP(K),K=1,NCOL)
151      WRITE(6,133) (AIRTMP(K),K=1,N)
152      WRITE(6,134) (ATMPRS(K),K=1,N)
153      WRITE(6,135) (VAPPRS(K),K=1,N)
154      IF(IMISFR.EQ.1) WRITE(6,129) (MISFIR(K),K=1,N)
155      IF(IH2OH2.EQ.1) WRITE(6,126) (H2OH2(K),K=1,N)
156      IF(IEGR.EQ.1) WRITE(6,127) (PCEGR(K),K=1,N)
157      WRITE(6,103) (RPM(K),K=1,N)
158      WRITE(6,110) (BMEP(K),K=1,N)
159      WRITE(6,128) (SA(K),K=1,N)
160      WRITE(6,111) (VCFWET(K),K=1,N)
161      WRITE(6,112) (PCH2(K),K=1,N)
162      WRITE(6,123) (PHI(K),K=1,N)
163      WRITE(6,124) (UPSIL(K),K=1,N)
164      WRITE(6,113) (BTE(K),K=1,N)
165      REWIND 10
166      DO 300 M = 1,N
167      IF(PPNOX(M).LT.1.0E6) GO TO 311
168      WRITE(10,2006)
169      GO TO 300
170      311      IF(PPNOX(M).LT.99.5) GO TO 310

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A7.2 Cont.

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171      WRITE(10,2001) PPN0X(M)
172      GO TO 300
173      310 IF (PPNOX(M).LT.9.95) GO TO 320
174      WRITE(10,2002) PPN0X(M)
175      GO TO 300
176      320 WRITE(10,2003) PPN0X(M)
177      300 CONTINUE
178      END FILE 10
179      REWIND 10
180      READ(10,9999) (ALINE(K),K=1,NCOL)
181      9999 FORMAT(7A1)
182      2001 FORMAT(F6.0,1X)
183      2002 FORMAT(F5.1,2X)
184      2003 FORMAT(F5.2,2X)
185      WRITE(6,104) (ALINE(K),K=1,NCOL)
186      REWIND 10
187      DO 330 M = 1,N
188      IF (PPNO(M).LT.1.0E6) GO TO 341
189      WRITE(10,2006)
190      GO TO 330
191      341 IF (PPNO(M).LT.99.5) GO TO 340
192      WRITE(10,2001) PPNO(M)
193      GO TO 330
194      340 IF (PPNO(M).LT.9.95) GO TO 350
195      WRITE(10,2002) PPNO(M)
196      GO TO 330
197      350 WRITE(10,2003) PPNO(M)
198      330 CONTINUE
199      END FILE 10
200      REWIND 10
201      READ(10,9999) (ALINE(K),K=1,NCOL)
202      WRITE(6,105) (ALINE(K),K=1,NCOL)
203      REWIND 10
204      DO 360 M = 1,N
205      IF (BSNOX(M).LT.99.5) GO TO 370
206      WRITE(10,2004)
207      2004 FORMAT(7H UNDEF. )
208      GO TO 360
209      370 IF (BSNOX(M).LT.9.95) GO TO 380
210      WRITE(10,2007) RSN0X(M)
211      GO TO 360
212      380 WRITE(10,2003) BSNOX(M)
213      360 CONTINUE
214      END FILE 10
215      REWIND 10
216      READ(10,9999) (ALINE(K),K=1,NCOL)
217      WRITE(6,114) (ALINE(K),K=1,NCOL)
218      WRITE(6,106) (O2DRY(K),K=1,N)
219      WRITE(6,115) (O2WET(K),K=1,N)
220      DO 5300 J=1,4
221      REWIND 10
222      DO 5400 M=1,N
223      IF (O2CLY(J,M).NE.-1.) GO TO 30
224      WRITE(10,2006)
225      GO TO 5400
226      30 WRITE(10,2005) O2CLY(J,M)
227      2005 FORMAT(F5.1,2X)

```

A7.2 Cont.

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228      2006 FORMAT(7H  N/A  )
229      5400 CONTINUE
230      END FILE 10
231      REWIND 10
232      READ(10,9999) (ALINE(K),K=1,NCOL)
233      WRITE(6,136) J,(ALINE(K),K=1,NCOL)
234      5300 CONTINUE
235      DO 5500 J=1,4
236      5500 WRITE(6,1128) J,(T(J,K),K=1,N)
237      DO 5000 J=1,4
238      5000 WRITE(6,1129) J,(TEXH1(J,K),K=1,N)
239      DO 5100 J=1,4
240      5100 WRITE(6,130) J,(TSEAT(J,K),K=1,N)
241      WRITE(6,131) (TH20(K),K=1,N)
242      DO 5200 J=1,3
243      5200 WRITE(6,132) A(J),(TGD(J,K),K=1,N)
244      IF(ISTOP.EQ.1) STOP
245      GO TO 10
246      101 FORMAT(1H1,15HTEST NO.          ,10(F5.0,2X))
247      103 FORMAT(1X,15HHRPM                ,10(F6.0,1X))
248      104 FORMAT(1X,15HNOX, PPM            ,96A1)
249      105 FORMAT(1X,15HNO, PPM             ,10(96A1))
250      106 FORMAT(1X,15HO2, MIX, % D       ,10(F5.1,2X))
251      110 FORMAT(1X,15HBMEP, PSI          ,10(F5.1,2X))
252      111 FORMAT(1X,15HVCF, %            ,10(F5.0,2X))
253      112 FORMAT(1X,15HVOL % H2          ,10(F5.1,2X))
254      113 FORMAT(1X,15HBTE, %            ,10(F5.1,2X))
255      114 FORMAT(1X,15HNOX, G/HP-HR       ,10(96A1))
256      115 FORMAT(1X,15HO2, MIX, % W       ,10(F5.1,2X))
257      123 FORMAT(1X,15HPhi                ,10(F5.2,2X))
258      124 FORMAT(1X,15HUPSILON            ,10(F5.2,2X))
259      126 FORMAT(1X,15HH2O/H2 RATIO       ,10(F5.1,2X))
260      127 FORMAT(1X,15HEGR, %            ,10(F5.1,2X))
261      128 FORMAT(1X,15HSA, DEG            ,10(F5.0,2X))
262      129 FORMAT(1X,15HMISFIRE, %        ,10(F5.0,2X))
263      1128 FORMAT(1X,5HT,EX ,I1,9H,R, F   ,10(F6.0,1X))
264      1129 FORMAT(1X,5HT,EX ,I1,9H,C, F   ,10(F6.0,1X))
265      130 FORMAT(1X,7HT,SEAT ,I1,7H, F    ,10(F6.0,1X))
266      131 FORMAT(1X,15HT,H2O, F           ,10(F6.0,1X))
267      132 FORMAT(1X,5HT,GD.,A1,9H, F     ,10(F6.0,1X))
268      133 FORMAT(1X,15HT,AIR, F          ,10(F5.0,2X))
269      134 FORMAT(1X,15HATM PRESS, PSI     ,10(F6.2,1X))
270      135 FORMAT(1X,15HVAP PRESS, PSI    ,10(F5.3,2X))
271      136 FORMAT(1X,7HO2, CYL,I1,7H, % D ,96A1)
272      138 FORMAT(1X,15HENG,EXP           ,96A1)
273      END

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QXQT TPF\$.FINAL/A

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A7.3 DEFINITIONS OF SYMBOLS USED
 IN EXPERIMENTAL RESULTS TABLE

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A7.3 DEFINITIONS OF SYMBOLS FOR EXPERIMENTAL RESULTS TABLES

<u>Fortran Symbol</u>	<u>Description</u>	<u>Units</u>
TEST NO.	Number of Particular Test	---
ENG, EXP	Table 1 Engine Configuration No., Table 7 Experiment No.	---
T, AIR, F	Air Temperature	$^{\circ}\text{F}$
ATM PRESS, PSI	Atmospheric Pressure	psi
VAP PRESS, PSI	Vapor Pressure of Humidity in the Air	psi
MISFIRE, %	Percentage of Cycles having SSLII - Induced Misfire	%
H2O/H2 RATIO	Mass Ratio of Water Injected to Hydrogen	---
EGR, %	Percentage of Mixture at Carburetor which is EGR Gas	%
RPM	Revolution per minute of the Crankshaft	rpm
BMEP, PSI	Brake Mean Effective Pressure	psi
SA, DEG	Spark Advance before Top Dead Center	
VCF, %	Volume Cylinder Filling, the percentage of the Cylinder that would be filled at 70°F and 14.7 psi with the Mixture including EGR Gas. (See Nomenclature - Section 16)	%
VOL % H2	Volume Percent Hydrogen, the percentage of Hydrogen in the Mixture not including Humidity or EGR Gas	%
PHI	Fuel-Air Equivalence Ratio as Fraction of Dry Mixture not including Exhaust Gas	---
UPSILON	Hydrogen/Oxygen Equivalence Ratio as Fraction of Dry Mixture including Oxygen in Recycled Exhaust Gas	---
BTE, %	Brake Thermal Efficiency	%
NOX, PPM	Total Nitric Oxide Emissions of Wet Exhaust Gas	ppm
NO, PPM	NO Emissions of Wet Exhaust Gas	ppm
NOX, G/HP-HR	Total Nitric Oxide Emissions	g/hp hr
O2, MIX, % D	Percent Oxygen in Dry Exhaust Gas	%
O2, MIX, % W	Percent Oxygen in Wet Exhaust Gas	%
O2, CYLi, % D	Percent Oxygen in Dry Exhaust From Cylinder Number i, i = 1-4	%

A7.3 (Cont.) DEFINITIONS OF SYMBOLS FOR EXPERIMENTAL RESULTS TABLES

<u>Fortran Symbol</u>	<u>Description</u>	<u>Units</u>
T, EX i, R, F	Temperature Measured by Thermocouple in Cylinder Number i Exhaust Port	°F
T, EX i, C, F	Calculated Temperature of Exhaust Gas in Cylinder i Exhaust Port taking into account Radiation Heat Losses from the Thermocouple (See Appendix 8)	°F
T, SEAT i, F	Temperature of Figure 3 Thermocouple T, Seat j for Cylinder i	°F
T, H2O, F	Temperature of Coolant Water as it leaves Cylinder Head	°F
T, GD, i, F	Temperature of Figure 3 Thermocouple T, Gd i. Cylinder Number 4 Exhaust Valve Guide. i = 1-3	°F

A7.4 EXPERIMENTAL RESULTS TABLE
 (See Section 10 for Description)

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A7.4

TEST NO.	1.	2.	3.	4.	5.
ENG,EXP	1,1	1,2	1,2	1,2	1,2
T,AIR, F	77.	77.	77.	77.	77.
ATM PRESS, PSI	14.69	14.69	14.69	14.69	14.69
VAP PRESS, PSI	.381	.381	.381	.381	.381
MISFIRE, %	0.	1.	2.	5.	10.
RPM	820.	815.	810.	800.	760.
BMEP, PSI	.0	.0	.0	.0	.0
SA, DEG	0.	0.	0.	0.	0.
VCF, %	19.	19.	19.	19.	19.
VOL % H2	30.6	30.6	30.9	31.2	32.1
PHI	1.05	1.05	1.06	1.08	1.13
UPSILON	1.05	1.05	1.06	1.08	1.13
BTE, %	.0	.0	.0	.0	.0
NOX, PPM	35.0	35.0	43.0	65.0	105.
NO, PPM	27.0	32.0	40.0	60.0	97.0
NOX, G/HP-HR	UNDEF.	UNDEF.	UNDEF.	UNDEF.	UNDEF.
O2, MIX, % D	3.0	3.0	3.1	3.4	4.0
O2, MIX, % W	2.1	2.1	2.2	2.4	2.9
O2, CYL1, % D	2.2	2.6	2.3	1.8	1.7
O2, CYL2, % D	2.4	2.4	2.1	3.6	3.6
O2, CYL3, % D	4.0	3.9	3.6	3.9	5.0
O2, CYL4, % D	3.4	3.4	3.1	3.1	3.0
T,EX 1,R, F	532.	530.	528.	535.	542.
T,EX 2,R, F	543.	547.	546.	501.	496.
T,EX 3,R, F	535.	538.	537.	528.	486.
T,EX 4,R, F	522.	525.	514.	528.	526.
T,EX 1,C, F	594.	592.	588.	591.	591.
T,EX 2,C, F	605.	609.	606.	557.	545.
T,EX 3,C, F	597.	600.	597.	584.	535.
T,EX 4,C, F	584.	587.	574.	584.	575.
T,SEAT 1, F	203.	206.	203.	206.	207.
T,SEAT 2, F	210.	213.	211.	210.	211.
T,SEAT 3, F	210.	212.	210.	210.	207.
T,SEAT 4, F	203.	204.	203.	202.	202.
T,H2O, F	180.	174.	181.	180.	179.
T,GD,A, F	193.	189.	193.	194.	191.
T,GD,B, F	190.	186.	190.	189.	189.
T,GD,C, F	189.	186.	189.	187.	188.

A7.4 Cont.

TEST NO.	6.	7.	8.	9.	10.
ENG,EXP	4,1	4,2	4,2	4,2	4,2
T,AIR, F	77.	77.	77.	77.	77.
ATM PRESS, PSI	14.69	14.69	14.69	14.69	14.69
VAP PRESS, PSI	.381	.381	.381	.381	.381
MISFIRE, %	0.	1.	2.	5.	10.
RPM	820.	820.	820.	820.	810.
BMEP, PSI	.0	.0	.0	.0	.0
SA, DEG	94.	94.	94.	94.	94.
VCF, %	85.	85.	85.	85.	85.
VOL % H2	7.2	7.2	7.1	7.1	7.2
PHI	.18	.18	.18	.18	.19
UPSILON	.18	.18	.18	.18	.19
BTE, %	.0	.0	.0	.0	.0
NOX, PPM	.40	.40	.40	.40	.40
NO, PPM	.10	.10	.10	.20	.20
NOX, G/HP-HR	UNDEF.	UNDEF.	UNDEF.	UNDEF.	UNDEF.
O2, MIX, % D	18.6	18.7	18.8	18.8	18.8
O2, MIX, % W	17.4	17.6	17.7	17.7	17.7
O2, CYL1, % D	18.8	18.8	18.8	19.0	18.5
O2, CYL2, % D	18.7	18.8	18.8	18.8	18.8
O2, CYL3, % D	18.5	18.6	18.6	18.6	18.8
O2, CYL4, % D	18.6	18.6	18.7	18.6	18.5
T,EX 1,R, F	311.	315.	309.	284.	302.
T,EX 2,R, F	317.	320.	322.	317.	280.
T,EX 3,R, F	324.	325.	335.	332.	289.
T,EX 4,R, F	329.	335.	336.	337.	329.
T,EX 1,C, F	319.	323.	317.	292.	309.
T,EX 2,C, F	325.	328.	330.	325.	287.
T,EX 3,C, F	332.	333.	343.	340.	296.
T,EX 4,C, F	337.	343.	344.	345.	336.
T,SEAT 1, F	185.	190.	193.	189.	183.
T,SEAT 2, F	190.	195.	199.	196.	187.
T,SEAT 3, F	191.	195.	200.	198.	189.
T,SEAT 4, F	187.	192.	196.	194.	186.
T,H2O, F	173.	178.	182.	179.	172.
T,GD,A, F	182.	187.	191.	189.	182.
T,GD,B, F	180.	185.	189.	187.	179.
T,GD,C, F	179.	184.	189.	186.	179.

A7.4 Cont.

TEST NO.	11.	12.	13.	14.	15.
ENG,EXP	1.6	1.6	1.6	1.6	1.6
T,AIR, F	74.	74.	74.	74.	74.
ATM PRESS, PSI	14.74	14.74	14.74	14.74	14.74
VAP PRESS, PSI	.398	.398	.398	.398	.398
MISFIRE, %	0.	1.	2.	5.	10.
RPM	1800.	1800.	1780.	1750.	1660.
BMEP, PSI	49.8	49.8	49.8	49.2	47.9
SA, DEG	0.	0.	0.	0.	0.
VCF, %	50.	50.	51.	51.	53.
VOL % H2	30.4	30.4	30.6	30.9	31.4
PHI	1.04	1.04	1.05	1.06	1.09
UPSILON	1.04	1.04	1.05	1.06	1.09
BTE, %	22.9	22.9	22.6	22.0	20.3
NOX, PPM	4600.	4300.	3700.	3300.	3000.
NO, PPM	4600.	4100.	3500.	3000.	2500.
NOX, G/HP-HR	19.4	18.5	16.1	15.0	15.1
O2, MIX, % D	1.0	1.0	1.0	1.3	2.0
O2, MIX, % W	.7	.7	.7	.9	1.4
O2, CYL1, % D	.4	.3	.3	.2	2.9
O2, CYL2, % D	.8	.6	1.0	2.8	.7
O2, CYL3, % D	1.8	1.8	2.2	1.4	1.3
O2, CYL4, % D	.8	1.4	.8	.6	3.1
T,EX 1,R, F	1116.	1128.	1107.	1096.	945.
T,EX 2,R, F	1137.	1139.	1110.	990.	1107.
T,EX 3,R, F	1115.	1118.	1088.	1125.	1107.
T,EX 4,R, F	1144.	1117.	1142.	1139.	981.
T,EX 1,C, F	1258.	1270.	1243.	1221.	1049.
T,EX 2,C, F	1279.	1281.	1246.	1115.	1211.
T,EX 3,C, F	1257.	1260.	1224.	1250.	1211.
T,EX 4,C, F	1286.	1259.	1278.	1264.	1085.
T,SEAT 1, F	252.	255.	255.	255.	244.
T,SEAT 2, F	273.	277.	274.	264.	275.
T,SEAT 3, F	277.	280.	279.	279.	282.
T,SEAT 4, F	275.	276.	278.	279.	264.
T,H2O, F	198.	198.	199.	198.	197.
T,GD,A, F	236.	234.	237.	236.	228.
T,GD,B, F	222.	222.	223.	224.	217.
T,GD,C, F	218.	219.	219.	221.	215.

A7.4 Cont.

TEST NO.	16.	17.	18.	19.	20.
ENG,EXP	4.6	4.6	4.6	4.6	4.6
T,AIR, F	74.	74.	74.	74.	74.
ATM PRESS, PSI	14.74	14.74	14.74	14.74	14.74
VAP PRESS, PSI	.398	.398	.398	.398	.398
MISFIRE, %	0.	1.	2.	5.	10.
RPM	1800.	1780.	1765.	1740.	1710.
BMEP, PSI	46.7	46.1	45.8	45.4	44.2
SA, DEG	34.	34.	34.	34.	34.
VCF, %	80.	81.	81.	81.	79.
VOL % H2	14.7	14.8	14.9	15.2	15.7
PHI	.41	.41	.42	.43	.44
UPSILON	.41	.41	.42	.43	.44
BTE, %	27.8	27.1	26.7	26.1	25.0
NOX, PPM	4.50	4.50	4.60	5.10	5.80
NO, PPM	1.60	1.70	1.80	2.00	2.10
NOX, G/HP-HR	.05	.05	.05	.05	.06
O2, MIX, % D	13.9	14.0	14.0	14.0	14.0
O2, MIX, % W	11.6	11.7	11.7	11.7	11.8
O2, CYL1, % D	13.8	13.9	14.0	14.5	14.1
O2, CYL2, % D	13.8	13.9	13.9	13.8	13.5
O2, CYL3, % D	13.9	13.9	13.9	13.8	13.7
O2, CYL4, % D	14.0	13.9	14.0	13.8	14.1
T,EX 1,R, F	718.	725.	692.	615.	617.
T,EX 2,R, F	728.	728.	728.	728.	731.
T,EX 3,R, F	706.	686.	707.	710.	714.
T,EX 4,R, F	717.	717.	697.	723.	626.
T,EX 1,C, F	757.	763.	729.	650.	648.
T,EX 2,C, F	767.	766.	765.	763.	762.
T,EX 3,C, F	745.	724.	744.	745.	745.
T,EX 4,C, F	756.	755.	734.	758.	657.
T,SEAT 1, F	225.	226.	225.	219.	220.
T,SEAT 2, F	235.	235.	236.	237.	238.
T,SEAT 3, F	238.	237.	238.	239.	240.
T,SEAT 4, F	227.	227.	226.	229.	223.
T,H2O, F	182.	182.	182.	182.	182.
T,GD,A, F	207.	208.	206.	208.	204.
T,GD,B, F	199.	199.	199.	200.	197.
T,GD,C, F	198.	197.	198.	198.	196.

A7.4 Cont.

TEST NO.	21.	22.	23.	24.	25.	26.	27.
ENG,EXP	1,3	1,3	1,3	1,3	1,3	1,3	1,3
T,AIR, F	74.	74.	74.	74.	74.	74.	74.
ATM PRESS, PSI	14.74	14.74	14.74	14.74	14.74	14.74	14.74
VAP PRESS, PSI	.398	.398	.398	.398	.398	.398	.398
MISFIRE, %	0.	0.	0.	0.	0.	0.	0.
RPM	1800.	1800.	1800.	1800.	1800.	1800.	1800.
BMEP, PSI	.0	13.5	17.7	30.3	35.8	49.8	60.4
SA, DEG	0.	0.	0.	0.	0.	0.	0.
VCF, %	19.	28.	30.	39.	41.	50.	56.
VOL % H2	31.5	30.9	30.9	30.5	30.4	30.4	30.5
PHI	1.09	1.06	1.06	1.04	1.04	1.04	1.05
UPSILON	1.09	1.06	1.06	1.04	1.04	1.04	1.05
BTE, %	.0	11.2	13.4	18.1	20.0	22.9	24.7
NOX, PPM	212.	1375.	1750.	3000.	3400.	4600.	5000.
NO, PPM	200.	1375.	1750.	3000.	3400.	4600.	5000.
NOX, G/HP-HR	UNDEF.	11.7	12.4	16.0	16.4	19.4	19.4
O2, MIX, % D	2.0	1.6	1.4	1.3	1.2	1.0	1.0
O2, MIX, % W	1.4	1.1	.9	.9	.8	.7	.7
O2, CYL1, % D	.2	.1	.1	.2	.2	.4	.9
O2, CYL2, % D	.3	.1	.1	.2	.2	.8	1.2
O2, CYL3, % D	4.0	3.4	3.4	2.9	2.8	1.8	1.1
O2, CYL4, % D	2.4	2.0	2.0	1.9	1.6	.8	.8
T,EX 1,R, F	876.	981.	1000.	1079.	1089.	1116.	1128.
T,EX 2,R, F	842.	975.	990.	1085.	1107.	1137.	1145.
T,EX 3,R, F	846.	975.	991.	1060.	1075.	1115.	1152.
T,EX 4,R, F	914.	999.	1020.	1086.	1101.	1144.	1159.
T,EX 1,C, F	995.	1119.	1138.	1225.	1236.	1258.	1264.
T,EX 2,C, F	961.	1113.	1128.	1231.	1254.	1279.	1281.
T,EX 3,C, F	965.	1113.	1129.	1206.	1222.	1257.	1288.
T,EX 4,C, F	1033.	1137.	1158.	1232.	1248.	1286.	1295.
T,SEAT 1, F	222.	232.	235.	239.	241.	252.	265.
T,SEAT 2, F	233.	245.	251.	258.	259.	273.	290.
T,SEAT 3, F	232.	244.	250.	255.	258.	277.	296.
T,SEAT 4, F	225.	236.	242.	252.	258.	275.	291.
T,H2O, F	184.	182.	183.	186.	189.	198.	204.
T,GD,A, F	202.	207.	213.	217.	221.	236.	246.
T,GD,B, F	197.	199.	204.	207.	211.	221.	232.
T,GD,C, F	196.	198.	199.	204.	207.	218.	227.

A7.4 Cont.

TEST NO.	28.	29.	30.	31.	32.	33.	34.	35.	36.
ENG,EXP	4.3	4.3	4.3	4.3	4.3	4.3	4.3	4.3	4.3
T,AIR, F	76.	76.	76.	76.	76.	74.	76.	76.	76.
ATM PRESS, PSI	14.69	14.69	14.69	14.69	14.69	14.74	14.69	14.69	14.69
VAP PRESS, PSI	.355	.355	.355	.355	.355	.398	.355	.355	.355
MISFIRE, %	0.	0.	0.	0.	0.	0.	0.	0.	0.
RPM	1800.	1800.	1800.	1800.	1800.	1800.	1800.	1800.	1800.
BMEP, PSI	.0	14.3	20.9	32.1	41.1	46.7	54.2	64.4	69.1
SA, DEG	94.	94.	82.	58.	45.	34.	28.	20.	16.
VCF, %	80.	81.	79.	79.	80.	80.	80.	79.	79.
VOL % H2	7.4	9.2	9.7	11.8	13.5	14.7	16.7	19.5	20.7
PHI	.19	.24	.26	.32	.37	.41	.48	.58	.62
UPSILON	.19	.24	.26	.32	.37	.41	.48	.58	.62
BTE, %	.0	13.5	19.3	24.1	26.6	27.8	28.5	29.2	29.6
NOX, PPM	1.10	1.60	1.70	2.40	3.20	4.50	11.0	165.	450.
NO, PPM	.00	.25	.50	.80	1.10	1.60	4.20	160.	450.
NOX, G/HP-HR	UNDEF.	.06	.04	.04	.04	.05	.10	.91	2.25
O2, MIX, % D	18.6	17.3	16.8	15.6	14.6	13.9	12.5	10.6	9.6
O2, MIX, % W	17.5	15.6	14.9	13.5	12.3	11.6	10.1	8.3	7.3
O2, CYL1, % D	18.4	17.2	16.7	15.5	14.5	13.8	12.4	10.4	9.6
O2, CYL2, % D	18.4	17.2	16.7	15.5	14.5	13.8	12.4	10.4	9.4
O2, CYL3, % D	18.8	17.6	16.9	15.8	14.8	13.9	12.8	10.8	10.2
O2, CYL4, % D	18.6	17.3	16.8	15.6	14.7	14.0	12.7	10.6	10.0
T,EX 1,R, F	342.	471.	513.	604.	667.	718.	774.	868.	892.
T,EX 2,R, F	338.	467.	522.	615.	672.	728.	793.	883.	920.
T,EX 3,R, F	302.	447.	491.	586.	652.	706.	775.	863.	899.
T,EX 4,R, F	353.	470.	506.	602.	670.	717.	776.	865.	901.
T,EX 1,C, F	348.	484.	531.	631.	700.	757.	823.	930.	961.
T,EX 2,C, F	344.	480.	540.	642.	705.	767.	842.	945.	989.
T,EX 3,C, F	308.	460.	509.	613.	685.	745.	824.	925.	968.
T,EX 4,C, F	359.	483.	524.	629.	703.	756.	825.	927.	970.
T,SEAT 1, F	197.	209.	211.	218.	223.	225.	230.	232.	234.
T,SEAT 2, F	198.	212.	220.	225.	233.	235.	244.	249.	253.
T,SEAT 3, F	195.	210.	219.	226.	235.	238.	246.	253.	256.
T,SEAT 4, F	172.	207.	213.	217.	224.	227.	234.	244.	247.
T,H2O, F	179.	181.	176.	183.	182.	182.	185.	186.	185.
T,GD,A, F	187.	193.	195.	201.	206.	207.	212.	218.	219.
T,GD,B, F	184.	188.	191.	195.	199.	199.	203.	207.	206.
T,GD,C, F	183.	186.	189.	193.	198.	198.	201.	204.	204.

A7.4 Cont.

TEST NO.	37.	38.	39.	40.
ENG,EXP	1.9	4.9	1.12	4.12
T,AIR, F	76.	76.	76.	76.
ATM PRESS, PSI	14.69	14.69	14.69	14.69
VAP PRESS, PSI	.355	.355	.355	.355
MISFIRE, %	0.	0.	0.	0.
RPM	3300.	3300.	3300.	3300.
BMEP, PSI	25.5	25.5	45.4	47.5
SA, DEG	0.	54.	0.	40.
VCF, %	37.	83.	48.	81.
VOL % H2	30.7	11.0	31.7	15.4
PHI	1.06	.29	1.10	.43
UPSILON	1.06	.29	1.10	.43
BTE, %	15.6	19.8	21.1	26.9
NOX, PPM	3150.	1.50	3500.	9.50
NO, PPM	3150.	.50	3500.	4.50
NOX, G/HP-HR	19.2	.03	15.3	.09
O2, MIX, % D	.6	16.0	.3	13.4
O2, MIX, % W	.4	13.9	.2	11.0
O2, CYL1, % D	.4	15.8	.3	13.2
O2, CYL2, % D	1.0	15.9	.4	13.2
O2, CYL3, % D	.4	16.2	.3	13.6
O2, CYL4, % D	.3	15.9	.3	13.2
T,EX 1,R, F	1281.	636.	1319.	818.
T,EX 2,R, F	1312.	642.	1352.	820.
T,EX 3,R, F	1304.	623.	1352.	797.
T,EX 4,R, F	1317.	638.	1355.	824.
T,EX 1,C, F	1456.	657.	1471.	856.
T,EX 2,C, F	1487.	663.	1504.	858.
T,EX 3,C, F	1479.	644.	1504.	835.
T,EX 4,C, F	1492.	659.	1507.	862.
T,SEAT 1, F	251.	227.	281.	241.
T,SEAT 2, F	271.	232.	308.	256.
T,SEAT 3, F	280.	236.	314.	257.
T,SEAT 4, F	279.	229.	313.	250.
T,H2O, F	191.	181.	210.	185.
T,GD,A, F	230.	203.	252.	212.
T,GD,B, F	214.	194.	234.	200.
T,GD,C, F	209.	190.	228.	196.

A7.4 Cont..

TEST NO.	41.	42.	43.
ENG,EXP	3.5	3.5	3.5
T,AIR, F	80.	80.	80.
ATM PRESS, PSI	14.77	14.77	14.77
VAP PRESS, PSI	.406	.406	.406
H2O/H2 RATIO	5.1	5.1	5.1
RPM	1800.	1800.	1800.
BMEP, PSI	61.9	71.7	78.1
SA, DEG	10.	10.	10.
VCF, %	57.	63.	67.
VOL % H2	31.0	31.4	31.3
PHI	1.07	1.09	1.08
UPSILON	1.07	1.09	1.08
BTE, %	24.5	25.4	26.3
NOX, PPM	870.	1050.	1250.
NO, PPM	870.	1050.	1250.
NOX, G/HP-HR	4.02	4.63	5.33
O2, MIX, % D	.5	.4	.4
O2, MIX, % W	.3	.2	.2
O2, CYL1, % D	.5	.4	.4
O2, CYL2, % D	.2	.4	.4
O2, CYL3, % D	1.0	.5	.8
O2, CYL4, % D	.7	.4	.4
T,EX 1,R, F	1040.	1049.	1045.
T,EX 2,R, F	1101.	1057.	1056.
T,EX 3,R, F	1053.	1079.	1083.
T,EX 4,R, F	1059.	1072.	1084.
T,EX 1,C, F	1140.	1141.	1135.
T,EX 2,C, F	1201.	1149.	1146.
T,EX 3,C, F	1153.	1171.	1173.
T,EX 4,C, F	1159.	1164.	1174.
T,SEAT 1, F	247.	266.	277.
T,SEAT 2, F	258.	286.	294.
T,SEAT 3, F	273.	289.	293.
T,SEAT 4, F	267.	286.	292.
T,H2O, F	184.	186.	186.
T,GD,A, F	236.	248.	251.
T,GD,B, F	220.	229.	232.
T,GD,C, F	214.	223.	225.

A7.4 Cont.

TEST NO.	44.	45.	46.	47.
ENG,EXP	6.5	6.5	6.5	6.5
T,AIP, F	80.	80.	80.	80.
ATM PRESS, PSI	14.77	14.77	14.77	14.77
VAP PRESS, PSI	.406	.406	.406	.406
H2O/H2 RATIO	5.0	5.1	5.1	5.1
RPM	1800.	1800.	1800.	1800.
BMEP, PSI	54.9	70.5	81.4	83.7
SA, DEG	34.	22.	14.	10.
VCF, %	76.	72.	71.	71.
VOL % H2	17.9	23.4	28.9	29.8
PHI	.52	.73	.97	1.01
UPSILON	.52	.73	.97	1.01
BTE, %	28.4	29.3	28.0	27.8
NOX, PPM	10.0	250.	1400.	1800.
NO, PPM	4.00	250.	1400.	1800.
NOX, G/HP-HR	.09	1.27	6.03	7.58
O2, MIX, % D	11.9	7.8	2.1	1.1
O2, MIX, % W	8.6	5.0	1.2	.6
O2, CYL1, % D	11.8	7.6	1.6	.6
O2, CYL2, % D	11.7	7.4	2.0	.6
O2, CYL3, % D	12.0	8.4	3.2	2.0
O2, CYL4, % D	11.8	7.7	2.1	1.0
T,EX 1,R, F	762.	888.	1027.	1075.
T,EX 2,R, F	778.	919.	1040.	1097.
T,EX 3,R, F	767.	889.	1034.	1074.
T,EX 4,R, F	773.	907.	1037.	1102.
T,EX 1,C, F	806.	951.	1114.	1171.
T,EX 2,C, F	822.	982.	1127.	1193.
T,EX 3,C, F	811.	952.	1121.	1170.
T,EX 4,C, F	817.	970.	1124.	1198.
T,SEAT 1, F	238.	253.	261.	274.
T,SEAT 2, F	253.	274.	289.	297.
T,SEAT 3, F	255.	273.	290.	291.
T,SEAT 4, F	241.	263.	285.	294.
T,H2O, F	182.	182.	186.	187.
T,GD,A, F	219.	232.	247.	253.
T,GD,B, F	208.	217.	229.	234.
T,GD,C, F	205.	212.	223.	227.

A7.4 Cont.

TEST NO.	48.	49.	50.	51.	52.	53.
ENG,EXP	3.8	3.8	3.8	3.8	3.8	3.8
T,AIR, F	81.	81.	81.	81.	81.	81.
ATM PRESS, PSI	14.76	14.76	14.76	14.76	14.76	14.76
VAP PRESS, PSI	.419	.419	.419	.419	.419	.419
H2O/H2 RATIO	.0	4.2	6.3	8.8	9.7	11.7
RPM	1800.	1800.	1800.	1800.	1800.	1800.
BMEP, PSI	51.0	50.9	51.0	50.9	50.4	51.0
SA, DEG	0.	6.	10.	10.	10.	10.
VCF, %	51.	49.	49.	50.	50.	51.
VOL % H2	31.2	31.1	31.4	31.4	31.3	31.3
PHI	1.08	1.07	1.09	1.09	1.09	1.09
UPSILON	1.08	1.07	1.09	1.09	1.09	1.09
BTE, %	22.5	23.4	23.1	22.8	22.5	22.6
NOX, PPM	3500.	625.	300.	70.0	55.0	30.0
NO, PPM	3500.	625.	300.	70.0	55.0	30.0
NOX, G/HP-HR	14.6	2.93	1.51	.39	.32	.18
O2, MIX, % D	.5	.7	.6	.7	.7	.5
O2, MIX, % W	.3	.4	.3	.3	.3	.2
O2, CYL1, % D	.1	.4	.4	.4	.4	.2
O2, CYL2, % D	.2	.4	.4	.5	.5	.2
O2, CYL3, % D	1.0	1.6	1.4	1.6	1.6	1.0
O2, CYL4, % D	.4	.8	.6	.8	.8	.4
T,EX 1,R, F	1103.	1029.	990.	981.	975.	943.
T,EX 2,R, F	1103.	1050.	1012.	990.	990.	961.
T,EX 3,R, F	1111.	1051.	1024.	997.	985.	974.
T,EX 4,R, F	1140.	1069.	1040.	1000.	1002.	982.
T,EX 1,C, F	1236.	1137.	1085.	1066.	1058.	1019.
T,EX 2,C, F	1236.	1158.	1107.	1075.	1073.	1037.
T,EX 3,C, F	1244.	1159.	1119.	1082.	1068.	1050.
T,EX 4,C, F	1273.	1177.	1135.	1085.	1085.	1058.
T,SEAT 1, F	282.	251.	249.	238.	235.	231.
T,SEAT 2, F	307.	274.	269.	248.	243.	237.
T,SEAT 3, F	317.	275.	271.	250.	245.	236.
T,SEAT 4, F	311.	275.	263.	239.	233.	228.
T,H2O, F	207.	186.	182.	179.	178.	184.
T,GD,A, F	260.	237.	227.	223.	220.	216.
T,GD,B, F	244.	223.	213.	210.	208.	206.
T,GD,C, F	239.	218.	210.	206.	205.	202.

A7.4 Cont.

TEST NO.	54.	55.	56.	57.	58.	59.
ENG,EXP	3.8	3.8	3.8	3.8	3.8	3.8
T,AIR, F	84.	84.	84.	84.	84.	84.
ATM PRESS, PSI	14.78	14.78	14.78	14.78	14.78	14.78
VAP PRESS, PSI	.410	.410	.410	.410	.410	.410
H2O/H2 RATIO	.0	3.9	5.9	7.8	8.8	10.5
RPM	1800.	1800.	1800.	1800.	1800.	1800.
BMEP, PSI	52.4	52.6	52.4	52.4	52.4	52.4
SA, DEG	0.	0.	0.	0.	0.	0.
VCF, %	52.	51.	53.	55.	55.	56.
VOL % H2	31.0	31.3	31.1	31.0	31.3	31.4
PHI	1.07	1.08	1.07	1.07	1.08	1.09
UPSILON	1.07	1.08	1.07	1.07	1.08	1.09
BTE, %	22.9	23.0	22.4	21.7	21.4	21.1
NOX, PPM	3700.	475.	150.	60.0	40.0	25.0
NO, PPM	3700.	475.	150.	60.0	40.0	25.0
NOX, G/HP-HR	15.2	2.23	.78	.34	.24	.16
O2, MIX, % D	.4	.4	.6	.8	.4	.4
O2, MIX, % W	.3	.2	.3	.4	.2	.2
O2, CYL1, % D	.3	.4	.6	.7	.3	.4
O2, CYL2, % D	.5	.5	.6	1.1	.3	.6
O2, CYL3, % D	.7	.7	.7	1.0	.5	.5
O2, CYL4, % D	.3	.3	.5	.6	.3	.3
T,EX 1,R, F	1123.	1099.	1089.	1077.	1073.	1043.
T,EX 2,R, F	1132.	1115.	1111.	1100.	1091.	1062.
T,EX 3,R, F	1130.	1114.	1101.	1089.	1087.	1064.
T,EX 4,R, F	1136.	1114.	1108.	1105.	1093.	1052.
T,EX 1,C, F	1262.	1219.	1201.	1179.	1172.	1132.
T,EX 2,C, F	1271.	1235.	1223.	1202.	1190.	1151.
T,EX 3,C, F	1269.	1234.	1213.	1191.	1186.	1153.
T,EX 4,C, F	1277.	1234.	1220.	1207.	1192.	1141.
T,SEAT 1, F	260.	238.	237.	235.	234.	228.
T,SEAT 2, F	281.	257.	255.	252.	250.	240.
T,SEAT 3, F	289.	263.	260.	258.	254.	244.
T,SEAT 4, F	254.	236.	228.	218.	238.	227.
T,H2O, F	201.	188.	183.	181.	181.	179.
T,GD,A, F	240.	226.	223.	222.	220.	213.
T,GD,B, F	226.	211.	209.	208.	207.	202.
T,GD,C, F	221.	208.	206.	204.	204.	200.

A7.4 Cont.

TEST NO.	60.	61.	62.	63.	64.
ENG,EXP	6.8	6.8	6.8	6.8	6.8
T,AIR, F	84.	84.	84.	84.	84.
ATM PRESS, PSI	14.78	14.78	14.78	14.78	14.78
VAP PRESS, PSI	.410	.410	.410	.410	.410
H2O/H2 RATIO	.0	4.6	7.0	9.8	11.8
RPM	1800.	1800.	1800.	1800.	1800.
BMEP, PSI	51.4	51.4	51.4	51.4	51.4
SA, DEG	30.	30.	30.	30.	30.
VCF, %	78.	76.	76.	75.	74.
VOL % H2	16.3	17.3	17.8	18.1	18.2
PHI	.46	.50	.52	.53	.53
UPSILON	.46	.50	.52	.53	.53
BTE, %	28.5	27.6	26.9	26.8	26.8
NOX, PPM	5.00	3.00	2.50	2.50	2.50
NO, PPM	2.00	1.00	.90	.90	.90
NOX, G/HP-HR	.04	.03	.03	.03	.03
O2, MIX, % D	12.8	12.1	11.9	11.8	11.7
O2, MIX, % W	10.4	8.8	8.3	7.8	7.4
O2, CYL1, % D	12.8	12.1	11.9	11.8	11.7
O2, CYL2, % D	12.7	11.9	11.7	11.6	11.5
O2, CYL3, % D	12.9	12.2	12.2	12.0	12.0
O2, CYL4, % D	12.8	12.0	12.0	11.8	11.8
T,EX 1,R, F	759.	777.	776.	740.	726.
T,EX 2,R, F	776.	785.	779.	775.	761.
T,EX 3,R, F	749.	760.	756.	737.	720.
T,EX 4,R, F	762.	787.	777.	760.	740.
T,EX 1,C, F	806.	824.	820.	780.	763.
T,EX 2,C, F	823.	832.	823.	815.	798.
T,EX 3,C, F	796.	807.	800.	777.	757.
T,EX 4,C, F	809.	834.	821.	800.	777.
T,SEAT 1, F	225.	227.	223.	216.	215.
T,SEAT 2, F	239.	235.	229.	226.	223.
T,SEAT 3, F	240.	235.	229.	223.	218.
T,SEAT 4, F	230.	225.	216.	213.	209.
T,H2O, F	181.	180.	177.	179.	182.
T,GD,A, F	207.	206.	203.	201.	195.
T,GD,B, F	198.	198.	195.	196.	190.
T,GD,C, F	195.	196.	194.	194.	188.

A7.4 Cont.

TEST NO.	65.	66.	67.	68.
ENG,EXP	6.8	6.8	6.8	6.8
T,AIR, F	84.	84.	84.	84.
ATM PRESS, PSI	14.78	14.78	14.78	14.78
VAP PRESS, PSI	.410	.410	.410	.410
H2O/H2 RATIO	.0	5.3	8.1	10.5
RPM	1800.	1800.	1800.	1800.
BMEP, PSI	51.4	51.4	51.4	51.4
SA, DEG	30.	36.	37.	37.
VCF, %	78.	76.	75.	75.
VOL % H2	16.3	17.1	17.7	17.9
PHI	.46	.49	.51	.52
UPSILON	.46	.49	.51	.52
BTE, %	28.5	27.8	27.2	27.0
NOX, PPM	5.00	3.00	2.70	2.60
NO, PPM	2.00	1.10	1.00	1.00
NOX, G/HP-HR	.04	.03	.03	.03
O2, MIX, % D	12.8	12.2	12.0	11.9
O2, MIX, % W	10.4	8.8	8.2	7.8
O2, CYL1, % D	12.8	12.2	12.0	11.9
O2, CYL2, % D	12.7	12.1	12.0	11.7
O2, CYL3, % D	12.9	12.4	11.8	12.2
O2, CYL4, % D	12.8	12.2	12.1	11.9
T,EX 1,R, F	759.	755.	740.	737.
T,EX 2,R, F	776.	763.	752.	742.
T,EX 3,R, F	749.	740.	724.	714.
T,EX 4,R, F	762.	765.	751.	752.
T,EX 1,C, F	806.	799.	780.	775.
T,EX 2,C, F	823.	807.	792.	780.
T,EX 3,C, F	796.	784.	764.	752.
T,EX 4,C, F	809.	809.	791.	790.
T,SEAT 1, F	225.	223.	221.	221.
T,SEAT 2, F	239.	235.	236.	232.
T,SEAT 3, F	240.	239.	237.	235.
T,SEAT 4, F	230.	230.	227.	226.
T,H2O, F	181.	182.	182.	181.
T,GD,A, F	207.	207.	205.	203.
T,GD,B, F	198.	198.	197.	195.
T,GD,C, F	195.	196.	195.	193.

A7.4 Cont.

TEST NO.	69.	70.	71.	72.	73.
ENG,EXP	3.11	3.11	6.11	3.14	6.14
T,AIR, F	82.	82.	82.	82.	82.
ATM PRESS, PSI	14.78	14.78	14.78	14.78	14.78
VAP PRESS, PSI	.352	.352	.352	.352	.352
H2O/H2 RATIO	2.9	2.9	3.0	12.0	12.0
RPM	3300.	3300.	3300.	3300.	3300.
BMEP, PSI	24.9	25.5	25.5	84.0	84.0
SA, DEG	12.	0.	64.	17.	17.
VCF, %	36.	39.	82.	74.	74.
VOL % H2	31.1	30.7	11.1	30.8	30.8
PHI	1.07	1.05	.30	1.06	1.06
UPSILON	1.07	1.05	.30	1.06	1.06
BTE, %	15.4	15.0	19.6	25.7	25.7
NOX, PPM	77.0	94.0	1.50	55.0	55.0
NO, PPM	77.0	94.0	.50	55.0	55.0
NOX, G/HP-HR	.52	.66	.03	.30	.30
O2, MIX, % D	.5	.4	15.9	.4	.4
O2, MIX, % W	.3	.2	13.3	.2	.2
O2, CYL1, % D	.2	.4	15.7	.4	.4
O2, CYL2, % D	.2	.6	15.8	.4	.4
O2, CYL3, % D	1.0	.5	16.1	.6	.6
O2, CYL4, % D	.5	.3	15.8	.3	.3
T,EX 1,R, F	1244.	1300.	637.	1290.	1290.
T,EX 2,R, F	1250.	1341.	655.	1325.	1325.
T,EX 3,R, F	1259.	1328.	631.	1312.	1312.
T,EX 4,R, F	1258.	1336.	649.	1314.	1314.
T,EX 1,C, F	1392.	1467.	658.	1371.	1371.
T,EX 2,C, F	1398.	1508.	676.	1406.	1406.
T,EX 3,C, F	1407.	1495.	652.	1393.	1393.
T,EX 4,C, F	1406.	1503.	670.	1395.	1395.
T,SEAT 1, F	267.	243.	222.	254.	254.
T,SEAT 2, F	293.	263.	229.	276.	276.
T,SEAT 3, F	298.	271.	230.	275.	275.
T,SEAT 4, F	293.	261.	229.	273.	273.
T,H2O, F	187.	185.	185.	185.	185.
T,GD,A, F	244.	226.	203.	242.	242.
T,GD,B, F	223.	209.	194.	225.	225.
T,GD,C, F	217.	204.	192.	219.	219.

A7.4 Cont.

TEST NO.	74.	75.	76.	77.	78.	79.	80.
ENG,EXP	5.8	5.8	5.8	5.8	5.8	5.8	5.8
T,AIR, F	76.	76.	76.	76.	76.	76.	76.
ATM PRESS, PSI	14.75	14.75	14.75	14.75	14.75	14.75	14.75
VAP PRESS, PSI	.262	.262	.262	.262	.262	.262	.262
EGR, %	.0	4.9	9.7	14.6	19.5	24.3	29.2
RPM	1800.	1800.	1800.	1800.	1800.	1800.	1800.
BMEP, PSI	51.4	51.4	51.0	51.2	51.0	51.4	51.4
SA, DEG	30.	28.	26.	26.	24.	24.	24.
VCF, %	79.	79.	77.	77.	78.	76.	78.
VOL % H2	16.0	16.8	18.2	19.5	20.6	22.5	23.6
PHI	.45	.48	.53	.58	.62	.69	.74
UPSILON	.45	.47	.50	.53	.55	.61	.64
BTE, %	28.4	28.2	28.0	27.8	27.7	27.6	27.6
NOX, PPM	9.00	16.0	8.00	15.0	9.00	11.0	10.0
NO, PPM	3.00	10.0	2.50	5.00	3.00	4.00	4.00
NOX, G/HP-HR	.08	.13	.07	.12	.07	.07	.06
O2, MIX, % D	13.1	12.7	11.6	10.8	10.0	8.5	7.8
O2, MIX, % W	10.8	10.4	9.3	8.5	7.8	6.4	5.8
O2, CYL1, % D	13.0	12.7	11.7	10.9	10.0	8.5	7.7
O2, CYL2, % D	13.1	12.7	11.7	10.9	10.0	8.5	7.8
O2, CYL3, % D	12.9	12.6	11.5	10.7	9.8	8.5	7.9
O2, CYL4, % D	13.1	12.8	11.7	10.8	10.0	8.5	7.7
T,EX 1,R, F	748.	743.	766.	768.	783.	799.	802.
T,EX 2,R, F	753.	752.	775.	775.	786.	806.	811.
T,EX 3,R, F	746.	754.	771.	773.	781.	796.	802.
T,EX 4,R, F	753.	754.	779.	784.	793.	815.	822.
T,EX 1,C, F	792.	786.	814.	815.	831.	850.	851.
T,EX 2,C, F	797.	795.	823.	822.	834.	857.	860.
T,EX 3,C, F	790.	797.	819.	820.	829.	847.	851.
T,EX 4,C, F	797.	797.	827.	831.	841.	866.	871.
T,SEAT 1, F	235.	236.	236.	237.	237.	239.	240.
T,SEAT 2, F	250.	252.	251.	253.	254.	256.	257.
T,SEAT 3, F	257.	259.	258.	259.	260.	260.	260.
T,SEAT 4, F	244.	247.	246.	248.	248.	250.	251.
T,H2O, F	182.	182.	182.	182.	183.	183.	184.
T,GD,A, F	220.	221.	221.	221.	222.	223.	223.
T,GD,B, F	208.	209.	209.	209.	210.	210.	211.
T,GD,C, F	204.	205.	205.	206.	206.	207.	207.

A7.4 Cont.

TEST NO.	81.	82.	83.	84.	85.	86.	87.
ENG,EXP	2,8	2,8	2,8	2,8	2,8	2,8	2,8
T,AIR, F	76.	76.	76.	76.	76.	76.	76.
ATM PRESS, PSI	14.75	14.75	14.75	14.75	14.75	14.75	14.75
VAP PRESS, PSI	.262	.262	.262	.262	.262	.262	.262
EGR, %	.0	4.6	9.2	13.9	18.7	23.6	28.5
RPM	1805.	1800.	1800.	1800.	1800.	1800.	1800.
BMEP, PSI	51.7	51.4	51.4	51.0	51.0	51.4	51.4
SA, DEG	0.	0.	0.	0.	0.	0.	0.
VCF, %	52.	53.	55.	57.	61.	67.	72.
VOL % H2	30.4	30.1	30.1	30.1	30.1	30.3	30.3
PHI	1.04	1.03	1.03	1.03	1.02	1.04	1.03
UPSILON	1.04	1.02	1.02	1.02	1.01	1.02	1.01
BTE, %	22.8	23.5	24.0	24.0	24.0	23.0	23.0
NOX, PPM	4700.	3300.	2600.	700.	185.	49.0	30.0
NO, PPM	4700.	3300.	2600.	680.	175.	43.0	20.0
NOX, G/HP-HR	19.7	13.5	10.5	2.86	.77	.22	.15
O2, MIX, % D	1.0	1.2	1.2	1.2	1.4	1.3	1.4
O2, MIX, % W	.7	.8	.8	.8	.9	.9	.9
O2, CYL1, % D	.4	.5	.6	.5	.7	.6	.6
O2, CYL2, % D	.6	.9	.9	1.0	1.2	1.1	1.2
O2, CYL3, % D	1.7	2.0	2.0	1.9	2.1	2.1	2.3
O2, CYL4, % D	1.0	1.2	1.3	1.3	1.5	1.5	1.5
T,EX 1,R, F	1096.	1080.	1062.	1049.	1036.	1049.	1045.
T,EX 2,R, F	1133.	1098.	1076.	1066.	1059.	1069.	1065.
T,EX 3,R, F	1097.	1072.	1056.	1051.	1032.	1037.	1032.
T,EX 4,R, F	1115.	1093.	1079.	1068.	1053.	1059.	1056.
T,CX 1,C, F	1230.	1207.	1101.	1162.	1140.	1146.	1136.
T,EX 2,C, F	1267.	1225.	1195.	1179.	1163.	1166.	1156.
T,EX 3,C, F	1231.	1199.	1175.	1164.	1136.	1134.	1123.
T,EX 4,C, F	1249.	1220.	1198.	1181.	1157.	1156.	1147.
T,SEAT 1, F	274.	266.	255.	252.	250.	251.	251.
T,SEAT 2, F	297.	288.	276.	273.	271.	271.	270.
T,SEAT 3, F	302.	291.	280.	277.	273.	273.	272.
T,SEAT 4, F	300.	288.	278.	271.	265.	265.	265.
T,H2O, F	200.	195.	190.	186.	184.	185.	183.
T,GD,A, F	251.	246.	235.	235.	235.	234.	234.
T,GD,B, F	234.	227.	220.	220.	217.	217.	217.
T,GD,C, F	229.	223.	214.	214.	213.	213.	213.

A7.4 Cont.

TEST NO.	88.	89.	90.	91.	92.	93.	94.
ENG,EXP	2,8	2,8	2,8	2,8	2,8	2,8	2,8
T,AIR, F	67.	67.	67.	67.	67.	67.	67.
ATM PRESS, PSI	14.82	14.82	14.82	14.82	14.82	14.82	14.82
VAP PRESS, PSI	.157	.157	.157	.157	.157	.157	.157
EGR, %	.0	4.6	9.2	13.9	18.7	23.6	28.5
RPM	1805.	1800.	1800.	1800.	1800.	1800.	1800.
BMEP, PSI	52.0	52.0	52.0	52.0	52.0	51.8	51.8
SA, DEG	0.	3.	7.	10.	12.	14.	16.
VCF, %	52.	53.	55.	56.	60.	61.	65.
VOL % H2	30.4	30.5	30.5	30.6	30.6	30.7	30.7
PHI	1.04	1.04	1.04	1.05	1.05	1.05	1.05
UPSILON	1.04	1.04	1.04	1.04	1.04	1.03	1.03
BTE, %	22.8	23.4	23.9	24.3	24.3	25.1	25.1
NOX, PPM	4500.	3500.	2700.	1900.	900.	350.	200.
NO, PPM	4500.	3500.	2700.	1900.	900.	340.	180.
NOX, G/HP-HR	18.7	14.2	10.7	7.36	3.50	1.33	.79
O2, MIX, % D	1.0	1.0	1.0	.9	1.1	1.3	1.4
O2, MIX, % W	.7	.7	.7	.6	.7	.9	.9
O2, CYL1, % D	.4	.4	.4	.4	.5	.9	.9
O2, CYL2, % D	.6	.6	.6	.6	.8	1.0	1.1
O2, CYL3, % D	1.8	1.8	1.9	1.9	2.2	2.3	2.5
O2, CYL4, % D	1.1	1.1	1.1	1.3	1.5	1.7	1.8
T,EX 1,R, F	1098.	1080.	1050.	1018.	990.	972.	952.
T,EX 2,R, F	1115.	1092.	1065.	1035.	1011.	987.	965.
T,EX 3,R, F	1099.	1080.	1049.	1014.	987.	962.	937.
T,EX 4,R, F	1115.	1091.	1062.	1034.	1000.	970.	948.
T,EX 1,C, F	1231.	1205.	1164.	1122.	1083.	1056.	1027.
T,EX 2,C, F	1248.	1217.	1179.	1139.	1104.	1071.	1040.
T,EX 3,C, F	1232.	1205.	1163.	1118.	1080.	1046.	1012.
T,EX 4,C, F	1248.	1216.	1176.	1138.	1093.	1054.	1023.
T,SEAT 1, F	259.	255.	254.	253.	251.	249.	246.
T,SEAT 2, F	282.	278.	276.	274.	269.	265.	261.
T,SEAT 3, F	287.	286.	282.	279.	271.	266.	263.
T,SEAT 4, F	284.	283.	276.	269.	261.	255.	250.
T,H2O, F	190.	189.	187.	185.	184.	183.	183.
T,GD,A, F	241.	238.	232.	226.	224.	222.	220.
T,GD,B, F	226.	218.	216.	213.	211.	210.	208.
T,GD,C, F	220.	212.	211.	209.	207.	206.	205.

A7.4 Cont.

TEST NO.	95.	96.	97.	98.	99.
ENG,EXP	5.5	5.5	5.5	5.5	5.5
T,AIR, F	76.	76.	76.	76.	76.
ATM PRESS, PSI	14.75	14.75	14.75	14.75	14.75
VAP PRESS, PSI	.262	.262	.262	.262	.262
EGR, %	14.5	14.3	14.2	14.1	14.0
RPM	1800.	1800.	1800.	1800.	1800.
BMEP, PSI	57.3	63.5	67.5	72.7	75.0
SA, DEG	24.	20.	16.	11.	6.
VCF, %	77.	77.	76.	75.	75.
VOL % H2	20.8	23.1	24.6	27.4	28.8
PHI	.62	.72	.78	.90	.96
UPSILON	.58	.68	.74	.87	.95
BTE, %	29.1	29.2	28.9	28.8	28.3
NOX, PPM	35.0	210.	600.	1700.	2200.
NO, PPM	23.0	210.	600.	1700.	2200.
NOX, G/HP-HR	.21	.94	2.48	6.36	7.89
O2, MIX, % D	9.8	7.8	6.1	3.8	2.3
O2, MIX, % W	7.6	5.8	4.4	2.6	1.6
O2, CYL1, % D	9.9	7.8	6.0	3.9	2.3
O2, CYL2, % D	9.8	7.8	6.2	3.8	2.1
O2, CYL3, % D	9.8	7.8	6.2	3.9	2.5
O2, CYL4, % D	9.7	7.8	6.1	3.8	2.3
T,EX 1,R, F	784.	860.	906.	964.	1021.
T,EX 2,R, F	794.	864.	913.	971.	1020.
T,EX 3,R, F	792.	856.	907.	960.	1005.
T,EX 4,R, F	795.	872.	921.	983.	1028.
T,EX 1,C, F	833.	921.	978.	1044.	1110.
T,EX 2,C, F	843.	925.	985.	1051.	1109.
T,EX 3,C, F	841.	917.	979.	1040.	1094.
T,EX 4,C, F	844.	933.	993.	1063.	1117.
T,SEAT 1, F	235.	243.	245.	251.	255.
T,SEAT 2, F	251.	262.	264.	273.	278.
T,SEAT 3, F	258.	267.	270.	279.	284.
T,SEAT 4, F	244.	254.	259.	272.	280.
T,H2O, F	180.	184.	185.	187.	190.
T,GD,A, F	217.	222.	227.	233.	240.
T,GD,B, F	205.	210.	212.	219.	224.
T,GD,C, F	202.	206.	208.	215.	219.

A7.4 Cont.

TEST NO.	100.	101.	102.
ENG,EXP	2.5	2.5	2.5
T,AIR, F	67.	67.	67.
ATM PRESS, PSI	14.82	14.82	14.82
VAP PRESS, PSI	.157	.157	.157
EGR, %	13.9	13.9	13.9
RPM	1800.	1800.	1800.
BMEP, PSI	52.0	58.7	62.2
SA, DEG	10.	8.	8.
VCF, %	56.	62.	66.
VOL % H2	30.6	30.4	30.2
PHI	1.05	1.04	1.03
UPSILON	1.04	1.03	1.02
BTE, %	24.3	24.9	25.2
NOX, PPM	1900.	2100.	2250.
NO, PPM	1900.	2100.	2250.
NOX, G/HP-HR	7.36	8.01	8.53
O2, MIX, % D	.9	1.0	1.0
O2, MIX, % W	.6	.7	.7
O2, CYL1, % D	.4	.7	.8
O2, CYL2, % D	.6	.8	.8
O2, CYL3, % D	1.9	1.6	1.6
O2, CYL4, % D	1.3	1.1	1.0
T,EX 1,R, F	1018.	1023.	1026.
T,EX 2,R, F	1035.	1031.	1041.
T,EX 3,R, F	1014.	1020.	1023.
T,EX 4,R, F	1034.	1029.	1030.
T,EX 1,C, F	1122.	1121.	1123.
T,EX 2,C, F	1139.	1129.	1138.
T,EX 3,C, F	1118.	1118.	1120.
T,EX 4,C, F	1138.	1127.	1127.
T,SEAT 1, F	253.	259.	265.
T,SEAT 2, F	274.	278.	281.
T,SEAT 3, F	279.	286.	290.
T,SEAT 4, F	269.	273.	277.
T,H2O, F	185.	186.	186.
T,GD,A, F	226.	232.	235.
T,GD,B, F	213.	219.	220.
T,GD,C, F	209.	214.	218.

A7.4 Cont.

TEST NO.	103.	104.	105.	106.	107.
ENG,EXP	5,11	2,11	2,11	2,14	5,14
T,AIR, F	75.	75.	75.	75.	75.
ATM PRESS, PSI	14.75	14.75	14.77	14.80	14.80
VAP PRESS, PSI	.241	.241	.302	.416	.416
EGR, %	15.0	13.9	13.9	13.9	13.9
RPM	3300.	3300.	3300.	3300.	3300.
BMEP, PSI	24.0	24.0	24.0	70.0	70.0
SA, DEG	52.	0.	12.	13.	13.
VCF, %	79.	42.	39.	74.	74.
VOL % H2	12.9	30.3	30.3	30.6	30.6
PHI	.35	1.03	1.03	1.05	1.05
UPSILON	.31	1.03	1.03	1.04	1.04
BTE, %	19.2	15.4	16.4	25.0	25.0
NOX, PPM	2.50	360.	800.	2300.	2300.
NO, PPM	.90	360.	800.	2300.	2300.
NOX, G/HP-HR	.04	2.23	4.67	8.73	8.73
O2, MIX, % D	15.2	.8	.8	.9	.9
O2, MIX, % W	13.2	.5	.5	.6	.6
O2, CYL1, % D	15.2	.7	.7	.8	.8
O2, CYL2, % D	15.2	.9	.9	.9	.9
O2, CYL3, % D	15.2	.7	.7	.9	.9
O2, CYL4, % D	15.2	.7	.8	.8	.8
T,EX 1,R, F	654.	1254.	1181.	1301.	1301.
T,EX 2,R, F	669.	1265.	1190.	1320.	1320.
T,EX 3,R, F	652.	1258.	1180.	1309.	1309.
T,EX 4,R, F	661.	1286.	1216.	1331.	1331.
T,EX 1,C, F	676.	1409.	1321.	1411.	1411.
T,EX 2,C, F	691.	1420.	1330.	1430.	1430.
T,EX 3,C, F	674.	1413.	1320.	1419.	1419.
T,EX 4,C, F	683.	1441.	1356.	1441.	1441.
T,SEAT 1, F	230.	262.	263.	284.	284.
T,SEAT 2, F	239.	287.	271.	302.	302.
T,SEAT 3, F	243.	294.	256.	305.	305.
T,SEAT 4, F	236.	287.	280.	306.	306.
T,H2O, F	181.	185.	186.	186.	186.
T,GD,A, F	207.	235.	223.	245.	245.
T,GD,B, F	197.	215.	207.	227.	227.
T,GD,C, F	194.	209.	202.	220.	220.

A7.4 Cont.

TEST NO.	108.	109.	110.	111.	112.	113.	114.	115.
ENG,EXP	EXTRA	EXTRA	EXTRA	EXTRA	EXTRA	EXTRA	EXTRA	EXTRA
T,AIR, F	67.	67.	67.	67.	67.	67.	67.	67.
ATM PRESS, PSI	14.68	14.68	14.68	14.68	14.68	14.68	14.68	14.68
VAP PRESS, PSI	.328	.328	.328	.328	.328	.328	.328	.328
RPM	1800.	1795.	1790.	1800.	1800.	1800.	1800.	1800.
BMEP, PSI	52.0	51.7	51.5	52.1	52.0	51.7	52.3	51.7
SA, DEG	29.	22.	24.	15.	12.	9.	8.	5.
VCF, %	78.	73.	69.	68.	63.	58.	58.	56.
VOL % H2	16.6	18.0	19.7	20.0	22.2	24.8	25.2	26.5
PHI	.47	.52	.58	.59	.68	.78	.80	.86
UPSILON	.47	.52	.58	.59	.68	.78	.80	.86
BTE, %	28.2	27.5	26.7	26.9	26.2	25.1	25.1	24.3
NOX, PPM	12.0	24.0	150.	110.	470.	2500.	2900.	4350.
NO, PPM	5.00	21.0	145.	110.	470.	2500.	2900.	4350.
NOX, G/HP-HR	.11	.16	.90	.63	2.46	12.1	13.7	20.1
O2, MIX, % D	12.7	11.9	10.8	10.7	9.0	6.7	6.5	4.9
O2, MIX, % W	10.3	9.6	8.5	8.4	6.9	4.9	4.8	3.5
O2, CYL1, % D	12.7	11.6	N/A	10.2	8.6	6.0	N/A	N/A
O2, CYL2, % D	12.8	11.6	N/A	10.1	9.6	6.3	N/A	N/A
O2, CYL3, % D	12.7	12.6	N/A	11.3	9.2	7.3	N/A	N/A
O2, CYL4, % D	12.8	12.1	N/A	10.9	6.9	6.8	N/A	N/A
T,EX 1,R, F	753.	799.	831.	853.	951.	964.	977.	1005.
T,EX 2,R, F	759.	807.	847.	862.	953.	966.	983.	1014.
T,EX 3,R, F	755.	782.	822.	837.	889.	952.	973.	1001.
T,EX 4,R, F	763.	799.	827.	850.	909.	970.	989.	1016.
T,EX 1,C, F	799.	851.	890.	915.	1030.	1056.	1072.	1111.
T,EX 2,C, F	805.	859.	906.	924.	1032.	1058.	1078.	1120.
T,EX 3,C, F	801.	834.	881.	899.	968.	1044.	1068.	1107.
T,EX 4,C, F	809.	851.	886.	912.	988.	1062.	1084.	1122.
T,SEAT 1, F	233.	237.	238.	241.	243.	247.	248.	250.
T,SEAT 2, F	246.	251.	255.	258.	262.	267.	266.	271.
T,SEAT 3, F	252.	251.	254.	258.	263.	270.	270.	274.
T,SEAT 4, F	238.	242.	245.	248.	255.	266.	263.	272.
T,H2O, F	180.	182.	179.	183.	185.	186.	187.	189.
T,GD,A, F	213.	215.	215.	218.	222.	227.	230.	232.
T,GD,B, F	202.	204.	204.	207.	208.	213.	215.	218.
T,GD,C, F	199.	201.	201.	203.	204.	209.	210.	213.

A7.4 Cont.

TEST NO.	116.	117.	118.	119.	120.	121.	122.	123.
ENG,EXP	EXTRA	EXTRA	EXTRA	EXTRA	EXTRA	EXTRA	EXTRA	EXTRA
T,AIR, F	67.	67.	67.	67.	67.	67.	67.	67.
ATM PRESS, PSI	14.68	14.68	14.68	14.68	14.68	14.68	14.68	14.68
VAP PRESS, PSI	.328	.328	.328	.328	.328	.328	.328	.328
RPM	1810.	1800.	1800.	1800.	1800.	1805.	1800.	1790.
BMEP, PSI	51.8	50.6	52.4	52.0	50.6	52.0	50.6	50.1
SA, DEG	9.	3.	2.	0.	3.	0.	0.	0.
VCF, %	56.	53.	53.	52.	51.	52.	50.	51.
VOL % H2	26.6	28.8	30.0	30.1	30.5	30.4	32.0	33.4
PHI	.86	.96	1.02	1.03	1.04	1.04	1.12	1.19
UPSILON	.86	.96	1.02	1.03	1.04	1.04	1.12	1.19
BTE, %	24.3	23.4	23.1	22.9	22.7	22.8	22.0	20.8
NOX, PPM	5000.	5000.	5000.	4000.	5100.	4500.	1700.	875.
NO, PPM	5000.	5000.	5000.	4000.	5100.	4500.	1700.	875.
NOX, G/HP-HR	22.8	21.9	20.9	16.8	21.5	18.7	7.04	3.68
O2, MIX, % D	5.1	3.2	1.7	1.6	.9	1.0	.1	.0
O2, MIX, % W	3.7	2.2	1.1	1.1	.6	.7	.1	.0
O2, CYL1, % D	4.3	2.6	N/A	N/A	.1	.4	N/A	N/A
O2, CYL2, % D	4.7	3.1	N/A	N/A	.4	.6	N/A	N/A
O2, CYL3, % D	5.7	4.4	N/A	N/A	1.7	1.8	N/A	N/A
O2, CYL4, % D	5.2	3.6	N/A	N/A	1.1	1.1	N/A	N/A
T,EX 1,R, F	999.	1082.	1088.	1097.	1097.	1098.	1074.	1051.
T,EX 2,R, F	1011.	1095.	1099.	1107.	1104.	1115.	1102.	1079.
T,EX 3,R, F	997.	1082.	1084.	1093.	1076.	1099.	1129.	1113.
T,EX 4,R, F	1013.	1098.	1095.	1110.	1091.	1115.	1126.	1110.
T,EX 1,C, F	1102.	1207.	1215.	1226.	1228.	1231.	1202.	1163.
T,EX 2,C, F	1114.	1220.	1226.	1236.	1235.	1248.	1230.	1191.
T,EX 3,C, F	1100.	1207.	1211.	1222.	1207.	1232.	1257.	1225.
T,EX 4,C, F	1116.	1223.	1222.	1239.	1222.	1248.	1254.	1222.
T,SEAT 1, F	251.	265.	256.	256.	261.	259.	269.	270.
T,SEAT 2, F	271.	285.	281.	281.	285.	282.	293.	293.
T,SEAT 3, F	276.	292.	288.	287.	290.	287.	304.	305.
T,SEAT 4, F	269.	289.	284.	284.	287.	284.	299.	300.
T,H2O, F	186.	202.	192.	192.	196.	190.	205.	206.
T,GD,A, F	228.	249.	241.	242.	244.	241.	251.	255.
T,GD,B, F	215.	234.	226.	226.	229.	226.	237.	239.
T,GD,C, F	211.	229.	221.	221.	223.	220.	233.	235.

A7.4 Cont.

TEST NO.	124.	125.	126.	127.	128.	129.	130.
ENG,EXP	EXTRA	EXTRA	EXTRA	EXTRA	EXTRA	EXTRA	EXTRA
T,AIR, F	76.	76.	76.	76.	76.	76.	76.
ATM PRESS, PSI	14.79	14.79	14.79	14.79	14.79	14.79	14.79
VAP PRESS, PSI	.444	.444	.444	.444	.444	.444	.444
RPM	1800.	1800.	1800.	1800.	1800.	1800.	1800.
BMEP, PSI	.0	.0	.0	.0	.0	.0	.0
SA, DEG	94.	81.	69.	57.	45.	33.	27.
VCF, %	82.	55.	41.	36.	31.	28.	26.
VOL % H2	7.5	8.9	10.9	12.6	15.1	17.0	18.5
PHI	.19	.23	.29	.34	.42	.49	.54
UPSILON	.19	.23	.29	.34	.42	.49	.54
BTE, %	.0	.0	.0	.0	.0	.0	.0
NOX, PPM	.80	.90	1.20	1.30	2.20	3.60	7.00
NO, PPM	.50	.50	.60	.70	.80	2.20	2.70
NOX, G/HP-HR	UNDEF.	UNDEF.	UNDEF.	UNDEF.	UNDEF.	UNDEF.	UNDEF.
O2, MIX, % D	18.5	17.5	16.3	15.4	14.1	12.9	11.8
O2, MIX, % W	17.3	15.9	14.3	13.3	11.9	10.6	9.5
O2, CYL1, % D	18.5	17.4	16.0	15.1	13.5	12.0	10.8
O2, CYL2, % D	18.5	17.7	16.1	15.0	13.3	11.8	10.6
O2, CYL3, % D	18.7	17.6	16.7	15.9	14.9	14.0	13.1
O2, CYL4, % D	18.6	17.4	16.4	15.6	14.4	13.4	12.4
T,EX 1,R, F	366.	500.	589.	652.	680.	696.	710.
T,EX 2,R, F	357.	470.	592.	649.	682.	702.	711.
T,EX 3,R, F	332.	478.	565.	614.	643.	684.	691.
T,EX 4,R, F	368.	497.	584.	637.	661.	684.	688.
T,EX 1,C, F	372.	520.	626.	701.	738.	763.	782.
T,EX 2,C, F	363.	490.	629.	698.	740.	769.	783.
T,EX 3,C, F	338.	498.	602.	663.	701.	751.	763.
T,EX 4,C, F	374.	517.	621.	686.	719.	751.	760.
T,SEAT 1, F	195.	204.	208.	209.	215.	216.	216.
T,SEAT 2, F	198.	203.	209.	213.	221.	223.	224.
T,SEAT 3, F	197.	205.	208.	211.	214.	219.	220.
T,SEAT 4, F	196.	201.	204.	206.	209.	214.	215.
T,H2O, F	178.	178.	178.	179.	176.	179.	180.
T,GD,A, F	187.	188.	189.	190.	191.	193.	193.
T,GD,B, F	184.	184.	184.	185.	187.	188.	188.
T,GD,C, F	182.	182.	183.	183.	186.	186.	187.

A7.4 Cont.

TEST NO.	131.	132.	133.	134.	135.	136.	137.
ENG,EXP	EXTRA	EXTRA	EXTRA	EXTRA	EXTRA	EXTRA	EXTRA
T,AIR, F	76.	76.	76.	76.	76.	76.	76.
ATM PRESS, PSI	14.79	14.79	14.79	14.79	14.79	14.79	14.79
VAP PRESS, PSI	.444	.444	.444	.444	.444	.444	.444
RPM	1800.	1800.	1800.	1800.	1800.	1800.	1800.
BMEP, PSI	.0	.0	.0	.0	.0	.0	.0
SA, DEG	21.	18.	17.	9.	7.	6.	3.
VCF, %	23.	22.	22.	19.	19.	18.	18.
VOL % H2	20.8	22.2	23.0	27.3	28.4	29.1	31.4
PHI	.62	.68	.71	.89	.94	.98	1.09
UPSILON	.62	.68	.71	.89	.94	.98	1.09
BTE, %	.0	.0	.0	.0	.0	.0	.0
NOX, PPM	19.0	46.0	78.0	290.	350.	310.	210.
NO, PPM	9.00	28.0	57.0	270.	320.	290.	200.
NOX, G/HP-HR	UNDEF.	UNDEF.	UNDEF.	UNDEF.	UNDEF.	UNDEF.	UNDEF.
O2, MIX, % D	10.4	9.4	8.5	4.9	4.0	3.1	1.7
O2, MIX, % W	8.1	7.2	6.4	3.5	2.8	2.1	1.2
O2, CYL1, % D	9.3	8.1	7.1	3.2	2.1	1.0	.1
O2, CYL2, % D	8.8	7.7	6.7	2.6	1.4	.5	.3
O2, CYL3, % D	11.9	11.0	10.2	6.9	6.0	5.3	3.2
O2, CYL4, % D	11.2	10.3	9.4	6.0	5.2	4.4	2.1
T,EX 1,R, F	713.	712.	713.	747.	759.	768.	740.
T,EX 2,R, F	727.	731.	733.	769.	782.	794.	746.
T,EX 3,R, F	702.	693.	693.	720.	733.	738.	764.
T,EX 4,R, F	705.	699.	698.	728.	739.	744.	782.
T,EX 1,C, F	790.	789.	794.	840.	855.	868.	832.
T,EX 2,C, F	804.	808.	814.	862.	878.	894.	838.
T,EX 3,C, F	779.	770.	774.	813.	829.	838.	856.
T,EX 4,C, F	782.	776.	779.	821.	835.	844.	874.
T,SEAT 1, F	214.	214.	215.	217.	217.	219.	218.
T,SEAT 2, F	223.	224.	225.	230.	231.	232.	231.
T,SEAT 3, F	220.	221.	223.	228.	229.	238.	233.
T,SEAT 4, F	215.	216.	217.	222.	223.	224.	227.
T,H2O, F	184.	181.	182.	182.	182.	182.	182.
T,GD,A, F	196.	194.	195.	196.	197.	197.	198.
T,GD,B, F	191.	189.	189.	191.	191.	192.	192.
T,GD,C, F	189.	187.	188.	190.	190.	190.	191.

A7.4 Cont.

TEST NO.	138.	139.	140.	141.	142.	143.
ENG,EXP	EXTRA	EXTRA	EXTRA	EXTRA	EXTRA	EXTRA
T,AIR, F	67.	67.	67.	67.	67.	67.
ATM PRESS, PSI	14.82	14.82	14.82	14.82	14.82	14.82
VAP PRESS, PSI	.157	.157	.157	.157	.157	.157
RPM	1775.	1790.	1800.	1800.	1790.	1780.
BMEP, PSI	51.2	51.4	51.4	51.4	51.4	51.2
SA, DEG	21.	18.	16.	13.	11.	9.
VCF, %	64.	65.	64.	64.	64.	64.
VOL % H2	21.4	21.2	21.2	21.2	21.3	21.4
PHI	.65	.64	.64	.64	.64	.65
UPSILON	.65	.64	.64	.64	.64	.65
BTE, %	25.8	26.1	26.3	26.3	26.1	25.9
NOX, PPM	850.	470.	350.	250.	180.	160.
NO, PPM	850.	470.	350.	250.	180.	160.
NOX, G/HP-HR	4.67	2.59	1.91	1.36	.99	.88
O2, MIX, % D	9.7	10.0	9.9	9.9	9.9	9.8
O2, MIX, % W	7.6	7.9	7.8	7.8	7.8	7.7
O2, CYL1, % D	9.5	N/A	N/A	N/A	N/A	N/A
O2, CYL2, % D	9.5	N/A	N/A	N/A	N/A	N/A
O2, CYL3, % D	9.9	N/A	N/A	N/A	N/A	N/A
O2, CYL4, % D	9.6	N/A	N/A	N/A	N/A	N/A
T,EX 1,R, F	884.	873.	882.	888.	899.	905.
T,EX 2,R, F	880.	883.	891.	897.	908.	915.
T,EX 3,R, F	864.	865.	874.	881.	888.	893.
T,EX 4,R, F	868.	872.	882.	893.	903.	909.
T,EX 1,C, F	952.	940.	951.	958.	971.	978.
T,EX 2,C, F	948.	950.	960.	967.	980.	988.
T,EX 3,C, F	932.	932.	943.	951.	960.	966.
T,EX 4,C, F	936.	939.	951.	963.	975.	982.
T,SEAT 1, F	243.	240.	241.	238.	242.	244.
T,SEAT 2, F	259.	256.	258.	255.	258.	260.
T,SEAT 3, F	262.	259.	260.	258.	259.	261.
T,SEAT 4, F	251.	251.	250.	250.	249.	252.
T,H2O, F	185.	186.	183.	183.	183.	183.
T,GD,A, F	220.	220.	219.	216.	221.	222.
T,GD,B, F	207.	207.	207.	204.	207.	208.
T,GD,C, F	205.	205.	204.	201.	204.	205.

A7.4 Cont.

TEST NO.	158.	159.	160.	161.	162.	163.
ENG,EXP	EXTRA	EXTRA	EXTRA	EXTRA	EXTRA	EXTRA
T,AIR, F	70.	70.	70.	70.	70.	70.
ATM PRESS, PSI	14.76	14.76	14.76	14.76	14.76	14.76
VAP PRESS, PSI	.279	.279	.279	.279	.279	.279
RPM	1800.	1800.	1800.	1800.	1800.	1800.
BMEP, PSI	54.2	54.2	64.4	69.1	69.1	69.1
SA, DEG	28.	28.	20.	16.	16.	16.
VCF, %	80.	80.	79.	79.	79.	79.
VOL % H2	16.9	16.9	19.4	20.9	20.9	20.9
PHI	.48	.48	.57	.63	.63	.63
UPSILON	.48	.48	.57	.63	.63	.63
BTE, %	28.1	28.1	29.3	29.2	29.2	29.2
NOX, PPM	N/A	N/A	N/A	N/A	N/A	N/A
NO, PPM	N/A	N/A	N/A	N/A	N/A	N/A
NOX, G/HP-HR	UNDEF.	UNDEF.	UNDEF.	UNDEF.	UNDEF.	UNDEF.
O2, MIX, % D	12.5	12.5	10.6	9.6	9.6	9.6
O2, MIX, % W	10.2	10.2	8.3	7.4	7.4	7.4
O2, CYL1, % D	N/A	N/A	N/A	N/A	N/A	N/A
O2, CYL2, % D	N/A	N/A	N/A	N/A	N/A	N/A
O2, CYL3, % D	N/A	N/A	N/A	N/A	N/A	N/A
O2, CYL4, % D	N/A	N/A	N/A	N/A	N/A	N/A
T,EX 1,R, F	778.	780.	853.	894.	899.	898.
T,EX 2,R, F	797.	797.	879.	921.	924.	925.
T,EX 3,R, F	779.	783.	853.	897.	898.	898.
T,EX 4,R, F	779.	779.	856.	900.	899.	901.
T,EX 1,C, F	826.	829.	915.	962.	967.	966.
T,EX 2,C, F	845.	846.	941.	989.	992.	993.
T,EX 3,C, F	827.	832.	915.	965.	966.	966.
T,EX 4,C, F	827.	828.	918.	968.	967.	969.
T,SEAT 1, F	232.	231.	238.	244.	244.	244.
T,SEAT 2, F	248.	248.	257.	274.	265.	265.
T,SEAT 3, F	252.	252.	263.	269.	270.	269.
T,SEAT 4, F	244.	243.	255.	259.	260.	260.
T,H2O, F	185.	185.	186.	186.	186.	186.
T,GD,A, F	213.	212.	219.	220.	220.	220.
T,GD,B, F	201.	201.	205.	205.	205.	205.
T,GD,C, F	198.	198.	201.	201.	201.	201.

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APPENDIX 8 EXHAUST GAS TEMPERATURE

A8.1 CALCULATION OF APPROXIMATE EXHAUST-GAS TEMPERATURE

The temperature of the wall of the exhaust manifold of an operating engine is different than the temperature of the flowing exhaust gas. This difference results in the temperature of thermocouple probes placed in the exhaust manifold being different than that of the exhaust gas. In order to arrive at an approximate value of the exhaust-gas temperature the equations presented in this Appendix, along with the assumptions given, were used.

The first equation, derived from the first law of thermodynamics,

$$q_{g-p} + q_c + q_{rad} = 0 , \quad (1)$$

states that, for steady state, the net heat transfer rate to the probe by convection from the exhaust gas (q_{g-p}) plus the heat conducted away from and along the probe (q_c) plus the net radiation heat transfer rate away from the probe (q_{rad}) is zero.

Calculations showed that

$$q_c \approx 0 . \quad (2)$$

The probe convection heat transfer rate was approximated by

$$q_{g-p} \approx h_p \chi (T_g - T_p) . \quad (3)$$

In Equation (3): (1) it is assumed that convection heat transfer to the probe occurs only when the exhaust valve is open for a fraction χ of the cycle (calculations showed that the convective heat transfer during the remaining $(1 - \chi)$ fraction of the cycle was negligibly small; $\chi = 0.35$ for the present-study engine); (2) T_g is the average exhaust gas temperature during the χ fraction of the cycle; (3) T_p is the temperature of the probe as indicated by its EMF output and (4) the heat transfer coefficient

at the probe-exhaust gas interface, h_p , is given by the equation for a circular cylinder in cross flow (Kreith, 1968). The exhaust-gas velocity used to calculate h_p by the method given in Kreith was a mass-flow averaged velocity.

The radiation heat transfer rate from the probe, $q_{rad, p}$, is given by

$$q_{rad, p} = \sigma \epsilon_p (T_p^4 - T_{env}^4) . \quad (4)$$

In Equation (4), which is for a small convex object in a large enclosure, σ is the Stefan-Boltzman constant (0.1714×10^{-8} BTU/hr $ft^2 R^4$), ϵ_p is the emissivity of the stainless-steel probe ($\epsilon_p = 0.05$ was used) and T_{env} is the temperature of probe's radiating environment. (Radiation heat transfer between the probe and the exhaust gas was found to be negligibly small). The value of T_{env} was approximated by

$$T_{env} \approx 0.95 T_{si} + 0.05 T_{gas \text{ assembly probe}} \quad (5)$$

where T_{si} is the temperature of the exhaust manifold inside wall and $T_{gas \text{ assembly probe}}$ is the temperature of the 0.25 in. OD exhaust-gas sampling probe placed near the thermocouple probe. The latter probe occupied about 5% of the solid angle "seen" by the thermocouple thus the presence of the fraction 0.95 and 0.05 in Equation (5). The inside walls of the exhaust manifold were assumed to have a uniform and constant temperature, T_{si} , as determined by the methods discussed below. The exhaust-gas sampling probe, as reflected by the following equation, was assumed to be at the same temperature as the thermocouple:

$$T_{gas \text{ sampling probe}} \approx T_p . \quad (6)$$

Note that while convective heat transfer to the probe, q_{g-p} , is assumed to occur only when the exhaust valve is opened ($\chi = 0.35$),

radiation heat transfer $q_{\text{rad}, p}$ is assumed to occur all the time.

The approximate exhaust-gas temperature can be calculated from Equations (1) - (6) with the exception that T_{si} , the temperature of the inside exhaust manifold wall is not known. To arrive at T_{si} the first law was applied to a section of the wall comprising the exhaust manifold yielding:

$$q_{\text{g-si}} = q_{\text{so-}\infty} . \quad (7)$$

Equation (7) states that, for steady state, the heat transfer rate from the exhaust gas to the inside exhaust manifold wall, $q_{\text{g-si}}$, is equal to the heat transfer rate from the outside manifold-wall surface to the atmosphere, $q_{\text{so-}\infty}$. The radiation heat transfer rate on the inside of the manifold was found to be negligibly small. The Equation (7) heat transfer rates were found as described below.

The convective heat transfer rate per unit area at the inside exhaust manifold wall is given by

$$q_{\text{g-si}} = h_{\text{g-si}} (T_{\text{g}} - T_{\text{si}}) \quad (8)$$

where $h_{\text{g-si}}$ is the average heat transfer coefficient at the exhaust gas/inside-exhaust manifold wall interface. The coefficient, $h_{\text{g-si}}$, was estimated by using the following correlations:

- (1) Kays (1969) page 170, Figure 9-8.
- (2) Rohsenow and Choi (1961) page 196, Figure 8.12.
- (3) Kays (1969) page 195, Figure 9-14.

To use the above-referenced correlations, the following assumptions were made:

- (1) The exhaust-gas velocity was constant during the fraction x of the cycle when the exhaust valve was open. The value of the velocity was a mass-averaged one.
- (2) The pressure in the exhaust manifold was atmospheric.
- (3) The Prandtl Number of the exhaust gas was 0.7.

- (4) The viscosity of the exhaust gas constituents were given by the Southerland formula (Rohsenow and Choi, 1961).
- (5) The viscosities of the exhaust-gas constituents were additive (Chapman and Cowling, 1964).
- (6) The thermal conductivity of the exhaust gas was approximated by an equation given by Tsederberg (1965).
- (7) The thermal conductivities of the exhaust-gas constituents were additive (Tsederberg, 1965).

The heat transfer at the outside surface of the exhaust manifold, q_{so-i} , was expressed in terms of an outside surface radiation plus convection heat transfer coefficient, h_{so} :

$$q_{so-i} = h_{so} (T_{so} - T_{\infty}) \quad (9)$$

where T_{so} is the temperature of the outside surface and T_{∞} is the room temperature. Calculations showed that there was a negligible temperature drop across the cast-iron exhaust manifold wall as reflected by the following equation:

$$T_{so} \approx T_{si} \quad (10)$$

To arrive at an estimation of h_{so} calibration tests were performed. These values of h_{so} were then used for subsequent tests.

A8.2 NOMENCLATURE/COMPUTER PROGRAM

The Data-Reduction Computer Program, Appendix 7, contains the above-described corrections to the exhaust gas thermocouple temperature measurements. Listed below are the terms used in the A8.1 analysis and their corresponding Appendix 7.1 Fortran Symbols.

<u>A8.1</u>	<u>A7.1</u>
T_{∞}	AIRTEMP
ϵ_p	EM
h_{si}	HIPO
h_{so}	HO
h_p	HTHO
T_p	T
$T_p - T_g$	TDEL
T_g	TEXHI
T_{so}	TS

A8.3 LIMITATIONS

One limitation of the model given in this Appendix is that the heat transfer coefficients used were based on the works of others. In addition, there were a number of simplifying assumptions made. Thus, while the model is felt to be useful for making comparisons, caution must be exercised in using the absolute values of exhaust-gas temperatures presented in Appendix 7.

Future activities will focus attention on determining which of the model assumptions are critical and how variations in values of heat transfer coefficients, as determined from the work of others, effect the results given by the model. If necessary, additional measurements in the present-study engine will be made along with modifications of the model and assumed heat transfer coefficients to arrive at a better approximation of exhaust-gas temperature levels.

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APPENDIX 9 CALCULATION OF EXCESS HYDROGEN IN THE EXHAUST AND H₂ UTILIZATION COEFFICIENT

The equation and measurements used to determine the unburned hydrogen in the exhaust and the H₂ Utilization Coefficient are given below:

Since the exhaust-gas was assumed to consist only of H₂, O₂, N₂, and H₂O the following chemical equation applies in which there are eight unknowns (a, b, c, d, e, f, g, h):

Four of the unknowns can be determined from measurements:

$$a = \dot{m}_{H_2} \quad (2)$$

is the measured mass flowrate of hydrogen;

$$b + c + d = \dot{m}_A \quad (3)$$

is the measured mass flowrate of atmospheric (wet) air ($\dot{m}_A$);

$$\frac{d}{(a + b + c + d)} = \frac{P_w}{P_{at}} \quad (4)$$

where P_w = partial pressure of water vapor in atmospheric air and P_{at} = atmospheric pressure; and

$$\frac{f}{e + f + g} = [O_2] \quad (5)$$

is the fraction of oxygen in dry exhaust gas.

Dry air is assumed to consist of 79% by volume nitrogen, therefore

$$c = 3.76 b \quad (6)$$

Conservation of mass applied to nitrogen, oxygen and hydrogen yields the following respective equations:

$$c = g \quad (7)$$

$$2b + d = 2f + h \quad (8)$$

$$a + d = e + h \quad (9)$$

Equations (1) - (9) can be solved to yield:

$$e = 2f + \dot{m}_{H_2} - \frac{2 \dot{m}_A}{4.76} \frac{P_{At} - P_w}{P_{At}} \quad (10)$$

where

$$f = 0.37 \dot{m}_{H_2} + \dot{m}_A \frac{P_{At} - P_w}{P_{At}} \quad (11)$$

The fraction of unburned hydrogen is therefore

$$[H_2] = \frac{e}{\dot{m}_{H_2}} \quad (12)$$

and the H_2 Utilization Coefficient

$$HU = \frac{\dot{m}_{H_2} - e}{\dot{m}_{H_2}} \cdot 100 \quad (13)$$

A10.1 - ELECTRICAL ANALOGUE - DESCRIPTION

An electrical heat transfer to heat flow analogue has been used to characterize heat transfer through the sodium- and lithium-filled exhaust valves used during testing (See Section 7.5 for a description of the hollow-stem exhaust valve used and Appendix 3 for a description of the method used to construct the lithium-filled valves). The analogue is similar in method to the electrical analogues used by Hires and Pochmara (1976), Cerruti et al (1953), and Stotter et al (1965) but different in detail. The analogue model was used to quantify the relative peak valve-head operating temperatures and heat transfer rates of the sodium-filled valves vis-a-vis the lithium-filled valves for specific test conditions. The model was not intended to be used for determination of absolute temperatures or heat transfer rates because of the inaccuracies associated with assumed boundary conditions. The test conditions used for the comparison were unthrottled operation at 34.8, 41.4 and 44.4 ft lb torque and 1800 RPM (Appendix 7, TEST NO. 34, 35 and 36 respectively). TEST NO. 36 corresponds to peak load as limited by flashback for unthrottled operation at 1800 RPM, for sodium filled valves with heavy deposits.

The analogue circuit was constructed by representing the heat transfer coefficients and thermal resistances by electrical resistances, R , using the following relationships: (1) For convective boundary conditions

$$R_{\text{convection}} = \frac{1}{hA} \quad (1)$$

where: h is the heat transfer coefficient and A the surface area;
(2) For heat conduction through an approximately slab-like solid

$$R_{\text{conduction}} = \frac{L}{kA} \quad (2)$$

where: L is the length of the heat transfer path, A is the area of the path and k is the approximate thermal conductivity of the material; (3) For heat conduction through approximately cylindrical geometry

$$\frac{\ln r_o/r_i}{2\pi k \Delta L} \quad (3)$$

where: r_o and r_i are the outer and inner radius of the cylindrical geometry and ΔL is the length of the cylinder.

Known temperatures were represented by voltage sources with the coolant temperature representing electrical ground. The resulting circuit (Figure A10-1) was solved using the Princeton Circuit Analysis Program PCAP, (1972).

A10.2 - ELECTRICAL ANALOGUE - ASSUMPTIONS

To construct the electrical analogue model, certain simplifying assumptions were incorporated. They are as follows:

- 1) Heat transfer across boundaries was assumed to be steady state.
- 2) Radial symmetry was assumed around the valve stem: The temperatures measured at the three locations, Gd a, Gd b, and Gd c, on the valve guide (See Section 7.5 and Figure 3) were assumed to be representative of the average temperature of the corresponding axial section at the outside surface of the valve guide.
- 3) The valve seat and the portion of the cylinder head surrounding the seat were separated into two equal 180° sections. The temperature measured at T, seat j (Figure 3) was assumed to be representative of the average temperature at the outside-diameter surface of the seat for one of the the 180° sections. The temperatures at the other 180° section was estimated by calculating representative, average resistance values for the cylinder head and water passage using cylinder head cross sections for length, L, and area, A, approximations (in an expression like Equation (3)).

A10.3 - KNOWN TEMPERATURES

Temperatures were measured at three locations along the valve guide (Figure 3) and at one location at the valve seat (Figure 3) to: (1) determine the respective boundary conditions at the various operating conditions that were evaluated and (2) to determine the value of the thermal resistance at the valve coolant-cavity wall

interface. (This determination was made by adjusting the value of this resistance until the heat transfer rate through the valve stem equaled the heat transfer rate to the coolant water as described below in A10.6). Figure 3 shows the location of the three valve guide thermocouples at the cylinder head-guide interface, which corresponds to nodes 6, 11 and 16 respectively in Figure A10-1. The thermocouples were located in the engine at locations where maximum axial temperature gradients were expected. The valve seat thermocouple was located in a position where the temperature was assumed to be representative of the average seat temperature for the 180° section; Node 22 (Figure A10-1) represents this temperature. Exhaust-gas temperatures were measured at the port exit. An approximation was used to correct these measurements for the radiation heat transfer effect on the thermocouple probe (See Appendix 8). The temperature drop of the exhaust gas from the valve to the port exit was approximated using the following relationship:

$$\Delta T = \frac{Q}{\dot{m} C_p} \quad (4)$$

where: the heat transfer rate, Q , was determined from the results of Rush (1976) for similar port geometry, and corrected for the flowrate and exhaust gas properties of the present-study; the exhaust gas mass flowrate, $\dot{m}$, was known from fuel and air mass flowrate measurements; and the specific heat of the exhaust gas, C_p , was approximated by taking the mass average for the constituents of the exhaust gas.

The resulting exhaust-gas temperature near the valve is represented by Node 24 in Figure A10-1.

The engine coolant temperature was measured at the front of the cylinder head where the coolant exits to the radiator. The temperature of the coolant in the vicinity of the exhaust valve of the #4 cylinder, the one where the Figure 3 temperatures were measured, was assumed to be 3°F cooler than the cylinder head exit temperature. Measurements verified this value as being a reasonable estimate; the value is also in agreement with results calculated by the authors for a four-cylinder Pontiac engine.

The cycle-average temperature of the gases in the

combustion chamber were obtained from the cylinder-pressure/time diagrams (See Section 8.6) making two approximations:

- (1) the temperatures, which were calculated at 10 degree intervals, were found using the ideal gas relation $PV=nRT$;
- (2) the change in number of moles due to combustion was assumed to occur instantaneously at TDC. The cycle average temperatures thus calculated were 895°F , 1269°F and 1308°F for TEST NO. 34, 35 and 36 respectively.

A10.4 - THERMAL CONDUCTIVITIES AND CONTACT RESISTANCE

The thermal conductivity (k) for the martinsitic material of the Alfa Romeo exhaust valve (Section 7.5) was assumed to be $1.2 \text{ BTU/hr in. }^{\circ}\text{F}$ and $1.4 \text{ BTU/hr in. }^{\circ}\text{F}$ at 200 and 900 degrees Fahrenheit respectively. The cylinder head was assumed to be 390 aluminum with $k = 6.0 \text{ BTU/hr in. }^{\circ}\text{F}$. The cast iron alloy seat was assumed to have a $k = 1.9 \text{ BTU/hr in. }^{\circ}\text{F}$. The Alfa Romeo brass alloy valve guide was assumed to have a $k = 5.75 \text{ BTU/hr in. }^{\circ}\text{F}$.

The contact resistance between two dissimilar metals is a difficult resistance to predict, being dependent upon surface condition and history. The General Electric Data Book (GE 1970) was consulted for approximate values for the contact resistance for metals similar to the valve seat and cylinder head. Data for a low alloy steel and aluminum (Duraluminum) showed the contact resistances to be small for similar finishes (compared to other resistances in the model). This information, in conjunction with the experimental results that thermal cycling of press fit parts has shown a marked reduction in contact resistance (Williams, 1977) suggests that is a good assumption to assign a negligible value to the contact resistance between the valve seat and the cylinder head.

A10.5 - HEAT TRANSFER COEFFICIENTS

The heat transfer coefficients between the sliding surface of the valve stem and the valve guide was assumed equal to $1.4 \text{ BTU/hr in. }^2^{\circ}\text{F}$ (Danis, 1973). The water-to-cylinder head convective coefficient for the 180° section in the vicinity of the exhaust valve was assumed equal to $8.0 \text{ BTU/hr in. }^2^{\circ}\text{F}$ (Danis, 1973). The heat transfer coefficient for the valve-to-seat

contact was assumed equal to 5.0 BTU/hr in.² °F (Stotter, 1965; Danis, 1973). In arriving at this value the valve was considered to be closed for 64% of the cycle (or approximately 461 degrees crank shaft rotation).

Previous investigators have used the value of approximately 2000°F for the average temperature of the gases in the combustion chamber when determining heat transfer coefficients at the valve face for known or assumed heat transfer rates. In one case this value was traced back to Taylor and Taylor (1966) who in turn referred to NACA TR 813 (1945). The NACA work used the values of approximately 2000 - 2300°F for gas temperature to calculate heat transfer coefficients on both sides of the exhaust valve head. These temperature values, as states in the NACA work, were not meant to be representative of cycle average temperature which are found to take on lower values. Thus, to arrive at an approximate heat transfer coefficient at the valve head, the coefficient given by Danis (1973) which was calculated using a 2100°F cycle average temperature was revised assuming 1350°F for his cycle average temperature. The resulting value for the coefficient was .25 BTU/hr in.² °F.

The heat transfer coefficients at the exhaust valve stem, fillet and backside of valve head were approximated by using the results of Danis (1973) (See Figure 2) and adjusted for the present-study Reynolds numbers using the relationship:

$$h \approx \frac{K}{D} \text{Re}^{.8} \quad (5)$$

An approximate value of the exhaust-port wall heat transfer coefficient was obtained from the data presented by Rush (1976) taking into account the present-study engine port geometry.

The heat transfer coefficient for the exhaust-valve fillet determined from the Danis paper (1973) agreed very well with the port wall heat transfer coefficient determined from the Rush paper (1976). This agreement was expected and lends credence to the values of heat transfer coefficients used in this Appendix.

A10.6 - METHOD OF SOLUTION

A uniform temperature of the coolant (lithium or sodium) was assumed to be representative of the heat transfer mechanism in the stem. The heat flow from the valve guide to the coolant was approximated by the electrical analogue model shown in Figure A10-2. The assumed resistance values for the aluminum and the coolant-to-cylinder head were the same as assumed for the valve model. The geometry of the model was chosen as being representative of an "average" axial section through the cylinder head at the valve stem, as approximated by measurements taken from cylinder head cross sections. Nodes 12, 13, 14 correspond to Figure 3 temperatures Gd a, Gd b and Gd c respectively and Node 0 to water temperature (ground in circuit). The values in the analogue circuit were set equal to the values as measured for TEST NO. 34, 35, 36, 158, 160 and 163 to obtain the approximate heat flow through the valve stem to the coolant for those operating conditions. Results of the model appear in Table A10-2. Once the stem heat flow was obtained for each data point the valve model was fit to this heat flow by varying the heat transfer coefficient at the coolant fluid / valve-stem-cavity-wall interface (h_{f-w}). It is felt reasonable to expect an unchanging value for this heat transfer coefficient for the three operating conditions considered since the RPM is constant and the fluid properties for the internal coolant do not change appreciably. Therefore, a mean value for this heat transfer coefficient was chosen, a value that approximated the stem heat flow for all three operating conditions.

Results of the valve stem heat flow for the sodium valves (with an average h_{f-w} of 14.0 BTU/hr in.² °F) and for the lithium valves (with an average h_{f-w} of 9.0 BTU/hr in.² °F), are compared to the results of the valve guide to coolant model heat flow in Table A10-1. The results are seen to be in good agreement.

A10.7 - RESULTS

A print-out of all node voltages and branch currents from PCAP is given in Table A10-2. To obtain the corresponding temperatures (°F) at any node, add the following corresponding coolant temperature to the node voltage value:

Test No.	34	35	36	158	160	163
Coolant Temperature ($^{\circ}\text{F}$)	182	183	182	182	183	183

The heat flow through any branch is given as branch currents in BTU/hr.

Temperatures and heat transfer rates of primary interest are given in Table A10-3. There it is seen that the lithium-filled exhaust valves did not exhibit the expected increase in valve stem heat transfer or reduction in valve-head temperature vis-à-vis the sodium-filled valves (Danis, 1973). Maximum load, as limited by flashback, was reached with the sodium-filled valves at 46.0 ft lbs. Maximum load with the lithium-filled valves occurred at 50.0 ft lbs. It is suspected that this difference in flashback-limited load was due to a difference in deposit build-up on the valve head and not a difference in valve-head temperature: Inspection of the valves revealed that the sodium-filled valves had heavy deposits on the valve head; the lithium-filled valves due to the short period of their use during unthrottled operation, however, were relatively deposit free. (Since exhaust-gas temperature has a strong influence on exhaust-valve temperature, it should be pointed out that for the Table A10-3 tests, due care was exercised to insure that constant exhaust-gas temperatures were maintained).

During the course of the present project, communication with the Valve Division of the Eaton Corporation has been maintained on a regular basis. These communications revealed that the activities of the present study regarding exhaust-valve temperature and heat transfer characteristics parallel closely to activities in industry where exhaust-valve-induced preignition is presently a limiting factor setting octane requirements for automotive engines (Eaton, 1978). Also, results similar to those of the present study have been recently experienced with lithium-filled valves during tests conducted at Ford. Close communications will be maintained in the future.

A10.8 - LIMITATIONS

One limitation of the model given in this Appendix is that

the heat transfer coefficients used were based on the works of others. In addition, there were a number of simplifying assumptions made regarding the geometry of the cylinder head and temperature profiles. Thus, while the model is felt to be useful for making comparisons of one type of exhaust valve with another, caution must be exercised in using absolute values of temperature and/or heat transfer rates.

Future activities will focus attention on determining which of the model assumptions are critical and how variations in values of heat transfer coefficients, as determined from the works of others, effect the results given by the model. If necessary, additional measurements in the present-study engine will be made along with modifications of the model and assumed heat transfer coefficients to arrive at a better approximation of exhaust-valve temperature levels and heat transfer rates.

TABLE A10-1 - VALVE STEM HEAT FLOW COMPARISON FOR THE GUIDE
TO COOLANT AND EXHAUST VALVE ANALOGUE MODELS

Test No.	34	35	36	158	160	163
Q_{gc}^*	343	392	400	317	366	368
Q_v^{**}	325	386	409	305	361	384

* Q_{gc} - resulting heat transfer (BTU/hr) from valve stem to coolant
from the valve guide to coolant analogue model
(See Figure A10-2)

** Q_v - resulting heat transfer (BTU/hr) from the valve stem for
the exhaust valve analogue model (See Figure A10-1)

TABLE A10-2

ELECTRICAL ANALOG FOR THE VALVE GUIDE TO COOLANT: TEST NO.34

DC ANALYSIS

B1 N(3,4), R=.073
 B2 N(4,5), R=.052
 B3 N(2,6), R=.024
 B4 N(6,7), R=.016
 B5 N(4,6), R=.42
 B6 N(6,8), R=.22
 B7 N(1,8), R=.21
 B8 N(8,9), R=.030
 B9 N(5,10), R=.28
 B10 N(7,10), R=.070
 B11 N(9,10), R=.070
 B12 N(1,C), R=.00001,E=19.
 B13 N(2,C), R=.00001,E=21.
 B14 N(3,C), R=.00001,E=30.
 B15 N(10,C), R=.00001
 TAB, NV(10), I(15)
 EXECUTE

NODES

NODE VOLTAGES

NODES	1- 7	8- 10	11- 12	13- 14	15- 16	17- 18	19- 20	21- 22
NODE VOLTAGES	1.89995+01	2.09930+01	2.99991+01	2.35333+01	1.98479+01	1.60994+01	1.31048+01	
	8.47797+00	5.93531+00	3.42781-03					

BRANCHES

BRANCH CURRENTS

BRANCHES	1- 7	8- 14	15- 15	16- 16	17- 17	18- 18	19- 19	20- 20
BRANCH CURRENTS	8.85728+01	7.08731+01	2.04106+02	1.87163+02	1.76997+01	3.46429+01	5.01025+01	
	8.47454+01	7.08731+01	1.87163+02	6.47454+01	-5.01633+01	-2.04206+02	-8.86440+01	
	3.42781+02							

EXECUTION TIME = .424 SEC

TABLE A10-2 Cont. ELECTRICAL ANALOG FOR THE VALVE GUIDE TO COOLANT: TEST NO.35

DC ANALYSIS

31 N(3,4), R=.073
 32 N(4,5), R=.052
 33 N(2,6), R=.024
 34 N(6,7), R=.016
 35 N(4,6), R=.42
 36 N(6,8), R=.22
 37 N(1,8), R=.21
 38 N(8,9), R=.030
 39 N(5,10), R=.28
 310 N(7,10), R=.070
 311 N(9,10), R=.070
 312 N(1,0) R=.00001,E=21.
 313 N(2,0), R=.00001,E=24.
 314 N(3,0), R=.00001,E=35.
 315 N(10,0), R=.00001
 TAB, NV(10), I(15)
 EXECUTE

NODES

NODE VOLTAGES

NODES	1- 7	2- 10	3- 14	4- 15	5- 16	6- 17	7- 18	8- 19
	2.09995+01	2.39977+01	3.49990+01	2.74086+01	2.31163+01	1.84065+01	1.49827+01	
	9.51462+00	6.66141+00	3.91634-03					

BRANCHES

BRANCH CURRENTS

BRANCHES	1- 7	2- 14	3- 15	4- 16	5- 17	6- 18	7- 19	8- 20
	1.03978+02	8.25442+01	2.32967+02	2.13983+02	2.14336+01	4.04174+01	5.46897+01	
	9.51071+01	8.25442+01	2.13983+02	9.51071+01	-5.47647+01	-2.33054+02	-1.04046+02	
	3.91634+02							

EXECUTION TIME = .444 SEC

TABLE A10-2 Cont. ELECTRICAL ANALOG FOR THE VALVE GUIDE TO COOLANT: TEST NO.36

DC ANALYSIS

R1 N(3,4), R=.073
 R2 N(4,5), R=.052
 R3 N(2,6), R=.024
 R4 N(6,7), R=.016
 R5 N(4,6), R=.42
 R6 N(6,8), R=.22
 P7 N(1,8), R=.21
 R8 N(8,9), R=.030
 R9 N(5,10), R=.28
 R10 N(7,10), R=.070
 R11 N(9,10), R=.070
 R12 N(1,0) R=.00001,E=22.
 R13 N(2,0) R=.00001,E=24.
 R14 N(3,0) R=.00001,E=37.
 R15 N(10,0) R=.00001
 TAG, NV(10), I(15)
 EXECUTE

NODES

NODE VOLTAGES

1- 7	2.19994+01	2.39977+01	3.69989+01	2.88532+01	2.43346+01	1.84836+01	1.50455+01
8- 10	9.77946+00	6.84582+00	3.99529-03				

BRANCHES

BRANCH CURRENTS

1- 7	1.11585+02	8.68952+01	2.29754+02	2.14879+02	2.46395+01	3.95643+01	5.81903+01
8- 14	9.77546+01	8.68952+01	2.14879+02	9.77546+01	-5.82695+01	-2.29836+02	-1.11723+02
15- 15	3.99529+02						

EXECUTION TIME = .428 SEC

TABLE A10-2 Cont. ELECTRICAL ANALOG FOR THE VALVE GUIDE TO COOLANT: TEST NO.158

DC ANALYSIS

B1 N(3,4), R=.073
 B2 N(4,5), R=.052
 B3 N(2,6), R=.024
 B4 N(6,7), R=.016
 B5 N(4,6), R=.42
 B6 N(6,8), R=.22
 B7 N(1,8), R=.21
 B8 N(8,9), R=.030
 B9 N(5,10), R=.28
 B10 N(7,10), R=.070
 B11 N(9,10), R=.070
 B12 N(1,3), R=.00001,E=16.
 B13 N(2,3), R=.00001,E=19.
 B14 N(3,3), R=.00001,E=31.
 B15 N(10,0), R=.00001
 TAB, NV(10), I(15)
 EXECUTE

NODES

NODE VOLTAGES

1- 7	1.59996+01	1.89982+01	3.09991+01	2.40707+01	2.03010+01	1.46555+01	1.19295+01
8- 10	7.39801+00	5.17955+00	3.16816-03				

BRANCHES

BRANCH CURRENTS

1- 7	9.49096+01	7.24924+01	1.80947+02	1.70376+02	2.24171+01	3.29885+01	4.09599+01
8- 14	7.39484+01	7.24924+01	1.70376+02	7.39484+01	-4.10080+01	-1.81055+02	-9.49860+01
15- 15	3.16616+02						

EXECUTION TIME = .429 SEC

TABLE A10-2 Cont. ELECTRICAL ANALOG FOR THE VALVE GUIDE TC COOLANT: TEST NO.160

DC ANALYSIS

B1 N(3,4), R=.073
 B2 N(4,5), R=.052
 B3 N(2,6), R=.024
 B4 N(6,7), R=.016
 B5 N(4,6), R=.42
 B6 N(6,8), R=.22
 B7 N(1,8), R=.21
 B8 N(8,9), R=.030
 B9 N(5,10), R=.28
 B10 N(7,10), R=.070
 B11 N(9,10), R=.070
 B12 N(1,0) R=.00001,E=18.
 B13 N(2,0), R=.00001,E=22.
 B14 N(3,0), R=.00001,E=36.
 B15 N(10,0), R=.00001
 TAB, NV(10), I(15)
 EXECUTE

NODES

NODE VOLTAGES

1- 7	1.79995+01	2.15979+01	3.59989+01	2.79459+01	2.35654+01	1.69625+01	1.38074+01
8- 10	8.43466+00	5.95536+00	3.65669+03				

BRANCHES

BRANCH CURRENTS

1- 7	1.10315+02	3.41635+01	2.09808+02	1.97196+02	2.61510+01	3.87630+01	4.55471+01
8- 14	8.43100+01	3.41635+01	1.97196+02	8.43100+01	-4.56095+01	-2.09904+02	-1.10388+02
15- 15	3.65669+02						

EXECUTION TIME = .413 SEC

TABLE A10-2 Cont. ELECTRICAL ANALOG FOR THE VALVE GUIDE TO COOLANT: TEST NO.163

DC ANALYSIS

B1 N(3,4), R=.073
 B2 N(4,5), R=.052
 B3 N(2,6), R=.024
 B4 N(6,7), R=.016
 B5 N(4,6), R=.42
 B6 N(6,8), R=.22
 B7 N(1,8), R=.21
 B8 N(8,9), R=.030
 B9 N(5,10), R=.28
 B10 N(7,10), R=.070
 B11 N(9,10), R=.070
 B12 N(1,0), R=.00001,E=18.
 B13 N(2,0), R=.00001,E=22.
 B14 N(3,0), R=.00001,E=37.
 B15 N(10,0), R=.00001
 TAB, NV(10), I(15)
 EXECUTE

NODES

NODE VOLTAGES

NODES	1- 7	8- 10	11- 13	14- 16	17- 19	20- 22	23- 25	26- 28
	1.79995+01	2.19979+01	3.69989+01	2.86671+01	2.41776+01	1.69916+01	1.38310+01	
	8.44151+00	5.91016+00	3.68247-03					

BRANCHES

BRANCH CURRENTS

BRANCHES	1- 7	8- 10	11- 13	14- 16	17- 19	20- 22	23- 25	26- 28
	1.14134+02	8.63355+01	2.08598+02	1.97533+02	2.77988+01	3.88638+01	4.55145+01	
	8.43783+01	8.63355+01	1.97533+02	8.43783+01	-4.55856+01	-2.08688+02	-1.14250+02	
	3.68247+02							

EXECUTION TIME = .431 SEC

*** END OF BATCH OF PROGRAMS ***

TOTAL EXECUTION TIME = 43.823 SEC

TABLE A1C-2 Cont. ELECTRICAL ANALOG FOR THE SODIUM-FILLED VALVE: TEST NO.34

DC ANALYSIS

B1	N(5,6),	R=.013
B2	N(10,11),	R=.096
B3	N(15,16),	R=.016
B4	N(4,5),	R=.016
B5	N(9,10),	R=.012
B6	N(14,15),	R=.021
B7	N(3,4),	R=1.7
B8	N(8,9),	R=1.3
B9	N(13,14),	R=2.2
B10	N(2,3),	R=.17
B11	N(7,8),	R=.12
B12	N(12,13),	R=.20
B13	N(1,2),	R=.28
B14	N(1,7),	R=.21
B15	N(1,12),	R=.36
B16	N(10,5),	R=.70
B17	N(15,10),	R=.68
B18	N(17,15),	R=.22
B19	N(18,19),	R=.073
B20	N(19,20),	R=.12
B21	N(21,22),	R=.20
B22	N(23,22),	R=.11
B23	N(24,23),	R=10.
B24	N(24,18),	R=16.
B25	N(24,21),	R=31.
B26	N(33,21),	R=2.2
B27	N(25,20),	R=.17
B28	N(26,25),	R=.14
B29	N(24,26),	R=9.9
B30	N(24,37),	R=7.2
B31	N(24,33),	R=20.
B32	N(24,34),	R=31.
B33	N(24,27),	R=10.
B34	N(24,17),	R=21.
B35	N(27,28),	R=.11
B36	N(34,28),	R=.20
B37	N(35,28),	R=.078
B38	N(28,29),	R=.039
B39	N(35,29),	R=.090
B40	N(29,0),	R=.031
B41	N(30,20),	R=.78
B42	N(31,30),	R=.59
B43	N(37,32),	R=.042
B44	N(38,32),	R=.042
B45	N(36,31),	R=1.9
B46	N(36,38),	R=.66
B47	N(36,35),	R=1.3
B48	N(32,33),	R=.20
B49	N(33,34),	R=2.2
B50	N(20,1),	R=.0001
B51	N(6,0),	R=1.E-3,E=19.
B52	N(11,0),	R=1.E-3,E=21.
B53	N(16,0),	R=1.E-3,E=30.
B54	N(24,0),	R=1.E-3,E=927.
P=F	N(22,0),	R=1.E-3,E=52.

TABLE A10-2 Cont.

856 N(36,0), R=1.E-3,E=713.
 857 N(22,28), R=1.4
 858 N(31,39), R=.57
 859 N(32,39), R=1.1
 860 N(39,25), R=1.5
 861 N(25,19), R=7.9
 862 N(19,12), R=8.2
 863 N(12,7), R=5.0
 864 N(7,2), R=5.5
 TAB, NV(39), I(64)
 EXECUTE

NODES

NODE VOLTAGES

1- 7	2.55357+02	2.25106+02	2.06686+02	2.24852+01	2.07516+01	1.91251+01	2.26826+02
8- 14	2.10540+02	3.41122+01	3.24837+01	2.11184+01	2.27649+02	2.11495+02	3.37955+01
15- 21	3.20993+01	3.01235+01	4.13725+01	2.63236+02	2.60209+02	2.55389+02	8.93086+01
22- 28	5.22654+01	6.17778+01	9.26540+02	2.81402+02	2.90398+02	5.30820+01	4.34740+01
29- 35	2.59731+01	3.54978+02	4.30308+02	4.96487+02	4.37367+02	8.12972+01	5.64220+01
36- 39	7.12040+02	4.98981+02	5.09383+02	4.18564+02			

BRANCHES

BRANCH CURRENTS

1- 7	1.25113+02	1.18388+02	1.23489+02	1.08353+02	1.35714+02	8.07723+01	1.08353+02
8- 14	1.35714+02	8.07723+01	1.08353+02	1.35714+02	8.07724+01	1.08040+02	1.35862+02
15- 21	7.69663+01	1.67601+01	-5.65262-01	4.21508+01	4.14565+01	4.01683+01	1.85216+02
22- 28	5.64762+01	8.64762+01	4.14565+01	2.70075+01	1.58208+02	1.53016+02	6.42568+01
29- 35	5.42568+01	5.93332+01	2.44587+01	2.72659+01	8.73458+01	4.21508+01	8.73458+01
36- 42	1.89116+02	1.66300+02	4.48741+02	3.38321+02	7.87063+02	1.27678+02	1.27678+02
43- 49	5.93833+01	3.07356+02	1.48280+02	3.07056+02	5.04322+02	2.95600+02	1.61850+02
50- 56	3.20654+02	1.25110+02	1.18385+02	1.23485+02	-4.59930+02	2.65408+02	-9.59747+02
57- 63	5.27958+00	2.06226+01	7.08390+01	9.14415+01	2.68259+00	3.97075+00	1.64636-01
64- 64	3.12780-01						

EXECUTION TIME = 2.339 SEC

TABLE A10-2 Ccnt. ELECTRICAL ANALOG FOR THE SODIUM-FILLED VALVE: TEST NO.35

DC ANALYSIS

B1	N(5,6),	R=.013
B2	N(10,11),	R=.096
B3	N(15,16),	R=.016
B4	N(4,5),	R=.016
B5	N(9,10),	R=.012
B6	N(14,15),	R=.021
B7	N(3,4),	R=1.7
B8	N(8,9),	R=1.3
B9	N(13,14),	R=2.2
B10	N(2,3),	R=.17
B11	N(7,8),	R=.12
B12	N(12,13),	R=.20
B13	N(1,2),	R=.28
B14	N(1,7),	R=.21
B15	N(1,12),	R=.36
B16	N(10,5),	R=.70
B17	N(15,10),	R=.68
B18	N(17,15),	R=.22
B19	N(18,19),	R=.073
B20	N(19,20),	R=.12
B21	N(21,22),	R=.20
B22	N(23,22),	R=.11
B23	N(24,23),	R=9.6
B24	N(24,18),	R=16.
B25	N(24,21),	R=30.
B26	N(33,21),	R=2.2
B27	N(25,20),	R=.17
B28	N(26,25),	R=.14
B29	N(24,26),	R=9.6
B30	N(24,37),	R=7.2
B31	N(24,33),	R=19.
B32	N(24,34),	R=30.
B33	N(24,27),	R=9.6
B34	N(24,17),	R=21.
B35	N(27,28),	R=.11
B36	N(34,28),	R=.20
B37	N(35,28),	R=.078
B38	N(28,29),	R=.039
B39	N(35,29),	R=.090
B40	N(29,0),	R=.033
B41	N(30,20),	R=.78
B42	N(31,30),	R=.59
B43	N(37,32),	R=.042
B44	N(38,32),	R=.042
B45	N(36,31),	R=1.9
B46	N(36,38),	R=.66
B47	N(36,35),	R=1.3
B48	N(32,33),	R=.20
B49	N(33,34),	R=2.2
B50	N(20,1),	R=.0001
B51	N(6,0),	R=.001, E=21.
B52	N(11,0),	R=.001, E=24.
B53	N(16,0),	R=.001, E=35.
B54	N(24,0),	R=.001, E=1082.
B	N(22,0),	R=.001, E=61.

TABLE A10-2 Cont.

B56 N(36,0), R=.001,E=845.
 B57 N(22,28), R=1.4
 B58 N(31,39), R=.57
 B59 N(32,39), R=1.1
 B60 N(39,25), R=1.5
 B61 N(25,19), R=7.9
 B62 N(19,12), R=8.2
 B63 N(12,7), R=5.0
 B64 N(7,2), R=5.5
 TAE, NV(39), I(64)
 EXECUTE

NODES

NODE VOLTAGES

1- 7	1.01656+02	2.65739+02	2.43867+02	2.51494+01	2.30909+01	2.11493+01	2.67849+02
8- 14	2.48552+02	3.95048+01	3.75752+01	2.41399+01	2.68862+02	2.49746+02	3.94712+01
15- 21	3.74640+01	3.51449+01	4.82876+01	3.10812+02	3.07296+02	3.01694+02	1.05321+02
22- 28	6.13182+01	7.28748+01	1.08145+03	3.32654+02	3.43417+02	6.33252+01	5.16592+01
29- 35	3.08507+01	4.19898+02	5.09309+02	5.87626+02	5.17769+02	9.65206+01	6.69696+01
36- 39	8.43861+02	5.90490+02	6.02956+02	4.95324+02			

BRANCHES

BRANCH CURRENTS

1- 7	1.49349+02	1.39550+02	1.44941+02	1.28658+02	1.60806+02	9.55794+01	1.28658+02
8- 14	1.60806+02	9.55794+01	1.28658+02	1.60806+02	9.55794+01	1.28274+02	1.60987+02
15- 21	5.10949+01	2.06518+01	-1.63486-01	4.91983+01	4.81649+01	4.66875+01	2.20014+02
22- 28	1.05060+02	1.05060+02	4.81649+01	3.25377+01	1.87476+02	1.82115+02	7.68787+01
29- 35	7.68786+01	6.81891+01	2.96675+01	3.28310+01	1.06055+02	4.91983+01	1.06055+02
36- 42	2.24307+02	1.96288+02	5.33549+02	4.01321+02	9.34870+02	1.51544+02	1.51544+02
43- 49	6.81893+01	3.65008+02	1.76080+02	3.65008+02	5.97609+02	3.49285+02	1.91476+02
50- 56	3.80363+02	1.49346+02	1.39947+02	1.44937+02	-5.48721+02	3.18168+02	-1.13879+03
57- 63	6.89930+00	2.45359+01	8.39110+01	1.08447+02	3.20976+00	4.68716+00	2.02605-01
64- 64	3.83558-01						

EXECUTION TIME = 3.045 SEC

TABLE A10-2 Cont. ELECTRICAL ANALOG FOR THE SODIUM-FILLED VALVE: TEST NO.36

DC ANALYSIS

B1 N(5,6), R=.012
 B2 N(10,11), R=.096
 B3 N(15,16), R=.016
 B4 N(4,5), R=.016
 B5 N(9,10), R=.012
 B6 N(14,15), R=.021
 B7 N(3,4), R=1.7
 B8 N(8,9), R=1.3
 B9 N(13,14), R=2.2
 B10 N(2,3), R=.17
 B11 N(7,8), R=.12
 B12 N(12,13), R=.20
 B13 N(1,2), R=.28
 B14 N(1,7), R=.21
 B15 N(1,12), R=.36
 B16 N(10,5), R=.70
 B17 N(15,10), R=.68
 B18 N(17,15), R=.22
 B19 N(18,19), R=.073
 B20 N(19,20), R=.12
 B21 N(21,22), R=.20
 B22 N(23,22), R=.11
 B23 N(24,23), R=9.6
 B24 N(24,18), R=16.
 B25 N(24,21), R=30.
 B26 N(33,21), R=2.2
 B27 N(25,20), R=.17
 B28 N(26,25), R=.14
 B29 N(24,26), R=9.6
 B30 N(24,27), R=7.2
 B31 N(24,23), R=19.
 B32 N(24,24), R=30.
 B33 N(24,27), R=9.6
 B34 N(24,17), R=21.
 B35 N(27,28), R=.11
 B36 N(34,28), R=.20
 B37 N(35,28), R=.078
 B38 N(28,29), R=.039
 B39 N(35,29), R=.090
 B40 N(29,0), R=.033
 B41 N(30,20), R=.78
 B42 N(31,30), R=.59
 B43 N(37,32), R=.042
 B44 N(38,32), R=.042
 B45 N(36,31), R=1.9
 B46 N(36,38), R=.66
 B47 N(36,35), R=1.3
 B48 N(32,33), R=.20
 B49 N(33,34), R=2.2
 B50 N(20,1), R=.0C01
 B51 N(6,0), R=.0C1,E=22.
 B52 N(11,0), R=.0C1,E=24.
 B53 N(16,0), R=.0C1,E=37.
 B=" N(24,0), R=.0C1,E=1125.
 B N(22,0), R=.0C1,E=65.

TABLE A10-2 Cont.

B56 N(36,0), R=1.0C1.E=907.
 B57 N(22,28), R=1.4
 B58 N(31,39), R=1.57
 B59 N(32,39), R=1.1
 B60 N(39,25), R=1.5
 B61 N(25,19), R=7.9
 B62 N(19,12), R=8.2
 B63 N(12,7), R=5.0
 B64 N(7,2), R=5.5
 TAB, NV(39), I(64)
 EXECUTE

NODES

NODE VOLTAGES

1- 7	3.20056+02	2.81906+02	2.58677+02	2.63896+01	2.42034+01	2.21574+01	2.84047+02
8- 14	2.63490+02	4.07816+01	3.87259+01	2.41518+01	2.85227+02	2.64933+02	4.17037+01
15- 21	3.95729+01	3.71513+01	5.08203+01	3.29501+02	3.25874+02	3.20096+02	1.12137+02
22- 28	6.53357+01	7.73338+01	1.12443+03	3.52890+02	3.63980+02	6.71878+01	5.50735+01
29- 35	3.29240+01	4.47551+02	5.43960+02	6.27654+02	5.52717+02	1.02787+02	7.16024+01
36- 39	9.05772+02	6.30535+02	6.44294+02	5.28557+02			

BRANCHES

BRANCH CURRENTS

1- 7	1.57386+02	1.51813+02	1.51347+02	1.36640+02	1.71314+02	1.01468+02	1.36640+02
8- 14	1.71314+02	1.01468+02	1.36640+02	1.71314+02	1.01468+02	1.36250+02	1.71468+02
15- 21	9.67469+01	2.07464+01	1.24561+00	5.11245+01	4.96833+01	4.81462+01	2.34007+02
22- 28	1.09073+02	1.09073+02	4.96834+01	3.37432+01	2.00264+02	1.92905+02	7.92140+01
29- 35	7.92140+01	6.85971+01	3.00904+01	3.40549+01	1.10130+02	5.11245+01	1.10130+02
36- 42	2.38563+02	2.11909+02	5.67937+02	4.29760+02	9.97697+02	1.63404+02	1.63404+02
43- 49	6.85973+01	3.96178+02	1.90427+02	3.96179+02	6.41669+02	3.74687+02	2.04513+02
50- 56	4.04434+02	1.57383+02	1.51810+02	1.51342+02	-5.65842+02	3.35742+02	-1.22836+03
57- 63	7.33015+00	2.70228+01	9.00885+01	1.17111+02	3.41979+00	4.95694+00	2.35863-01
64- 64	3.89428-01						

EXECUTION TIME = 2.974 SEC

TABLE A10-2 Cont. ELECTRICAL ANALOG FOR THE LITHIUM-FILLED VALVE: TEST NO.158

DC ANALYSIS		
B1	N(5.6),	R=.013
B2	N(10,11),	R=.096
B3	N(15,16),	R=.016
B4	N(4,5),	R=.016
B5	N(9,10),	R=.012
B6	N(14,15),	R=.021
B7	N(3,4),	R=1.7
B8	N(8,9),	R=1.3
B9	N(13,14),	R=2.2
B10	N(2,3),	R=.17
B11	N(7,8),	R=.12
B12	N(12,13),	R=.20
B13	N(1,2),	R=.44
B14	N(1,7),	R=.33
B15	N(1,12),	R=.56
B16	N(10,5),	R=.70
B17	N(15,10),	R=.68
B18	N(17,15),	R=.22
B19	N(18,19),	R=.073
B20	N(19,20),	R=.18
B21	N(21,22),	R=.20
B22	N(23,22),	R=.11
B23	N(24,23),	R=10.
B24	N(24,18),	R=16.
B25	N(24,21),	R=31.
B26	N(33,21),	R=2.2
B27	N(25,20),	R=.27
B28	N(26,25),	R=.14
B29	N(24,26),	R=9.9
B30	N(24,37),	R=7.2
B31	N(24,33),	R=20.
B32	N(24,34),	R=31.
B33	N(24,27),	R=10.
B34	N(24,17),	R=21.
B35	N(27,28),	R=.11
B36	N(34,28),	R=.20
B37	N(35,28),	R=.078
B38	N(28,29),	R=.039
B39	N(35,29),	R=.090
B40	N(29,01),	R=.033
B41	N(30,20),	R=1.2
B42	N(31,30),	R=.59
B43	N(37,32),	R=.042
B44	N(38,32),	R=.042
B45	N(36,31),	R=1.9
B46	N(36,38),	R=.66
B47	N(36,35),	R=1.3
B48	N(32,33),	R=.20
B49	N(33,34),	R=2.2
B50	N(20,1),	R=.0001
B51	N(6,0),	R=.001,E=18.
B52	N(11,0),	R=.001,E=19.
B53	N(16,0),	R=.001,E=31.
	N(24,0),	R=.001,E=930.
	N(22,0),	R=.001,E=62.

TABLE A10-2 Cont.

B56 N(36,0), R=.001,E=713.
 B57 N(22,28), R=1.4
 B53 N(31,39), R=.57
 B59 N(32,39), R=1.1
 B60 N(39,25), R=1.3
 B61 N(25,19), R=7.9
 B62 N(19,12), R=8.2
 B63 N(12,7), R=5.3
 B64 N(7,2), R=5.5
 TAB, NV(39), I(54)
 EXECUTE

NODES

NODE VOLTAGES

1- 7	2.54958+02	2.10242+02	1.92885+02	1.93110+01	1.76773+01	1.61198+01	2.12858+02
8- 14	1.97541+02	3.16034+01	3.00717+01	1.91141+01	2.15586+02	2.00496+02	3.45138+01
15- 21	3.29295+01	3.11135+01	4.22252+01	2.65231+02	2.62200+02	2.54988+02	9.88785+01
22- 28	6.22559+01	7.16922+01	9.29542+02	2.95648+02	3.04487+02	5.36777+01	4.40432+01
29- 35	2.62613+01	3.85599+02	4.49816+02	5.01945+02	4.42777+02	8.22817+01	5.68477+01
36- 39	7.12059+02	5.04425+02	5.14516+02	4.33183+02			

BRANCHES

BRANCH CURRENTS

1- 7	1.19808+02	1.14141+02	1.13497+02	1.02102+02	1.27644+02	7.54466+01	1.02102+02
8- 14	1.27644+02	7.54466+01	1.02102+02	1.27644+02	7.54466+01	1.01627+02	1.27575+02
15- 21	7.03075+01	1.77062+01	4.20264+00	4.22532+01	4.15194+01	4.00686+01	1.83113+02
22- 28	8.57850+01	8.57850+01	4.15194+01	2.67956+01	1.56317+02	1.50593+02	6.31369+01
29- 35	6.31369+01	5.90441+01	2.43383+01	2.73310+01	8.75864+01	4.22532+01	8.75864+01
36- 42	1.91192+02	1.64160+02	4.55947+02	3.39849+02	7.95795+02	1.08842+02	1.08842+02
43- 49	5.90443+01	2.99308+02	1.38022+02	2.99307+02	5.04008+02	2.95840+02	1.63861+02
50- 56	2.99473+02	1.19807+02	1.14138+02	1.13494+02	-4.57916+02	2.55883+02	-9.41422+02
57- 63	1.30090+01	2.91799+01	6.25104+01	9.16902+01	4.23390+00	5.68469+00	5.45483-01
54- 64	4.75660-01						

EXECUTION TIME = 2.952 SEC

TABLE A10-2 Cont. ELECTRICAL ANALOG FOR THE LITHIUM-FILLED VALVE: TEST NO.160

DC ANALYSIS

B1	N(5,6),	R=.013
B2	N(10,11),	R=.096
B3	N(15,16),	R=.016
B4	N(4,5),	R=.016
B5	N(9,10),	R=.012
B6	N(14,15),	R=.021
B7	N(3,4),	R=1.7
B8	N(8,9),	R=1.3
B9	N(13,14),	R=2.2
B10	N(2,3),	R=.17
B11	N(7,8),	R=.12
B12	N(12,13),	R=.20
B13	N(1,2),	R=.44
B14	N(1,7),	R=.33
B15	N(1,12),	R=.56
B16	N(10,5),	R=.70
B17	N(15,10),	R=.58
B18	N(17,15),	R=.22
B19	N(18,19),	R=.073
B20	N(19,20),	R=.18
B21	N(21,22),	R=.20
B22	N(23,22),	R=.11
B23	N(24,23),	R=9.6
B24	N(24,18),	R=16.
B25	N(24,21),	R=30.
B26	N(33,21),	R=2.2
B27	N(25,20),	R=.27
B28	N(26,25),	R=.14
B29	N(24,26),	R=9.6
B30	N(24,37),	R=7.2
B31	N(24,33),	R=19.
B32	N(24,34),	R=30.
B33	N(24,27),	R=9.6
B34	N(24,17),	R=21.
B35	N(27,28),	R=.11
B36	N(34,28),	R=.20
B37	N(35,28),	R=.078
B38	N(28,29),	R=.039
B39	N(35,29),	R=.090
B40	N(29,0),	R=.033
B41	N(30,20),	R=1.2
B42	N(31,30),	R=.59
B43	N(37,32),	R=.042
B44	N(38,32),	R=.042
B45	N(36,31),	R=1.9
B46	N(36,38),	R=.66
B47	N(36,35),	R=1.3
B48	N(32,33),	R=.20
B49	N(33,34),	R=2.3
B50	N(20,1),	R=.0001
B51	N(6,0),	R=.001, E=18.
B52	N(11,0),	R=.001, E=22.
B53	N(16,0),	R=.001, E=36.
B54	N(24,0),	R=.001, E=1073.
B55	N(22,0),	R=.001, E=72.

TABLE A10-2 Cont.

B56 N(36,0), R=.001,E=845.
 B57 N(22,28), R=1.4
 B58 N(31,39), R=.57
 B59 N(32,39), R=1.1
 B60 N(39,25), R=1.5
 B61 N(25,19), R=7.9
 B62 N(19,12), R=8.2
 B63 N(12,7), R=5.0
 B64 N(7,2), R=5.5
 TAB, NV(39), I(64)
 EXECUTE

NODES

NODE VOLTAGES

1- 7	3.00616+02	2.47736+02	2.27208+02	2.19230+01	1.99909+01	1.81422+01	2.50889+02
3- 14	2.32798+02	3.68160+01	3.50069+01	2.21341+01	2.54082+02	2.36253+02	4.01346+01
15- 21	3.82626+01	3.61331+01	4.89847+01	3.12378+02	3.08910+02	3.00652+02	1.15988+02
22- 28	7.23068+01	8.36370+01	1.07246+03	3.48867+02	3.59268+02	6.33649+01	5.18023+01
29- 35	3.09233+01	4.56067+02	5.32479+02	5.95333+02	5.26336+02	9.57554+01	6.70773+01
36- 39	8.43884+02	5.98100+02	6.10204+02	5.12880+02			

BRANCHES

BRANCH CURRENTS

1- 7	1.42207+02	1.34092+02	1.33094+02	1.20756+02	1.50756+02	8.91447+01	1.20756+02
3- 14	1.50755+02	8.91448+01	1.20756+02	1.50756+02	8.91448+01	1.20183+02	1.50690+02
15- 21	8.30972+01	2.14514+01	4.78778+00	4.87370+01	4.75053+01	4.58769+01	2.18404+02
22- 28	1.03003+02	1.03003+02	4.75053+01	3.18825+01	1.86522+02	1.78575+02	7.42911+01
29- 35	7.42911+01	6.58836+01	2.87435+01	3.25569+01	1.05114+02	4.87370+01	1.05114+02
36- 42	2.19766+02	1.95833+02	5.35359+02	4.01711+02	9.37070+02	1.29513+02	1.29513+02
43- 49	6.58835+01	3.54061+02	1.63897+02	3.54062+02	5.97544+02	3.44987+02	1.87209+02
50- 56	3.53928+02	1.42205+02	1.34089+02	1.33090+02	-5.37872+02	3.06754+02	-1.11560+03
57- 63	1.46460+01	3.43847+01	7.49574+01	1.09342+02	5.05790+00	6.68630+00	6.38660-01
64- 64	5.73201-01						

EXECUTION TIME = 2.392 SEC

TABLE A10-2 Cont. ELECTRICAL ANALOG FOR THE LITHIUM-FILLED VALVE: TEST NO. 163

DC ANALYSIS

B1	N(5,6),	R=.013
B2	N(10,11),	R=.096
B3	N(15,16),	R=.016
B4	N(4,5),	R=.016
B5	N(9,10),	R=.012
B6	N(14,15),	R=.021
B7	N(3,4),	R=1.7
B8	N(8,9),	R=1.3
B9	N(13,14),	R=2.2
B10	N(2,3),	R=.17
B11	N(7,8),	R=.12
B12	N(12,13),	R=.20
B13	N(1,2),	R=.44
B14	N(1,7),	R=.33
B15	N(1,12),	R=.56
B16	N(10,5),	R=.70
B17	N(15,10),	R=.68
B18	N(17,15),	R=.22
B19	N(18,19),	R=.073
B20	N(19,20),	R=.18
B21	N(21,22),	R=.20
B22	N(23,22),	R=.11
B23	N(24,23),	R=9.6
B24	N(24,18),	R=16.
B25	N(24,21),	R=30.
B26	N(33,21),	R=2.2
B27	N(25,20),	R=.27
B28	N(26,25),	R=.14
B29	N(24,26),	R=9.6
B30	N(24,37),	R=7.2
B31	N(24,33),	R=19.
B32	N(24,34),	R=30.
B33	N(24,27),	R=9.6
B34	N(24,17),	R=21.
B35	N(27,28),	R=.11
B36	N(34,28),	R=.20
B37	N(35,28),	R=.078
B38	N(28,29),	R=.039
B39	N(35,29),	R=.090
B40	N(29,0),	R=.033
B41	N(30,20),	R=1.2
B42	N(31,30),	R=.59
B43	N(37,32),	R=.042
B44	N(38,32),	R=.042
B45	N(36,31),	R=1.9
B46	N(36,38),	R=.66
B47	N(36,35),	R=1.3
B48	N(32,33),	R=.20
B49	N(33,34),	R=2.2
B50	N(20,1),	R=.0001
B51	N(6,0),	R=.001.E=18.
B52	N(11,0),	R=.001.E=22.
B53	N(16,0),	R=.001.E=37.
B54	N(24,0),	R=.001.E=1125.
---	N(22,0),	R=.001.E=77.

TABLE A10-2 Cont.

B56 N(36,0), R=.001,E=906.
 B57 N(22,28), R=1.4
 B58 N(31,39), R=.57
 B59 N(32,39), R=1.1
 B60 N(39,25), R=1.5
 B61 N(25,19), R=7.9
 B62 N(19,12), R=8.2
 B63 N(12,7), R=5.0
 B64 N(7,2), R=5.5
 TAB, NV(39), I(64)
 EXECUTE

NODES NODE VOLTAGES

1- 7	3.18642+02	2.62386+02	2.40548+02	2.21691+01	2.01138+01	1.81510+01	2.65706+02
8- 14	2.46447+02	3.78091+01	3.58832+01	2.21431+01	2.69103+02	2.50126+02	4.13870+01
15- 21	3.93945+01	3.71408+01	5.06437+01	3.30920+02	3.27300+02	3.18680+02	1.23548+02
22- 28	7.73235+01	8.91858+01	1.12444+03	3.69876+02	3.80722+02	6.77894+01	5.56820+01
29- 35	3.32265+01	4.85750+02	5.67892+02	6.33566+02	5.58624+02	1.03831+02	7.20239+01
36- 39	9.04796+02	6.36412+02	6.49793+02	5.46179+02			

BRANCHES BRANCH CURRENTS

1- 7	1.50986+02	1.43126+02	1.40851+02	1.28458+02	1.60491+02	9.48815+01	1.28458+02
8- 14	1.60490+02	9.48816+01	1.28458+02	1.60491+02	9.48816+01	1.27855+02	1.60415+02
15- 21	8.84638+01	2.25278+01	5.16356+00	5.11331+01	4.95949+01	4.78871+01	2.31125+02
22- 28	1.07839+02	1.07839+02	4.95949+01	3.33630+01	1.97762+02	1.89616+02	7.74705+01
29- 35	7.74705+01	6.77815+01	2.97797+01	3.40203+01	1.10068+02	5.11331+01	1.10068+02
36- 42	2.40744+02	2.09512+02	5.75782+02	4.31082+02	1.00686+03	1.39225+02	1.39225+02
43- 49	6.77815+01	3.86368+02	1.77318+02	3.86368+02	6.40594+02	3.74706+02	2.06724+02
50- 56	3.76740+02	1.50984+02	1.43123+02	1.40847+02	-5.61203+02	3.23498+02	-1.20438+03
57- 63	1.54582+01	3.80930+01	7.94422+01	1.17535+02	5.38945+00	7.09721+00	6.79417-01
64- 64	6.03516-01						

EXECUTION TIME = 2.421 SEC

TABLE A10-3 - COMPARISON OF RESULTS FOR EXHAUST VALVE ANALOGUE MODEL

Test No.	Valve head Temp. °F		Valve Seat Temp. °F		Heat Transfer from stem (BTU/hr)	Heat Transfer from seat (BTU/hr)	Heat Transfer to valve head (BTU/hr)	Heat Transfer to valve stem, seat and valve head, back side (BTU/hr)
	(31 ⁺)	(38 ⁺)	(22 ⁺)	(28 ⁺)				
34*	612	691	234	225	325	320	455	188
158**	632	697	244	225	305	320	437	188
35*	692	786	244	235	386	379	541	223
160**	715	793	255	235	361	374	518	217
36*	726	826	247	237	409	405	586	228
163**	751	833	260	239	334	405	563	225

+ - Corresponding Node in Figure A10-1

* - Sodium-filled valves

** - Lithium-filled valves

24 = Time-Average Exhaust-Gas Temperature
 36 = Time-Average Combustion-Chamber-Gas Temperature

6 = Gd c Temperature (See Figure 3)

7 = Gd b Temperature (See Figure 3)

8 = Gd a Temperature (See Figure 3)

22 = Seat Temperature (See Figure 3)

NOTE: NOT TO SCALE

* Underlined Numbers Are Nodes

** Non-Underlined Numbers Are Branches

Figure A10-1 - Schematic of Electrical Analogue Model of Heat Transfer in the Exhaust Valves.

/// = Zero Heat Flux Surface (Insulated)

1 = Gd c Temperature

2 = Gd b Temperature

3 = Gd a Temperature

0 = Coolant Temperature

NOTE: Not To Scale

* Underlined Numbers Are Nodes

** Non-Underlined Numbers Are Branches

Figure A10-2 - Schematic of Electrical Analogue Model of Heat Transfer from the Valve Guide to the Coolant.

APPENDIX 11 STEADY-STATE DIFFUSION (SWIRL) MODEL

A behavior of hydrogen alluded to in reports and at conferences as being an important characteristic is its high rate of diffusion in air. It is sometimes tacitly assumed that because of its high diffusion rate rapid mixing should take place in intake manifolds of carburetted engines alleviating cylinder-to-cylinder distribution problems. An order of magnitude analysis of the diffusion equation in the absence of pressure gradients, temperature gradients and force fields (Fick's law), as presented below, however, shows that the diffusion rate of hydrogen in air is of such a magnitude as to make it unimportant with respect to residence times actually encountered by the working fluid in intake manifolds of automotive engines.

The one dimensional form of Fick's law in terms of diffusion velocity, $C_H - C_A$, for a binary mixture is (Chapman and Cowling, 1964)

$$C_H - C_A = - \frac{D_{HA}}{u_H u_A} \frac{du_H}{dx}, \quad (1)$$

where D_{HA} is the coefficient of diffusion; u_H is the concentration of hydrogen; u_A is the concentration of air which is considered to be a single component gas and du_H/du_A is the concentration gradient. Consider a near stoichiometric mixture of hydrogen and air: $u_H=0.3$; $u_A=0.7$; du_H is of the order u_H and dx is of the order of the dimensions in an intake manifold: 1 ft., and $D_{HA} = 0.00458 \text{ ft}^2/\text{sec}$. For these values Fick's law yields diffusion velocities having the order of magnitude of 0.001 ft/sec. Thus, in an operating engine where gas velocities are of the order of 100 ft/sec molecular diffusion of hydrogen is of no real importance in intake manifolds.

An order of magnitude analysis similar to the above was performed to estimate the order of the diffusion velocity in the presence of a pressure gradient caused by induced swirl of the fresh charge and in the presence of temperature gradients resulting from the introduction of fresh charge into the hot cylinder. This analysis also showed negligibly small diffusion velocities. From these order of magnitude considerations, it

is apparent that induced swirl and/or temperature gradients will not cause homogeneous hydrogen-air mixtures to stratify. The analysis does show that if the intake process involves the induction of fresh air with induced swirl and if hydrogen is introduced in such a manner as to result in a stratified charge, turbulence and not diffusion will be the governing mechanism in causing the stratified charge to become homogeneous.

An order of magnitude analysis of the diffusion velocity in the presence of a force field, as would be found in the vicinity of the spark plug during ignition, has not been carried out. Such an analysis will be performed and discussed in another report.

One work area given in the proposal for this project is the formulation of a mathematical model which would give a first-order estimate of the hydrogen concentration distribution in the fresh charge. In the light of the above-described order of magnitude analysis it is evident that concentration distributions predicted by such a model will not be attained from a homogeneous mixture because of the relatively low diffusion velocities. For completeness, however, the analysis is presented below. Original plans were to include only the effect of swirling motion of the fresh charge on the distribution. Subsequent analysis has shown however, that at low swirl rates corresponding to low engine speeds, a nonuniform temperature similar in magnitude to the nonuniformity found during normal engine operation is much more effective in producing concentration gradients than the swirl. Because of this dominant influence of temperature nonuniformity on concentration distribution it has been incorporated into the analysis.

ANALYSIS

The kinetic theory of nonuniform gases (Chapman and Cowling, 1964) yields the following general equation of diffusion for a binary mixture (Assumption 1; the assumptions made are listed at the end of the analysis):

$$\bar{C}_1 - \bar{C}_2 = \frac{n^2}{n_1 n_2} D_{12} \left[\Delta u + \frac{u n_2 (m_2 - m_1)}{\rho} \Delta \ln p - \frac{\rho_1 \rho_2}{\rho p} (\underline{F}_1 - \underline{F}_2) + K_T \frac{1}{T} \Delta T \right] \quad (2)$$

where:

$\bar{C}_1 - \bar{C}_2$ = velocity of diffusion vector;

n_s = number density of molecule s ;

n = number density of molecules in mixture;

$$= n_1 + n_2; \quad (3)$$

D_{12} = coefficient of mutual diffusion of the binary mixture;

$$u = n_1 / (n_1 + n_2); \quad (4)$$

m_s = mass of molecule s ;

p = pressure;

ρ_s = partial density of constituent s ;

$$\rho = \rho_1 + \rho_2; \quad (5)$$

$\underline{F}_s$ = external force vector acting on molecule s divided by mass of molecule s ; i.e. external force vector on molecule s equals $m_s \underline{F}_s$;

K_T = thermal diffusion ratio;

T = temperature;

Δ = gradient operator.

Except for the region in the vicinity of the spark plug gap during ignition, the external force vector, $\underline{F}_s$, is zero. The electrostatic force field produced during ignition has no significant effect on the swirl; therefore,

$$\underline{F}_1 - \underline{F}_2 = 0. \quad (6)$$

The effect of the electrostatic force field during ignition on the fresh charge in the spark plug gap will be the subject of another report.

The gradient operator in cylindrical coordinates is:

$$\Delta = \underline{u}_r \frac{\partial}{\partial r} + \underline{u}_\theta \frac{1}{r} \frac{\partial}{\partial \theta} + \underline{u}_z \frac{\partial}{\partial z}; \quad (7)$$

where the unit vector $\underline{u}_r$ is in the radial direction, $\underline{u}_\theta$ in the tangential direction and $\underline{u}_z$ along the axis of the piston-cylinder. For axisymmetry (Assumption 2) and no variation with z (Assumption 3) Equation (7) becomes

$$\Delta = \underline{u}_r \frac{d}{dr}. \quad (8)$$

For steady state (Assumption 4)

$$\bar{C}_1 - \bar{C}_2 = 0 \quad (9)$$

Equations (2), (7), (8) and (9) yield

$$0 = \frac{du}{dr} + \frac{u n_2 (m_2 - m_1)}{p} \frac{d \ln p}{dr} + K_T \frac{1}{T} \frac{dT}{dr}. \quad (10)$$

To determine the pressure field consider the equation of motion for a Newtonian or Stokes fluid (Assumption 5) with constant viscosity (Assumption 6) (Prandtl and Tietgens, 1934)

$$\rho \left(\frac{\partial \underline{v}}{\partial t} + (\underline{v} \cdot \underline{\Delta}) \underline{v} \right) = \rho \underline{G} - \underline{\Delta} p + \frac{\mu}{3} \underline{\Delta} (\underline{\Delta} \cdot \underline{v}) + \mu \underline{\Delta}^2 \underline{v}; \quad (11)$$

Where:

$\underline{v}$ = bulk velocity vector

$$= u_r \underline{V}_r + u_\theta \underline{V}_\theta + u_z \underline{V}_z$$

$\underline{G}$ = external body force vector

$$= 0 \quad (12)$$

μ = viscosity.

In this first-order approximation analysis the swirling gas mixture is considered to be rotating with a velocity distribution

$$V_r = 0 \quad (13)$$

$$V_\theta = r\omega \quad (14)$$

$$V_z = 0, \quad (15)$$

where ω is the angular velocity of the swirling gas mixture (in this approximation the nonuniform gas rotates as a solid body).

Equations (8), (11), (12), (13), (14) and (15) and (Assumption 4) yields

$$\frac{dp}{dr} = - \rho \omega^2 r. \quad (16)$$

Using Equation (16), the following equation for density

$$p = n_1 m_1 + n_2 m_2, \quad (17)$$

the equation of state for a perfect gas mixture (Assumption 1)

$$p = kT(n_1 + n_2), \quad (18)$$

(where the Boltzmann constant (Present, 1958)

$$k = 1.3804 \times 10^{-16} \text{ erg/}^\circ\text{K}, \quad (19)$$

and the fact that

$$\frac{n_2}{n_1 + n_2} = 1 - u, \quad (20)$$

Equation (10) becomes

$$0 = \frac{du}{dr} + \frac{(m_2 - m_1)\omega^2}{kT} u(1-u)r + K_T \frac{1}{T} \frac{dT}{dr}. \quad (21)$$

In order to arrive at a first-order estimate of the influence of nonuniform charge temperature on the hydrogen concentration distribution, the temperature is assumed to vary according to the following equation:

$$T = T_0 + \Delta T (r/r_w)^2; \quad (22)$$

where T_0 is the temperature at center of the cylinder;

$$\Delta T = T_w - T_0; \quad (23)$$

and T_w is the temperature at the outer radius of the cylinder, r_w . Equation (22) is consistent with the assumption of axisymmetry and is of the same functional form as the solution to heat transfer in a long solid circular cylinder with uniformly distributed heat sinks and constant thermal conductivity. The Equation (22) temperature distribution is not the exact one for the binary perfect gas mixture. It is useful for the present analysis however, because it reduces the non-steady state heat transfer process in the fresh charge to a steady state process by the inclusion of heat sinks and should give the desired first-order effect of nonuniform temperature on the concentration field at steady state.

Substitution of Equation (22) into Equation (21) and expressing the thermal diffusion ratio, K_T , in terms of the thermal diffusion factor, α , (Chapman and Cowling, 1964)

$$K_T = \alpha u(1-u) \quad (24)$$

yields

$$-\frac{du}{u(1-u)} = r \frac{X dx}{T_0 + \Delta T X^2} \quad (25)$$

where

$$\Gamma = \frac{(m_2 - m_1) \omega^2 r_w^2}{k} + 2\alpha \Delta T \quad (26)$$

and

$$X = r/r_w \quad (27)$$

The relation between u and r is found by integration of Equation (16) from the center line of the cylinder where the hydrogen concentration is u_o to r . This integration yields:

$$\ln \left(\frac{1-u}{u} \cdot \frac{u_o}{1-u_o} \right) = \frac{\Gamma}{2\Delta T} \ln \left[1 + \frac{\Delta T}{T_o} \left(\frac{r}{r_w} \right)^2 \right] \quad (28)$$

To determine the effects of swirl alone, temperature nonuniformity alone, and the combined effects of swirl and nonuniform temperature on the hydrogen concentration distribution Equation (28) was solved for u as a function of r for the following three cases:

CASE 1 - Swirl with Uniform Temperature ($\omega \neq 0$, $\Delta T = 0$)

$$u = 1 + \frac{1-u_o}{u_o} \exp \left[\frac{\Gamma}{T_o} \frac{1}{2} \left(\frac{r}{r_w} \right)^2 \right]^{-1} \quad (29)$$

where

$$\Gamma_w = \frac{(m_2 - m_1) \omega^2 r_w^2}{k} \quad (30)$$

CASE 2 - Zero Swirl with Nonuniform Temperature ($\omega = 0$, $\Delta T \neq 0$)

$$u = \left\{ 1 + \frac{1-u_o}{u_o} \left[1 + \frac{\Delta T}{T_o} \left(\frac{r}{r_w} \right)^2 \right]^\alpha \right\}^{-1} \quad (31)$$

CASE 3 - Swirl with Nonuniform Temperature ($\omega \neq 0$, $\Delta T \neq 0$)

$$u = \left\{ 1 + \frac{1-u_o}{u_o} \left[1 + \frac{\Delta T}{T_o} \left(\frac{r}{r_w} \right)^2 \right]^{2\Delta T/\Gamma} \right\}^{-1} \quad (32)$$

ASSUMPTIONS AND APPROXIMATIONS

The swirling fresh charge is considered to have a cylindrical shape and the following assumption and approximations are incorporated in the model.

1. The charge is a binary perfect gas mixture
2. Axisymmetry
3. No variation with the axial dimension z
4. Steady state
5. The charge is a Newtonian or Stokes Fluid

6. Constant viscosity (this assumption is not necessary in the light of assumption 7).
7. The velocity distribution is the same as that of a solid rotating at an angular velocity ω .
8. The nonsteady state heat transfer process has been reduced to quasi-steady state by considering the gas to be composed of heat sinks and the temperature distribution for steady state heat transfer in a long cylinder of constant thermal conductivity is used for the temperature profile.

Numerical examples of the three cases are given below. In arriving at these examples, it was necessary to assume values for the angular velocity, ω , of the swirling charge. The angular velocity is related to the engine speed, N , by the swirl ratio (Huebner et al, 1972; Willis et al, 1966).

$$\Omega = \frac{\omega}{2\pi N} \quad (33)$$

In the numerical examples the swirl ratio is assumed to be independent of engine speed. This is a reasonable approximation for present purposes. The value of $\Omega=10$ used in the examples is attainable in a reasonably well-designed engine.

It was also necessary to assume a value for ΔT in the examples. The value chosen is 200°F . This is a reasonable, if not conservative, estimate of ΔT : Calculations assuming a fuel charge temperature of 80°F , a compression ratio of 10, and a pressure of about 150 psi after compression (Toyota specification) yield average charge temperatures on the order of 100°F , the exact value depending on the temperature and amount of residuals. Thus as a first-order approximation $T_o \approx 100^{\circ}\text{F}$. Since combustion chamber wall temperatures are on the order of 400°F , $T_w \approx 400^{\circ}\text{F}$. These estimates of T_o and T_w yield, according to Equation (23), $\Delta T = 400 - 100 = 300^{\circ}\text{F}$. Thus the value of $\Delta T = 200^{\circ}\text{F}$ used in the calculations is a reasonable, conservative estimate. The remainder of the values of the parameters used in the numerical examples are:

$$m_1 = \text{mass of hydrogen molecule} \\ = 0.332 \times 10^{-23} \text{ gm}$$

$$m_2 = \text{mass of "air" molecule} \\ = 4.81 \times 10^{-23} \text{ gm}$$

$$\alpha = 0.28$$

$$T = 300^\circ\text{K} \text{ (540}^\circ\text{R)}$$

The value of $r_w = 1.675$ in. is the radius of the Toyota 2TC engine to be used in this project.

In Table All the percent hydrogen in the swirling charge at steady state (100u) is given: at five values of radius (0.0, 0.4, 0.7, 1.0, 1.5, 1.675 in.); for three engine speeds (800 RPM or idle; 1800 RPM and 3000 RPM); and for four values of U_o (4.1% or lean limit; 10%; 20%; and 30%, which is near the stoichiometric value of 29.6). The Table All entries have been plotted in Figures All-1, All-2 and All-3. The results of calculations performed for an engine speed of 6000 RPM omitted from the table is included in Figure All-4 for a near stoichiometric mixture. This latter result corresponds to near peak power operation.

DISCUSSION

As discussed above, these results are presented for completeness. The resulting concentration distributions apply to steady state conditions which, as a result of the relatively short residence time of the mixture in a hydrogen engine are not obtainable if an initially homogeneous mixture is ingested by the engine. Non-homogeneous, stratified charges, however, can be realized by proper design.

TABLE All - Percent Hydrogen, 100 u, as a Function of Radius r, for the Three Cases [See Text: Case (1) $\Delta T=0$; $\omega \neq 0$; Case (2) $\omega=0$, $\Delta T \neq 0$; Case (3) $\Delta T \neq 0$, $\omega \neq 0$] for Various Values of Cylinder Centerline Hydrogen Concentration, u_o , and Engine Speed, RPM. (For all cases the swirl ratio, Ω , equals 10).

CASE	RPM	R A D I U S			-	I N C H E S			u_w-u_o
		0.0	0.4	0.7		1.0	1.5	1.675	
1	800	4.10	4.10	4.09	4.09	4.08	4.07	0.03	
2	800	4.10	4.06	3.98	3.87	3.65	3.57	0.53	
3	800	4.10	4.06	3.98	3.86	3.64	3.55	0.55	
1	1800	4.10	4.09	4.07	4.05	4.01	3.98	0.12	
2	1800	4.10	4.06	3.98	3.87	3.65	3.57	0.53	
3	1800	4.10	4.05	3.96	3.84	3.59	3.49	0.61	
1	3300	4.10	4.07	4.02	3.94	3.74	3.66	0.44	
2	3300	4.10	4.06	3.98	3.87	3.65	3.57	0.53	
3	3300	4.10	4.03	3.91	3.73	3.40	3.28	0.82	
1	800	10.00	10.00	9.99	9.98	9.95	9.94	0.06	
2	800	10.00	9.91	9.72	9.47	8.97	8.78	1.22	
3	800	10.00	9.90	9.72	9.46	8.93	8.74	1.26	
1	1800	10.00	9.98	9.95	9.90	9.78	9.73	0.27	
2	1800	10.00	9.91	9.72	9.47	8.97	8.78	1.22	
3	1800	10.00	9.98	9.68	9.36	8.81	8.60	1.40	
1	3300	10.00	9.94	9.82	9.63	9.18	8.99	1.01	
2	3300	10.00	9.91	9.72	9.47	8.97	8.78	1.22	
3	3300	10.00	9.85	9.56	9.16	8.37	8.09	1.91	
1	800	20.00	19.99	19.98	19.96	19.91	19.89	0.11	
2	800	20.00	19.83	19.51	19.06	18.15	17.81	2.19	
3	800	20.00	19.83	19.49	19.03	18.08	17.73	2.27	
1	1800	20.00	19.97	19.92	19.83	19.61	19.52	0.48	
2	1800	20.00	19.83	19.51	19.06	18.15	17.81	2.19	
3	1800	20.00	19.81	19.43	18.91	17.86	17.47	2.53	
1	3300	20.00	19.89	19.67	19.34	18.54	18.19	1.81	
2	3300	20.00	19.83	19.51	19.06	18.15	17.81	2.19	
3	3300	20.00	19.73	19.21	18.49	17.06	16.53	3.47	
1	800	30.00	29.99	29.97	29.95	29.88	29.85	0.15	
2	800	30.00	29.78	29.36	28.76	27.54	27.08	2.92	
3	800	30.00	29.77	29.33	28.72	27.45	26.98	3.02	
1	1800	30.00	29.96	29.89	29.78	29.50	29.37	0.63	
2	1800	30.00	29.78	29.36	28.76	27.54	27.08	2.92	
3	1800	30.00	29.74	29.25	28.56	27.16	26.63	3.37	
1	3300	30.00	29.86	29.57	29.13	28.06	27.60	2.40	
2	3300	30.00	29.78	29.36	28.76	27.54	27.08	2.92	
3	3300	30.00	29.64	28.96	28.00	26.07	25.35	4.65

Figure All-1 - Diffusion Model Equilibrium Hydrogen Concentration Distribution, $u_0 = 10\%$.

Figure All-2 - Diffusion Model Equilibrium Hydrogen Concentration Distribution, $u_o = 20\%$.

Figure All-3 - Diffusion Model Equilibrium Hydrogen Concentration Distribution, $u_o = 30\%$.

Figure All-4 - Diffusion Model Equilibrium Hydrogen Concentration Distribution at High Speed, High Load.

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APPENDIX 12 BISHOP/TAYLOR ITE FROM BTE COMPUTER PROGRAM
AND RESULTS

See Section 12 for a Description of this Appendix

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* Results are Plotted in Either Figure 39 or 40 as Indicated in the Tables.

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A12.1 DEFINITIONS OF FORTRAN SYMBOLS USED IN COMPUTER PROGRAM
AND RESULTS TABLES

<u>Fortran Symbol</u>	<u>Description</u>	<u>Units</u>
BALDIA	Diameter of the bearing in balance shafts	in.
BALLEN	Total length of balance shaft bearing divided by number of pistons	in./piston
BMEP	Brake Mean Effective Pressure at operating conditions	psi
BORE	Engine cylinder diameter	in.
BTE	Brake Thermal Efficiency at operating conditions	---
CCR	Correction factor to correct Indicated Thermal Efficiency to 9.0 to 1 compression ratio	---
CIDENG	Cubic inch displacement of the engine	cu in.
CRATIO	Compression ratio	---
IMEP	Indicated Mean Effective Pressure	psi
INTCYL	Number of intake valves per cylinder	---
INTDIA	Head diameter of intake valve	in.
ITE	Indicated Thermal Efficiency	---
ITECCR	Indicated Thermal Efficiency corrected to 9.0 to 1 compression ratio	---
MAIDIA	Diameter of main bearings	in.
MAILEN	Total length of all main bearings in inches divided by number of cylinders	in./cyl.
NCYL	Number of cylinders	---
PA	Atmospheric Pressure	psi
PE	Exhaust back pressure	psi
PHI	Fuel-Air Equivalence Ratio, ϕ , at operating condition	---
PI	Manifold vacuum	psi
PISLEN	Average length of the piston	in.
PISROD	Number of piston per connecting rod bearing	---
RINPIS	Number of rings on the piston	---
RODDIA	Diameter of the connecting rod bearing	in.
RODLEN	Length of connecting rod bearing	in.
RPM	Revolutions per minute	RPM
STROKE	Length of the stroke of the engine	in.
X	Indicated Mean Effective Pressure divided by Brake Mean Effective Pressure	---

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A12.2 BISHOP/TAYLOR ITE FROM BTE COMPUTER
PROGRAM

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A12.2

035322086*TPF\$(0).IMEP/PRGM SYM CREATED ON 27 JUL 78 AT 13:58:04

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1      DIMENSION BORE(10),CIDENG(10),PISLEN(10),CRATIO(10),STROKE(10),
2      1      RODDIA(10),RODLEN(10),PISROD(10),TESTNO(10),TESTNX(10),
3      2      RINPIS(10),PA(10),PE(10),P1(10),RPM(10),BMEP(10),
4      3      FMEP5(10),FMEP6(10),FMEP8(10),FMEP12(10),FMEP14(10),
5      4      FMEP15(10),FMEP17(10),FMEP18(10),FMEP19(10)
6      5      ,BALDIA(10),BALLEN(10),PHI(10),BTE(10),X(10),CCR(10)
7      REAL NCYL(10),INTCYL(10),INTDIA(10),MAIDIA(10),MAILEN(10),
8      1      IMEP(10),ITE(10),ITECCR(10)
9      ISTOP=0
10     1 DO 10 I=1,10
11         READ(5,100) TESTNO(I),BORE(I),NCYL(I),CIDENG(I),INTCYL(I),
12         1      INTDIA(I),PISLEN(I),CRATIO(I),STROKE(I),MAIDIA(I),
13         2      RODDIA(I),MAILEN(I),RODLEN(I),PISROD(I),RINPIS(I),PA(I)
14         IF(TESTNO(I).EQ.-1.) GO TO 999
15         IF(TESTNO(I).EQ.-2.) GO TO 20
16         READ(5,100) TESTNX(I),PE(I),P1(I),BALDIA(I),BALLEN(I)
17         1      ,RPM(I),PHI(I),BTE(I),BMEP(I)
18         IF(TESTNO(I).NE.TESTNX(I)) WRITE(6,111)
19     111 FORMAT(//1X,'DATA ERROR INCONSISTANT TESTNO')
20     N=I
21     C FMEP5 IS FMEP MISC. AND PUMPS
22         FMEP5(I)=.39*(RPM(I)/1000.）**1.5
23     C FMEP6 IS FMEP CAM BEARINGS
24         FMEP6(I)=(30.-(4.0*RPM(I)/1000.))*INTCYL(I)*(INTDIA(I)**1.75)/
25         1      (BORE(I)**2 * STROKE(I))
26     C FMEP8 IS FMEP BEARINGS
27         FMEP8(I)=(6.0*((MAIDIA(I)**2)*MAILEN(I)+RODDIA(I)**2
28         1      *RODLEN(I)/PISROD(I)+BALDIA(I)**2*BALLEN(I))
29         2      /BORE(I)**3)
30         2      *(BORE(I)/STROKE(I))*(RPM(I)/1000.)
31     C FMEP12 IS FMEP VALVE PUMPING
32         FMEP12(I)=1.30*(RPM(I)/1000.）**1.7
33         1      *(.0758/(INTCYL(I)*INTDIA(I)**2 *NCYL(I)/CIDENG(I))）**1.28
34     C FMEP14 IS FMEP THROTTLING
35         FMEP14(I)=(PE(I)/2.75)+P1(I)
36     C FMEP15 IS FMEP BLOWBY
37         FMEP15(I)=(SQRT((PA(I)-P1(I))/14.2)) *
38         1      (1.72*CRATIO(I)**.4-(.49+.015*CRATIO(I))*(RPM(I)/1000.))**1.185)
39     C FMEP17 IS FMEP VISCOUS DRAG ON PISTON
40         FMEP17(I)=(21.93*PISLEN(I)/(BORE(I)*STROKE(I)))
41         1      *(STROKE(I)*RPM(I)/6000.)
42     C FMEP18 IS FMEP STATIC RING TENSION
43         FMEP18(I)=(2.11*STROKE(I)*RINPIS(I))/BORE(I)**2
44     C FMEP19 IS FMEP DUE TO GAS PRESSURE ON RINGS
45         FMEP19(I)=((PA(I)-P1(I))/14.2)*((12.35*STROKE(I)/BORE(I)**2)
46         1      *(.088*CRATIO(I))+(.182*CRATIO(I))**(.133-(.121*STROKE(I)
47         2      *RPM(I)/6000.)))
48         FMEP19(I)=FMEP5(I)+FMEP6(I)+FMEP8(I)+FMEP12(I)+FMEP14(I)
49         1      +FMEP15(I)+FMEP17(I)+FMEP18(I)+FMEP19(I)
50         FMEPVT=FMEP19(I)*1.12
51         IMEP(I)=BMEP(I)+FMEPVT
52         X(I)=IMEP(I)/BMEP(I)
53         ITE(I)=BTE(I)*X(I)
54         CCR(I)=(1.0-(1./9.))**.4/(1.0-(1./CRATIO(I))**.4)
55         ITECCR(I)=ITE(I)*CCR(I)
56     10 CONTINUE

```

A12.2 Cont.

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57      GO TO 20
58      999 ISTOP=1
59      20 WRITE(6,1000) (TESTNO(K),K=1,N)
60      WRITE(6,1001) (BORE(K),K=1,N)
61      WRITE(6,1002) (NCYL(K),K=1,N)
62      WRITE(6,1003) (CIDENG(K),K=1,N)
63      WRITE(6,1004) (INTCYL(K),K=1,N)
64      WRITE(6,1005) (INTDIA(K),K=1,N)
65      WRITE(6,1006) (PISLEN(K),K=1,N)
66      WRITE(6,1007) (CRATIO(K),K=1,N)
67      WRITE(6,1008) (STROKE(K),K=1,N)
68      WRITE(6,1009) (MAIDIA(K),K=1,N)
69      WRITE(6,1010) (RODDIA(K),K=1,N)
70      WRITE(6,1011) (MAILEN(K),K=1,N)
71      WRITE(6,1012) (RODLEN(K),K=1,N)
72      WRITE(6,1013) (PISROD(K),K=1,N)
73      WRITE(6,1014) (RINPIS(K),K=1,N)
74      WRITE(6,1015) (PA(K),K=1,N)
75      WRITE(6,1019) (PE(K),K=1,N)
76      WRITE(6,1020) (P1(K),K=1,N)
77      WRITE(6,1021) (BALDIA(K),K=1,N)
78      WRITE(6,1022) (BALLEN(K),K=1,N)
79      WRITE(6,1016) (RPM(K),K=1,N)
80      WRITE(6,1017) (BMEP(K),K=1,N)
81      WRITE(6,1018) (IMEP(K),K=1,N)
82      WRITE(6,1023) (X(K),K=1,N)
83      WRITE(6,1024) (ITE(K),K=1,N)
84      WRITE(6,1025) (CCR(K),K=1,N)
85      WRITE(6,1028) (BTE(K),K=1,N)
86      WRITE(6,1026) (ITECCR(K),K=1,N)
87      WRITE(6,1027) (PHT(K),K=1,N)
88      IF(ISTOP.EQ.1) STOP
89      GO TO 1
90      1000 FORMAT(1H1,9HTESTNO      ,10(F6.0,1X))
91      1001 FORMAT(1X,9HBORE      ,10(F6.2,1X))
92      1002 FORMAT(1X,9HNCYL      ,10(F5.0,2X))
93      1003 FORMAT(1X,9HCIDENG      ,10(F6.0,1X))
94      1004 FORMAT(1X,9HINTCYL      ,10(F5.0,2X))
95      1005 FORMAT(1X,9HINTDIA      ,10(F6.2,1X))
96      1006 FORMAT(1X,9HPISLEN      ,10(F6.2,1X))
97      1007 FORMAT(1X,9HCRATIO      ,10(F5.1,2X))
98      1008 FORMAT(1X,9HSTROKE      ,10(F7.3))
99      1009 FORMAT(1X,9HMAIDIA      ,10(F5.1,2X))
100     1010 FORMAT(1X,9HRODDIA      ,10(F5.1,2X))
101     1011 FORMAT(1X,9HMAILEN      ,10(F7.3))
102     1012 FORMAT(1X,9HRODLEN      ,10(F7.3))
103     1013 FORMAT(1X,9HPISROD      ,10(F5.0,2X))
104     1014 FORMAT(1X,9HRINPIS      ,10(F5.0,2X))
105     1015 FORMAT(1X,9HPA      ,10(F6.1,1X))
106     1016 FORMAT(1X,9HRPM      ,10(F7.0))
107     1017 FORMAT(1X,9HBMEP      ,10(F6.1,1X))
108     1018 FORMAT(1X,9HIMEP      ,10(F6.1,1X))
109     1019 FORMAT(1X,9HPE      ,10(F6.1,1X))
110     1020 FORMAT(1X,9HP1      ,10(F6.1,1X))
111     1021 FORMAT(1X,9HBALDIA      ,10(F7.3))
112     1022 FORMAT(1X,9HBALLEN      ,10(F7.3))
113     1023 FORMAT(1X,9HX      ,10(F7.3))

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A12.2 Cont.

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114      1024 FORMAT(1X,9HITE      ,10(F6.1,1X))
115      1025 FORMAT(1X,9HCCR      ,10(F7.3))
116      1026 FORMAT(1X,9HITECCR   ,10(F6.1,1X))
117      1027 FORMAT(1X,9HPHI      ,10(F6.2,1X))
118      1028 FORMAT(1X,9HBTE      ,10(F6.1,1X))
119      100  FORMAT(16F5.0)
120      END
@XQT TPF$.IMEP/A

```


A12.3 ITE FROM BTE DATA OF OTHERS

DATA NO. REF*	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23
REF*	1	1	2	2	3	3	4	4	4	4	4	4	4	5	5	5	5	5	5	6	6	6	6
BOPE	4.00	4.00	3.50	3.50	3.50	3.50	4.32	4.32	4.32	4.32	4.32	4.32	4.25	4.32	4.32	4.32	4.32	4.32	4.25	3.25	3.25	3.25	3.25
ACYL	9.	6.	4.	4.	4.	4.	8.	8.	8.	8.	8.	8.	8.	8.	8.	8.	8.	8.	8.	1.	1.	1.	1.
CIDENC	350.	350.	140.	140.	140.	140.	440.	440.	440.	440.	440.	440.	454.	440.	440.	440.	440.	440.	454.	37.	37.	37.	37.
INTCYL	1.	1.	1.	1.	1.	1.	1.	1.	1.	1.	1.	1.	1.	1.	1.	1.	1.	1.	1.	1.	1.	1.	1.
INTDIA	1.94	1.94	1.63	1.63	1.63	1.63	2.08	2.08	2.08	2.08	2.08	2.08	2.06	2.08	2.08	2.08	2.08	2.08	2.06	1.38	1.38	1.38	1.38
PISLEN	2.15	2.15	2.12	2.12	2.12	2.12	2.27	2.27	2.27	2.27	2.27	2.27	2.10	2.27	2.27	2.27	2.27	2.27	2.10	5.74	5.74	5.74	5.74
GRATIC	8.5	8.5	6.0	6.0	6.0	6.0	12.0	12.0	12.0	12.0	12.0	12.0	12.9	12.0	12.0	12.0	12.0	12.0	12.9	10.0	10.0	10.0	10.0
STPGME	3.460	3.460	3.625	3.625	3.625	3.625	3.750	3.750	3.750	3.750	3.750	3.750	4.000	3.750	3.750	3.750	3.750	3.750	4.000	4.500	4.500	4.500	4.500
MAIDIA	2.4	2.4	2.3	2.3	2.3	2.3	2.7	2.7	2.7	2.7	2.7	2.7	2.7	2.7	2.7	2.7	2.7	2.7	2.7	3.0	3.0	3.0	3.0
PCDDIA	2.1	2.1	2.0	2.0	2.0	2.0	2.4	2.4	2.4	2.4	2.4	2.4	2.2	2.4	2.4	2.4	2.4	2.4	2.2	3.5	3.0	3.0	3.0
MAIDIA	.523	.523	.942	.942	.942	.942	.625	.625	.625	.625	.625	.625	.653	.625	.625	.625	.625	.625	.653	5.162	5.162	5.162	5.162
POBLEM	.797	.797	.807	.807	.807	.807	.927	.927	.927	.927	.927	.927	.847	.927	.927	.927	.927	.927	.847	1.943	1.943	1.943	1.943
PISQDC	1.	1.	1.	1.	1.	1.	1.	1.	1.	1.	1.	1.	1.	1.	1.	1.	1.	1.	1.	1.	1.	1.	1.
FINPIS	3.	3.	3.	3.	3.	3.	3.	3.	3.	3.	3.	3.	3.	3.	3.	3.	3.	3.	3.	5.	5.	5.	5.
PA	14.7	14.7	14.7	14.7	14.7	14.7	12.4	12.4	12.4	12.4	12.4	12.4	12.4	12.4	12.4	12.4	12.4	12.4	12.4	14.7	14.7	14.7	14.7
PE	.5	.5	.5	.5	.5	.5	.5	.5	.5	.5	.5	.5	.5	.5	.5	.5	.5	.5	.5	.5	.5	.5	.5
PI	.5	.5	.5	.5	.5	.5	.5	.5	.5	.5	.5	.5	.5	.5	.5	.5	.5	.5	.5	.5	.5	.5	.5
3ALDIA	.000	.000	.000	.000	.000	.000	.000	.000	.000	.000	.000	.000	.000	.000	.000	.000	.000	.000	.000	4.000	4.000	4.000	4.000
BALLEN	.000	.000	.000	.000	.000	.000	.000	.000	.000	.000	.000	.000	.000	.000	.000	.000	.000	.000	.000	2.000	2.000	2.000	2.000
PPP	1500.	1500.	2039.	2570.	3018.	2634.	2000.	2000.	2000.	2000.	2000.	2000.	2000.	3000.	3000.	3000.	3000.	3000.	3000.	1500.	1500.	1500.	1500.
9MEP	24.1	64.9	47.3	57.8	59.0	65.5	80.4	80.4	61.3	65.9	78.5	95.0	88.6	71.2	71.2	76.3	95.0	103.9	85.7	35.7	54.3	72.9	86.4
IMEP	41.5	82.3	70.3	85.0	90.1	95.0	101.8	101.8	82.7	87.3	99.9	116.4	110.3	99.9	99.9	104.7	123.7	132.6	114.7	86.3	104.9	123.5	137.0
X	1.721	1.263	1.487	1.471	1.527	1.450	1.267	1.267	1.350	1.325	1.273	1.226	1.245	1.404	1.404	1.376	1.303	1.277	1.339	2.416	1.931	1.693	1.555
ITE	39.6	35.3	32.9	39.3	40.5	39.1	45.0	45.0	44.1	43.5	41.8	39.2	47.3	45.6	45.6	43.1	43.0	39.6	50.9	42.8	40.6	37.9	34.6
CCP	1.017	1.017	1.035	1.035	1.035	1.035	.928	.928	.928	.928	.928	.928	.913	.928	.928	.928	.928	.928	.913	.972	.972	.972	.972
BTE	23.0	31.0	22.1	26.7	25.5	27.0	35.5	35.5	32.7	32.8	32.8	32.0	38.0	32.5	32.5	31.3	33.0	31.0	39.0	17.7	21.0	22.4	21.8
ITECCF	40.2	40.0	34.0	40.7	41.9	40.5	41.7	41.7	41.0	40.4	38.8	36.4	43.2	42.3	42.3	40.0	39.9	36.7	46.4	41.5	39.4	36.9	33.6
PHI	.30	.60	.63	.65	.60	.74	.62	.62	.45	.45	.60	.85	.73	.62	.62	.60	.85	1.00	.73	.45	.60	.80	1.05

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*REF	REFERENCE (See Section 15)	Plotted in Figure
1	Finegold, 1974	39
2	Thomas and Sorenson, 1976	39
3	Thomas and Sorenson, 1976	40
4	BEC, 1976	39
5	BEC, 1976	40
6	King and Rand, 1955	39

A12.4 ITE FROM BTE DATA OF PRESENT STUDY

TEST NO.	29	30	31	32	33	34	35	36	45	46	47	38	40	71	72-73
BORE	3.35	3.35	3.35	3.35	3.35	3.35	3.35	3.35	3.35	3.35	3.35	3.35	3.35	3.35	3.35
NCYL	4.	4.	4.	4.	4.	4.	4.	4.	4.	4.	4.	4.	4.	4.	4.
CIDENG	97.	97.	97.	97.	97.	97.	97.	97.	97.	97.	97.	97.	97.	97.	97.
INTCYL	1.	1.	1.	1.	1.	1.	1.	1.	1.	1.	1.	1.	1.	1.	1.
INTDIA	1.59	1.59	1.59	1.59	1.59	1.59	1.59	1.59	1.59	1.59	1.59	1.59	1.59	1.59	1.59
PISLEN	2.25	2.25	2.25	2.25	2.25	2.25	2.25	2.25	2.25	2.25	2.25	2.25	2.25	2.25	2.25
CRATIO	9.0	9.0	9.0	9.0	9.0	9.0	9.0	9.0	9.0	9.0	9.0	9.0	9.0	9.0	9.0
STROME	2.760	2.760	2.760	2.760	2.760	2.760	2.760	2.760	2.760	2.760	2.760	2.760	2.760	2.760	2.760
MAIDIA	2.3	2.3	2.3	2.3	2.3	2.3	2.3	2.3	2.3	2.3	2.3	2.3	2.3	2.3	2.3
RODDIA	1.9	1.9	1.9	1.9	1.9	1.9	1.9	1.9	1.9	1.9	1.9	1.9	1.9	1.9	1.9
MAILEN	1.240	1.240	1.240	1.240	1.240	1.240	1.240	1.240	1.240	1.240	1.240	1.240	1.240	1.240	1.240
RODLEN	1.040	1.040	1.040	1.040	1.040	1.040	1.040	1.040	1.040	1.040	1.040	1.040	1.040	1.040	1.040
PISROD	1.	1.	1.	1.	1.	1.	1.	1.	1.	1.	1.	1.	1.	1.	1.
RINPIS	3.	3.	3.	3.	3.	3.	3.	3.	3.	3.	3.	3.	3.	3.	3.
PA	14.7	14.7	14.7	14.7	14.7	14.7	14.7	14.7	14.7	14.7	14.7	14.7	14.7	14.7	14.7
PE	.5	.5	.5	.5	.5	.5	.5	.5	.5	.5	.5	.5	.5	.5	.5
P1	.5	.5	.5	.5	.5	.5	.5	.5	.5	.5	.5	.5	.5	.5	.5
BALGIA	.000	.000	.000	.000	.000	.000	.000	.000	.000	.000	.000	.000	.000	.000	.000
BALLEN	.000	.000	.000	.000	.000	.000	.000	.000	.000	.000	.000	.000	.000	.000	.000
RPM	1800.	1800.	1800.	1800.	1800.	1800.	1800.	1800.	1800.	1800.	1800.	3300.	3300.	3300.	3300.
BMEP	14.3	20.9	32.1	41.1	46.7	54.2	64.4	69.1	70.5	81.4	83.7	25.5	47.5	25.5	84.0
IMEP	37.1	43.7	54.9	63.9	69.5	77.0	87.2	91.9	93.3	104.2	106.5	59.9	81.9	59.9	118.4
X	2.592	2.089	1.709	1.554	1.487	1.420	1.353	1.329	1.323	1.280	1.272	2.351	1.725	2.351	1.410
ITE	35.0	40.3	41.2	41.3	41.4	40.5	39.5	39.4	38.8	35.8	35.4	46.5	46.4	46.1	36.2
CCR	1.000	1.000	1.000	1.000	1.000	1.000	1.000	1.000	1.000	1.000	1.000	1.000	1.000	1.000	1.000
BTE	13.5	19.3	24.1	26.6	27.8	28.5	29.2	29.6	29.3	28.0	27.8	19.8	26.9	19.6	25.7
ITECCR	35.0	40.3	41.2	41.3	41.4	40.5	39.5	39.4	38.8	35.8	35.4	46.5	46.4	46.1	36.2
PHI	.24	.26	.32	.37	.41	.48	.58	.62	.73	.97	1.01	.29	.43	.30	1.06
PLOTTED IN FIGURE	39	39	39	39	39	39	39	39	39	39	39	40	40	40	40