

DA-44-009-AMC-824(T)

30 September 1964 through 31 July 1966

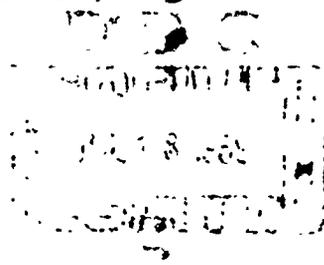
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Final Technical Report

Development of an Ammonia-Burning Gas Turbine Engine

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**placed by
U. S. Army Engineer Research and Development Laboratories
Fort Belvoir, Virginia**



CLEARINGHOUSE

H. SOLAR

DIVISION OF INTERNATIONAL HARVESTER COMPANY
2200 PACIFIC HIGHWAY SAN DIEGO, CALIFORNIA 92112

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3 April 1968
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PREFACE

This report covers work conducted by Solar Division of International Harvester Company in investigating the overall practicality of utilizing ammonia as a fuel in gas turbine engines. The work was authorized by Contract DA-44-009-AMC-824(T); and was administered by Research and Development Procurement Office, U. S. Army Engineer Research and Development Laboratories, Fort Belvoir, Virginia. The contract comprised two consecutive phases: Phase I, which covered the period 30 September 1964 to 28 June 1965; and Phase II, which extended to 31 July 1966. The work completed under Phase I was presented in a Final Report on Phase I - Combustion System Investigation, Development of an Ammonia-Burning Gas Turbine Engine, 23 July 1965. This final report summarizes the work covered in the Phase I report and contains details of the work completed during Phase II.

The report is submitted in response to the requirements of Contract item 2C.

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SUMMARY

Combustion rig testing with ammonia fuel was conducted using a single can burner configuration. Ammonia was injected into the burner in the liquid state, a combination of vapor and liquid, and in the vapor state. Vaporized ammonia injection gave the most encouraging preliminary results; consequently systems using liquid ammonia injection were abandoned early in the program.

Ammonia vapor combustor tests showed it was possible to burn ammonia in a manner similar to that used in conventional hydrocarbon burning combustors, but with a significant reduction in range of flammability. Improvements in combustion performance were obtained by the use of catalytic aids, and techniques using catalytic oxidation of ammonia showed the greatest potential. Two different ammonia combustion systems were developed for use with a gas turbine engine in the 250 hp size range. One system was essentially similar to a conventional hydrocarbon, single-can combustor, but of increased size. The second system incorporated an oxidizing catalyst bed integral with the combustor as a means of increasing fuel reaction rate and reducing combustor volume.

A standard hydrocarbon burning engine was modified to enable operation using both types of ammonia combustion systems in addition to its normal hydrocarbon system. A development test program was conducted to obtain satisfactory engine operation with each type of combustion system. Performance measurements were made with the engine operating with each type of combustion system. The results are plotted and allow comparison between ammonia fuel and hydrocarbon fuel engine performance.)

INTRODUCTION

The work described in this report was in support of the U. S. Army's Nuclear Powered Energy Depot (NPED) concept evaluation. The object of the program was to investigate and demonstrate the practicality of using ammonia as a fuel in gas turbine engines. This was accomplished in two consecutive phases of work as follows:

- Phase I encompassed a broad range of tests and analytical studies to determine both the feasibility and practicality of meeting the objectives of the program. The work was broadly directed toward all Army gas turbine engines, exclusive of aircraft. Major areas of investigation include combustion systems, control and accessories, and materials. Complete details of this work were presented in the Phase I Final Report and are not included in this report except to the extent necessary to provide a basic understanding of the work completed within Phase II.
- Phase II consisted of modifying a 250 hp Solar T-350 gas turbine engine to operate on ammonia fuel or hydrocarbon with minimum change-over effort. Development of the engine and its attendant accessory systems to provide satisfactory performance and operation was also included, as was calibration of engine performance when operating with two different types of ammonia combustors and the standard engine hydrocarbon burning combustor.

During the Phase I study, it was determined that the engine investigation should be directed toward developing both an ammonia vapor and an oxidizing catalyst combustion system, which represented a significant increase in the program work scope. The initial requirement was that the ammonia burning engine be subjected to a full range of environmental tests and a 500-hour endurance test. To meet specification requirements, it would have been necessary to develop a completely self-contained engine package. Since both ammonia combustion systems considered for use on the engine utilized ammonia vapor rather than liquid fuel, this dictated an additional requirement for an exhaust heat ammonia vaporizing system to process liquid ammonia to the dry vapor state. A conceptual design study of an ammonia exhaust heat vaporizer and its controls was completed and the results included in the Phase I report. To enable the program to include development of the two different ammonia combustion systems without an increase in the overall work scope, the contract was modified to eliminate the environmental and endurance tests and the need for an engine exhaust heat ammonia vaporizing system. Ammonia vapor necessary to operate the engine would be provided from an external source.

1

SUMMARY OF PHASE I AMMONIA COMBUSTION TESTS

1.1 AMMONIA CHARACTERISTICS

Ammonia is volatile and toxic. At atmospheric pressure, it boils at minus 28°F; at normal ambient temperatures, it is stored as a liquid at pressures under 200 psi.

Previous research shows that ammonia is difficult to burn, as is evident when compared with a hydrocarbon having excellent combustion properties in a gas turbine (Table I). Because of the relatively low rate of flame propagation and flammability, a larger combustion space is indicated for ammonia than for a hydrocarbon. Alternatively, a catalyst might be used to speed up the combustion process. As stoichiometric combustion (provision of the correct amount of air for complete reaction) results in approximately the same heat release per pound of air for both fuels, the air distribution in a hydrocarbon combustor will be very similar to that of an ammonia combustor.

TABLE I
AMMONIA VERSUS PROPANE COMBUSTION CHARACTERISTICS

	Ammonia	Propane
Flammability Range	1.8:1	4:1
Relative Flame Speed	1.0	38:0
Lower Heating Value Liquid (Btu/lb)	7410	19,800
Auto Ignition Temperature (°F)	1204	898
Vapor Pressure at 100°F (psi)	212	190
Stoichiometric Air/Fuel (lb/lb)	6.06:1	17.6:1
Heat Release/pound of Stoichiometric Air (Btu)	1220	1130
Heat of Vaporization at minus 28°F (Btu/lb)	590	180

The high volatility of ammonia requires special provisions for metering and injection. To meter through the control system as a liquid necessitates maintaining sufficient back pressure to prevent boiling within the system.

Vaporization of the liquid ammonia prior to injection gives better flame performance because part of the combustion process (vaporizing the liquid) is done by external heat input. Metering and injection of ammonia is simpler in vapor form; however, to prevaporize the fuel requires a heat exchanger and attendant controls.

Ammonia is chemically incompatible with some materials commonly used in a gas turbine. In the fuel system, copper and zinc alloys must be eliminated because of corrosion; and in a pumping system, special attention to bearing surfaces is needed because of the low lubricity of ammonia.

1.2 COMBUSTION INVESTIGATION

It was apparent early in the program that several methods of injection and burning ammonia existed. The following methods were evaluated by rig test.

- Ammonia liquid injection
- Combined vapor and liquid injection
- Ammonia vapor
- Ammonia vapor, catalytical aided

The results of liquid ammonia injection into modified hydrocarbon burning combustors was completely unsuccessful and stable combustion was obtained only by adding a large percentage of hydrocarbon fuel.

Use of ammonia vapor as a pilot flame, with liquid ammonia used for main fuel injection gave relatively satisfactory performance. This technique was considered likely to add complexity to both the combustor and the ammonia fuel metering and distribution system. In view of these shortcomings this approach was not pursued.

Although preliminary ammonia vapor combustion tests indicated this was a practical approach, the use of catalysts as a means of improving the combustion of ammonia vapor was considered. Two methods were investigated, oxidation catalysts and decomposition catalysts. Tests indicated that both methods could be successfully adapted for use in gas turbine engines. However, in comparing the two approaches, it was concluded that use of the decomposition catalyst would necessitate use of a larger combustor with a slower response. Since only a limited number of methods of ammonia combustion could be fully explored, it was decided to

concentrate on (a) vapor combustion and (b) oxidation catalyst-aided vapor combustor. The results of the rig tests conducted on these two approaches are discussed in the subsequent paragraphs.

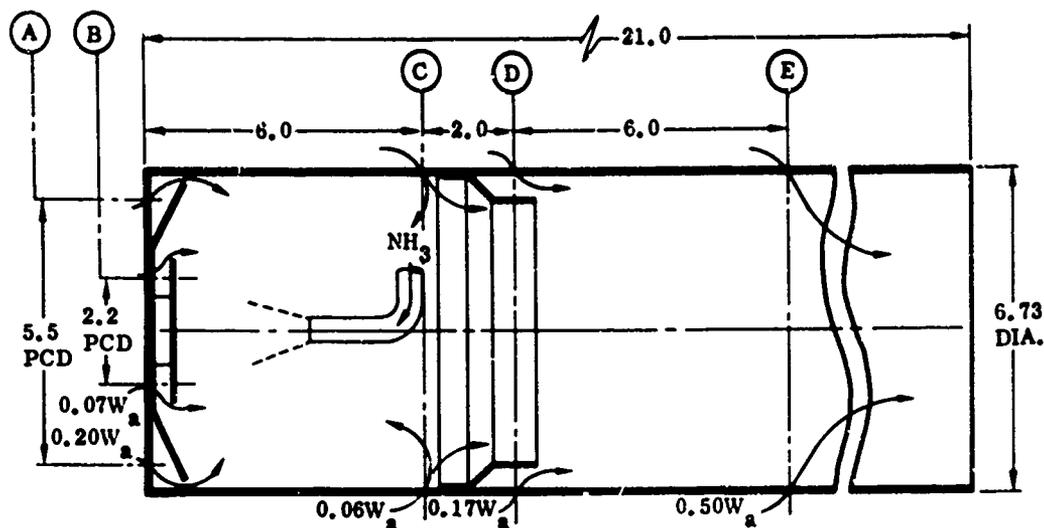
1.2.1 Vapor Combustor Rig Tests

The ultimate program objective was to modify an existing hydrocarbon burning engine to operate on ammonia fuel. Test conditions and flow rates, therefore, were generally confined within the range required for this engine. The engine was required to operate in an ambient environment from minus 65 to plus 130°F at altitudes up to 8000 feet.

Preliminary tests indicated that the effect of air-to-fuel ratio, pressure, and temperature on the flame performance of ammonia corresponded to that obtained with a hydrocarbon system. Therefore, there seemed to be justification in supposing that the same factors that affect flame performance in a hydrocarbon combustor have a similar effect in an ammonia combustor.

The aim of the combustor tests was to arrive at a combustor size which would allow stable and efficient flame performance over the range of operation. To set up a particular condition of pressure, temperature, and air-to-fuel ratio is no problem. However, variations in combustor mass flow also change combustor pressure drop. To have adjusted the combustor hole size at each differing mass flow to maintain pressure loss constant would have entailed a prohibitive amount of work and, therefore, was not considered. The combustor tests were run on a rig facility having a heat exchanger or cold air turbine upstream of the combustor. Air temperatures were controllable within the range of minus 40 to plus 1000°F.

After a number of inconclusive tests using modified combustors from different sized hydrocarbon-burning engines, it was decided to run tests on a series of identical scaled combustors. These combustors were simple, straight-sided tubes with flat domes and were constructed in 5, 6, 7, and 8-inch diameter sizes. Test results showed that size has a significant effect on flame performance. A 6-inch combustor could sustain a flame at nearly twice the velocity of a 5-inch combustor. However, the differences between the 6, 7, and 8-inch combustors were not so significant. It seemed evident that a scale effect was responsible. This scale effect was noticed in hydrocarbon combustors and was presumed to be the result of increased surface-to-volume ratio quenching the flame reaction. Normally, the significance of wall effects on a hydrocarbon combustor would not be noticed in combustors above 3 inches in size except when high combustor loadings occur. To a considerable degree, this wall effect can be minimized by keeping the combustor walls at temperatures above 1200°F.



W_a = AIR MASS FLOW TOTAL

HOLE PATTERN		
LOCATION	N _{O.}	DIAMETER (in.)
A	24	5/16
B	8	5/16
C	8	9/32
D	24	9/32
E	12	11/16

FIGURE 1. AMMONIA COMBUSTOR FOR RIG TESTS

Based on the scale tests, a combustor was designed to maintain a high wall-temperature and sized to minimize the scale effect. Tests were conducted with various configurations, but the best results were obtained with the combustor shown in Figure 1. The most significant effect seemed to be in the method of fuel injection; upstream injection was superior to downstream injection. In hydrocarbon combustors, such a change is not critical, and it is not of particular importance where injection takes place provided the required degree of mixing is obtained. Tests were then performed over a range of engine operations, including cold, standard, and altitude conditions. The resulting flame stability loops are shown in Figures 2 and 3. The combustor total temperature rise is that over the complete system including the dilution zone. Velocity is based on inlet air conditions to the primary flame zone and on its cross-sectional area.

Next, ignition tests were conducted. The ability to obtain reliable ignition, with a stoichiometric flame zone, was mainly dependent on having maximum ignition energy and minimum combustor velocity (Fig. 4).

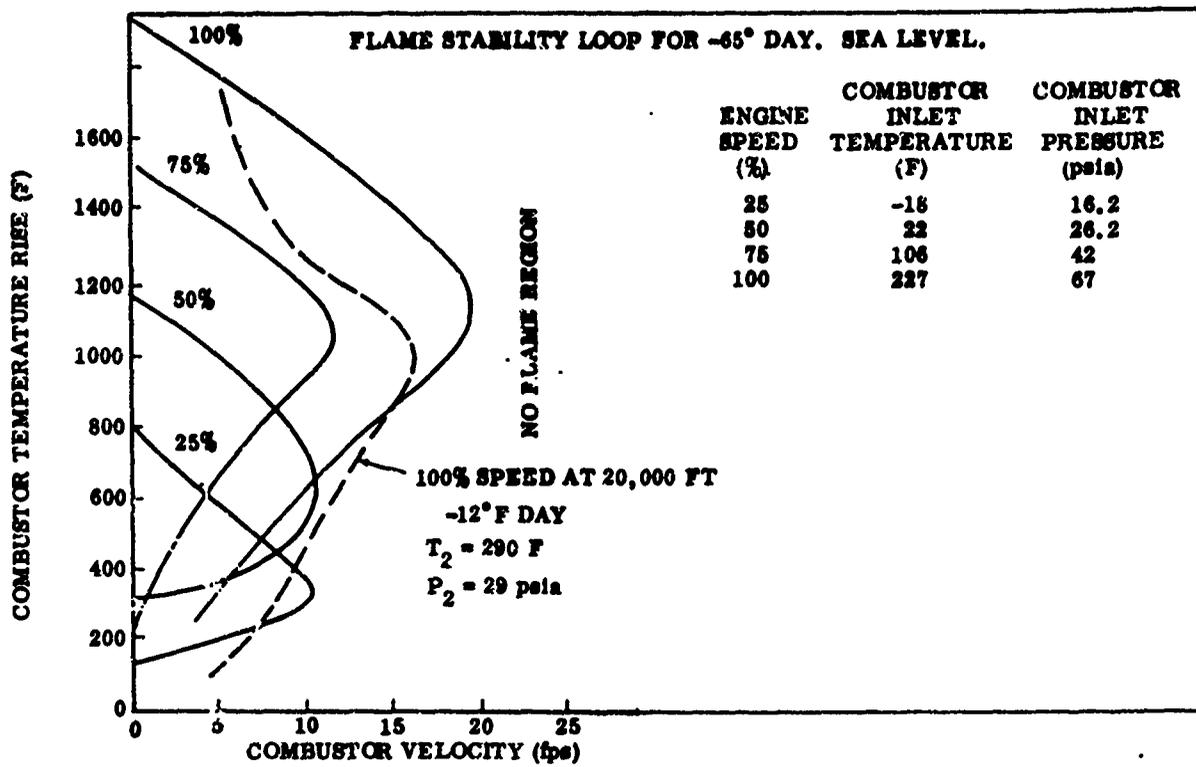


FIGURE 2. FLAME STABILITY LOOPS FOR AMMONIA VAPOR
(60°F Day, Sea Level)

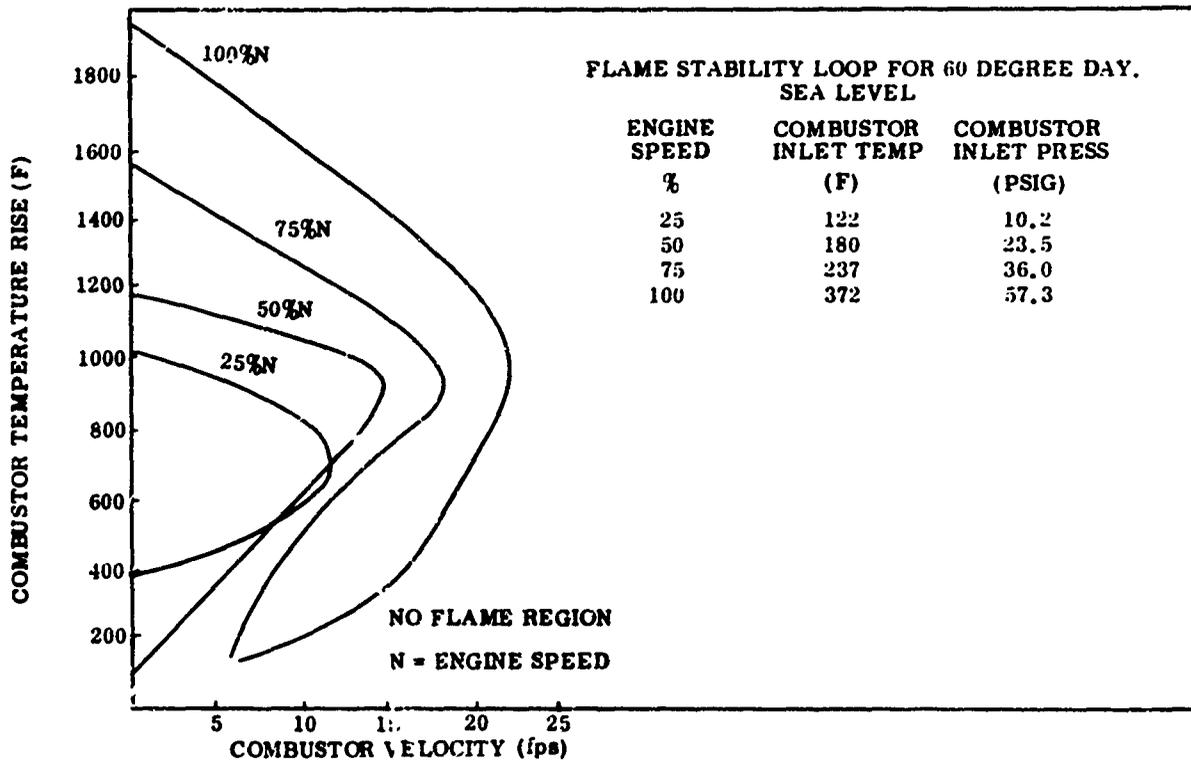


FIGURE 3. FLAME STABILITY LOOPS FOR AMMONIA VAPOR
(-65° F Day, Sea Level)

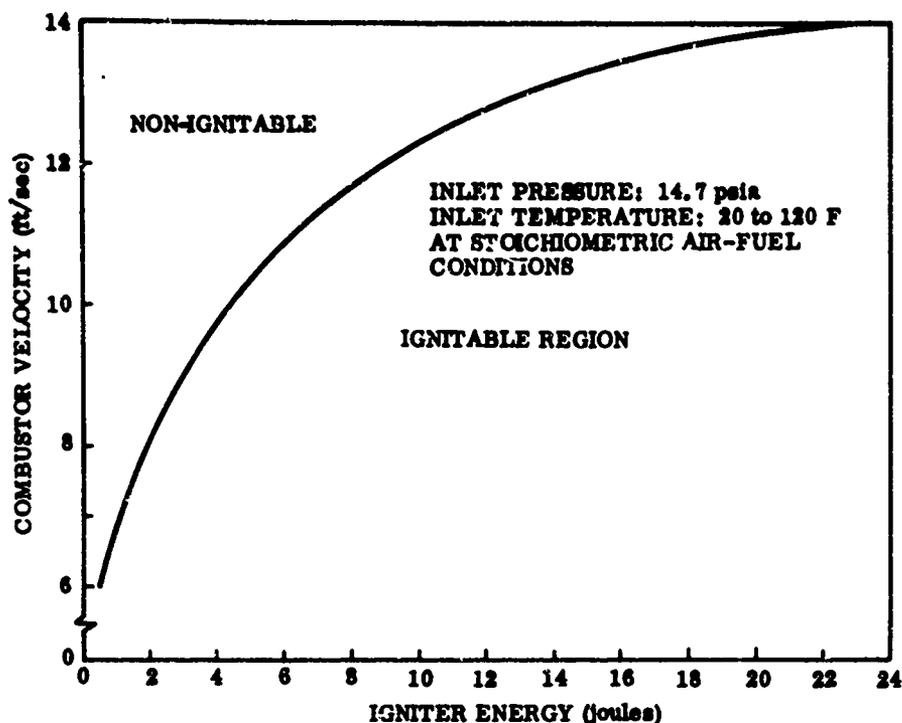


FIGURE 4. IGNITION CHARACTERISTICS FOR AMMONIA VAPOR COMBUSTOR

1.2.2 Oxidation Catalyst Combustor Rig Tests

Catalysts have the ability of speeding up a chemical reaction without being consumed in the process. By burning a portion of the ammonia with air on a catalyst surface, a sufficient amount of heat is provided to permit catalytic decomposition of the remainder of the ammonia into H_2 and N_2 .

Ammonia vapor and air was passed through a catalyst bed of 1/8-inch cylindrical pellets of alumina impregnated with platinum. Preheating the bed to a temperature of 400°F resulted in a reaction between ammonia and air. By varying the air-to-fuel mixture ratio, the bed temperature could be varied over a wide range of conditions. When operated on the fuel-rich side of stoichiometric, vigorous flame could be supported downstream of the catalyst as combustion of the excess ammonia was completed. By varying the flow rate of the mixture and the air-to-fuel ratio, measurements of catalyst bed temperature and bed pressure loss were obtained. When the catalyst bed temperature was increased, more complete cracking of the ammonia occurred.

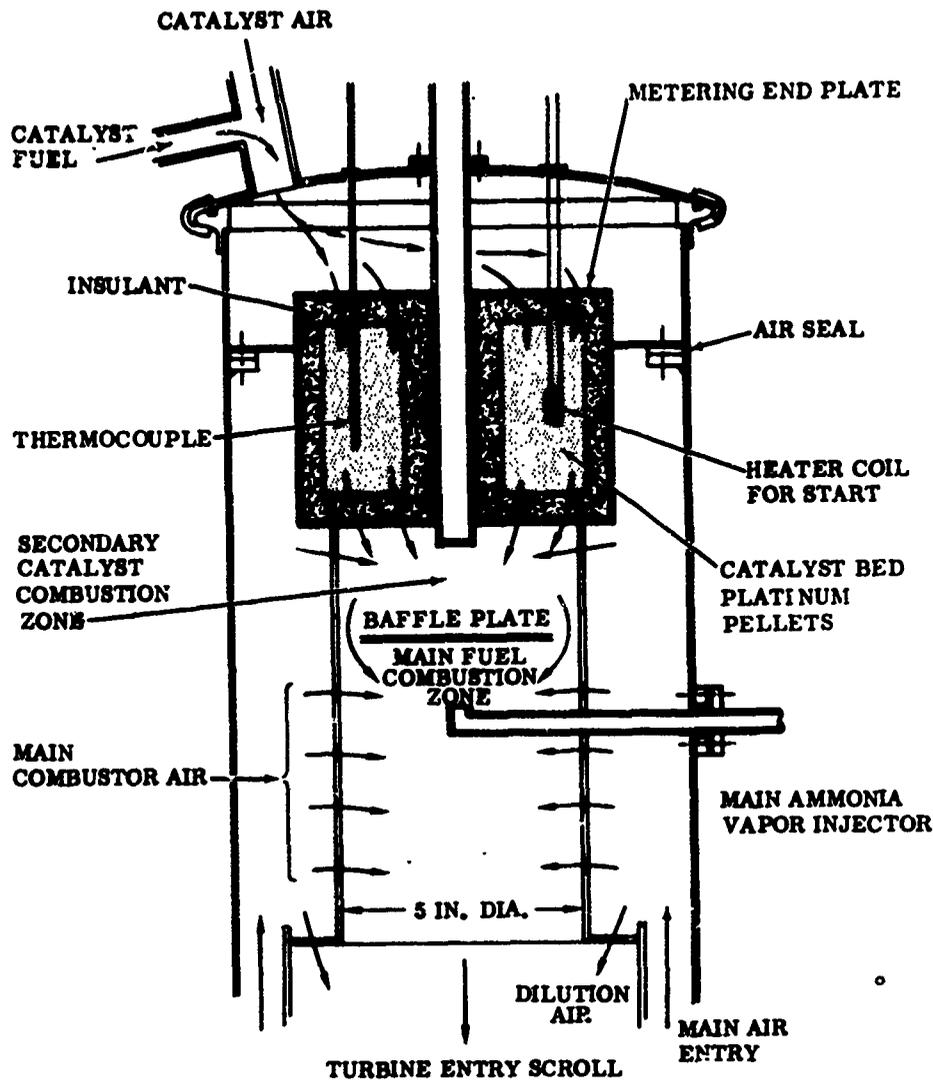


FIGURE 5. OXIDATION CATALYST COMBUSTOR

The results indicated that a significant reduction in combustor size and response time was possible by use of an oxidation catalyst integral with the combustor. Although external heat energy was required to provide the initial reaction, once started it would be self sustaining.

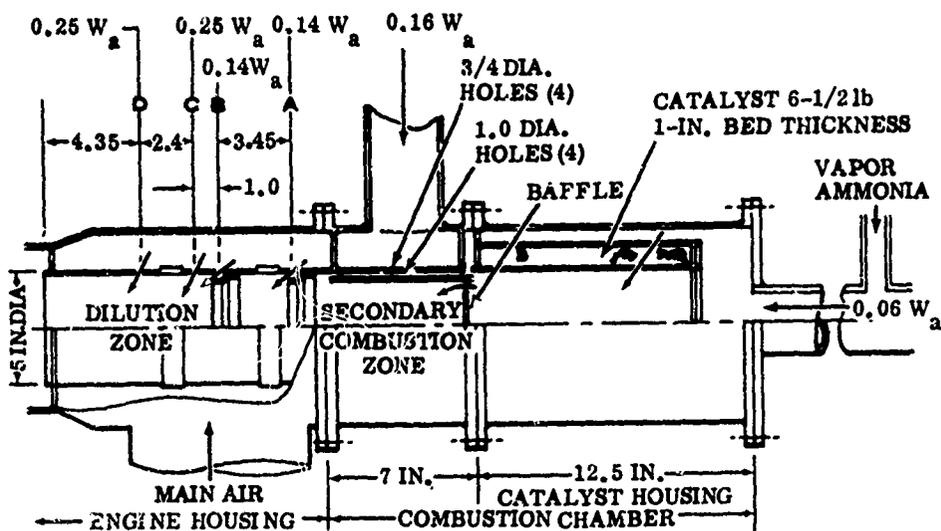
A full-size, engine type, combustor rig was fabricated using a catalyst bed containing platinum impregnated alumina pellets as a pilot burner. The principle of operation was based upon the ammonia flow into the catalyst being fixed at a value somewhat less than the estimated no-load engine fuel flow. Main fuel injection was downstream of the catalyst bed (Fig. 5). By this means, flow and pressure losses in the catalyst were kept to acceptable levels. The reaction lag occurring in the catalyst would not affect the response of the combustor, since the catalyst flow rate and fuel flow would be held constant at all engine loads.

The results of the rig tests were disappointing. When run at engine design conditions, the limiting velocity (based on the primary zone cross section) was only 11 fps. Measurements of catalyst temperature showed that a large temperature gradient existed across the catalyst bed. The maximum temperature of 1900°F was achieved in only a very small layer of the catalyst near the outlet of the bed. Temperatures at the inlet face of the catalyst were only slightly above the inlet air and fuel mixture temperature and were not, therefore, contributing to the catalytic reaction. The efficiency of the catalyst is considerably reduced at lower temperatures, and it was obvious that the shape of the bed used for this design was grossly inefficient. It was concluded that if the frontal area of the catalyst was increased and the length reduced, the average catalyst temperature could be raised close to the maximum temperature. Such a change might improve the catalyst efficiency by an order of magnitude. Reducing the catalyst bed length by one-half and doubling the surface area also reduces the pressure drop by the cube root. Based on this approach, a review of the sizing required for a catalyst combustor capable of handling the total engine flow showed that the actual volume required would be only slightly larger than the original pilot flow catalyst. The most significant difference with passing all the fuel flow through the catalyst would be the method of control and the increased catalyst loading.

A new catalyst combustor capable of handling full engine fuel flow was designed (Fig. 6). The catalyst bed was in the form of a hollow cylinder with a catalyst thickness of 1.0 inch.

While maintaining a catalyst bed temperature of 1900°F, tests were run over a range of inlet air temperatures from 200 to 400°F, pressures from 15 to 46 psia, and combustor reference velocities of up to 56 fps. Catalyst loading went as high as 400,000 scf/ft³/hr. Combustion efficiency was maintained at a high level and varied between 92 and 96 percent from high to low velocities.

The radial and axial temperature gradients through the bed were very low, less than 150°F. By varying the catalyst bed temperature at a fixed fuel flow with control of airflow, it was found (Fig. 7) that an almost unique relation between catalyst temperature and combustor temperature rise existed over the range of velocities used (10 to 56 fps). As the combustor velocity was increased from 10 to 56 fps, the rich blowout of the catalyst (extinguishment of the combustion flame in the secondary zone downstream of the catalyst) varied from 1550 to 1650°F. When rich blowout occurred (as a result of a deficiency of air) a flame could be immediately relit by an increase in airflow. Similarly, the fuel could be reduced to zero and, if returned to normal within a few seconds, ignition would immediately re-occur. The stability range of the catalysts in transients was therefore infinite. However, steady-state allowable variations in catalyst air-to-fuel ratio were only about 1.5:1 and indicated that a rather precise air-fuel control was needed.



W_a = TOTAL AIR FLOW

DILUTION HOLE PATTERN		
LOCATION	NO.	DIAMETER (IN.)
A	26	11/32
B	26	11/32
C	4	1.0
D	4	1.0

FIGURE 6. SCHEMATIC OF OXIDATION CATALYST COMBUSTOR

Variations in the size of the baffle downstream of the catalyst were made in an attempt to improve the rich blowout. Removal of the baffle caused a sharp deterioration in stability but otherwise, no matter what size baffle was used, no change in stability was found.

Ignition of the catalyst was first done by preheating with an electric heating element so that a local catalyst temperature of 400°F was exceeded. A fuel-rich ammonia-air mixture was then passed through the bed. Catalytic oxidation would commence immediately at the hot spots, spreading throughout the bed within a few minutes. During this warmup period, large temperature gradients occurred with the bed. Any error in air-to-fuel ratio in the catalyst could produce temperatures capable of destroying portions of the catalyst container within a few seconds. There was no means of monitoring actual maximum temperature; the system was obviously too hazardous.

To avoid this problem, the catalyst was preheated by blowing warm air through it. Provided a temperature of 400°F or more was achieved through all the catalyst bed, the addition of ammonia would start an immediate reaction throughout the bed.

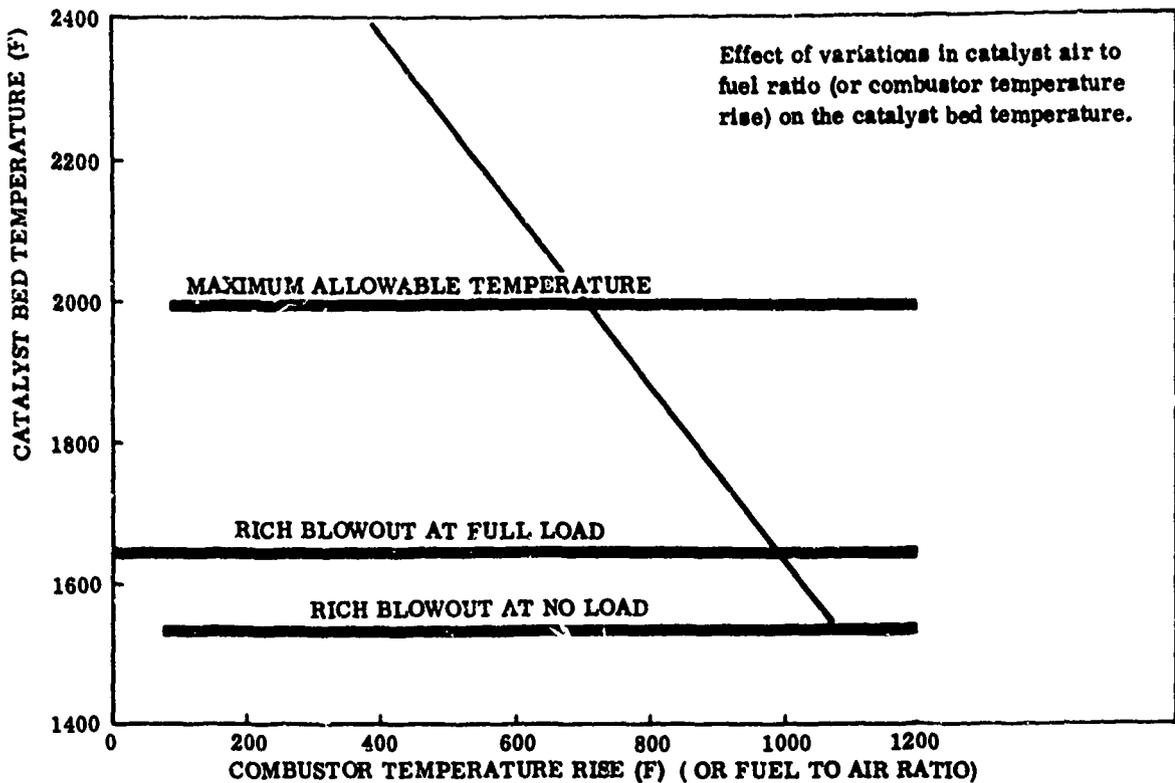


FIGURE 7. OXIDATION CATALYST PERFORMANCE

thereby minimizing temperature gradients. It was possible to limit the maximum catalyst temperature to less than 2000°F, and the variations in temperature during the preburn period to less than 200°F.

The temperature of the bed prior to admission of the ammonia controlled the rate of temperature rise in the bed. Ignition of the combustion system occurred automatically downstream of the catalyst at a bed temperature of about 1800°F provided the correct air-to-fuel ratio in both the catalyst bed and the secondary combustion zone was available. After 30 hours of combustor testing, considerable deterioration in the catalyst basket had occurred because of the high temperatures (up to 2400°F) at which it had run.

It was apparent from the test results that the flow pressure losses could be kept to acceptable levels using this particular bed configuration. The transient time and hence flame response could be kept low. Because of the low temperature gradients in the catalyst, efficiency could be maintained at a high level without the need for a large combustor volume. All the engine fuel could be allowed to pass through the catalyst. The control system would have to provide close control of the catalyst mixture ratio to prevent flameout or overtemperature.

2

ENGINE INVESTIGATION

2.1 DESCRIPTION OF TEST ENGINE

The engine selected for the investigation was the Solar Model T-350 gas turbine. This is a single can burner, simple-cycle, fixed shaft engine with a nominal 250 hp standard day rating. The complete engine, together with its attendant accessory systems, was mounted on a portable skid base. Figure 8 shows the final engine package incorporating the catalyst combustor. All control instrumentation, switches, and electrical sequencing systems were contained in a separate control console. The contract specification required that the engine be capable of operating on both hydrocarbon fuel and ammonia fuel with minimum change-over effort. The single can combustor configuration simplified combustor change-over, but it was necessary to have separate fuel systems for both the ammonia and the hydrocarbon fuels. Although many of the control functions were adapted to allow use in both modes of operation, a considerable degree of accessory system complexity was inevitable.

2.1.1 Power Section

The power plant was a single-stage, centrifugal compressor and a single-stage, radial inflow turbine, with the compressor and turbine overhung in a back-to-back configuration. A cross section of the engine is shown in Figure 9. Changes to the power plant for ammonia fuel operation were minor.

The Phase I investigation of the effects of ammonia combustion products on a typical range of engine materials showed that certain materials were adversely affected. MULTIMET N155, which is substantially used in the hot section of the standard engine, showed reduced resistance to oxidation and thermal fatigue. Inconel 600 showed little or no deterioration and was, therefore, used for the turbine nozzle assembly, turbine scroll, and combustor liner. The standard turbine wheel and turbine exducer, made from Inconel X and Inconel 713C, respectively, were considered satisfactory.

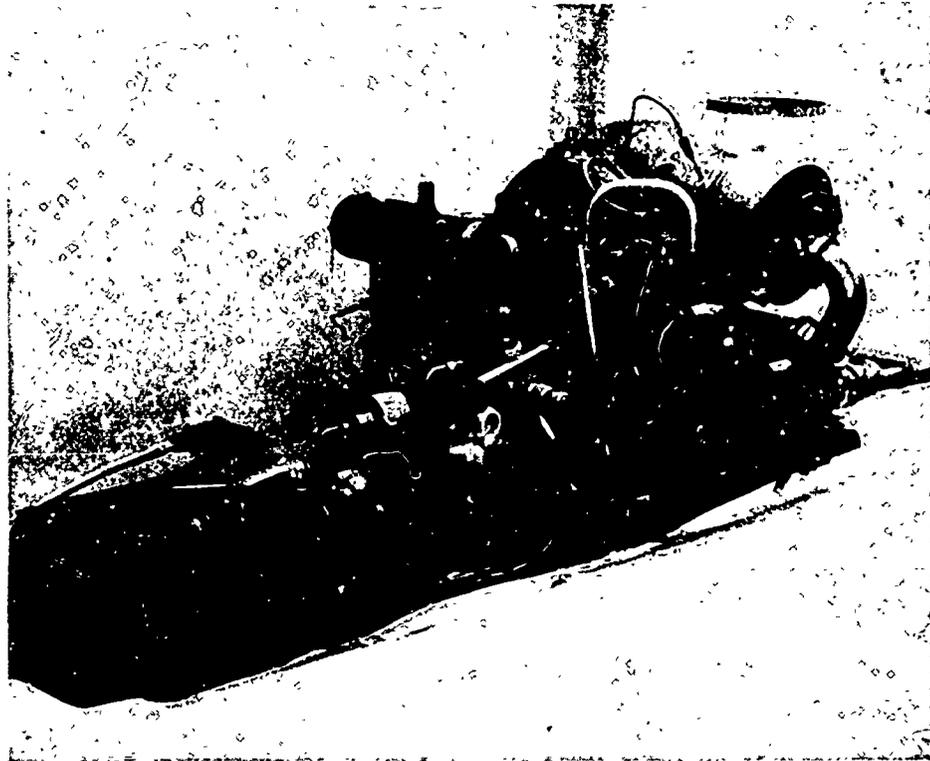


FIGURE 8. SKID-MOUNTED AMMONIA BURNING ENGINE

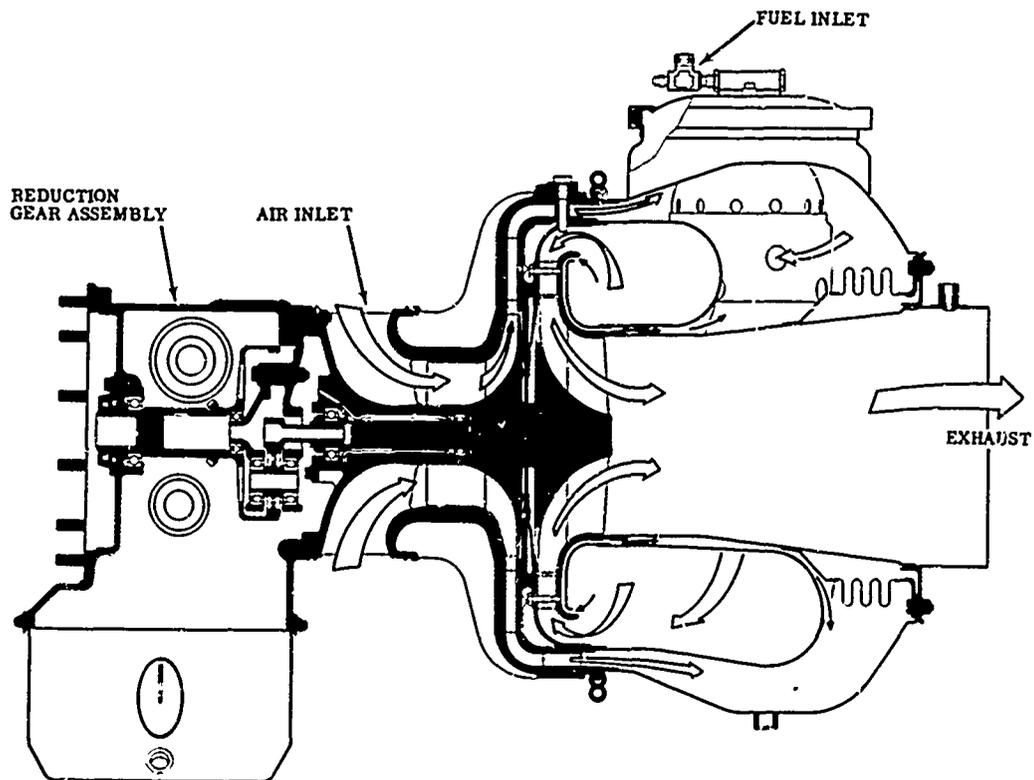


FIGURE 9. T-350 GAS TURBINE ENGINE CROSS SECTION

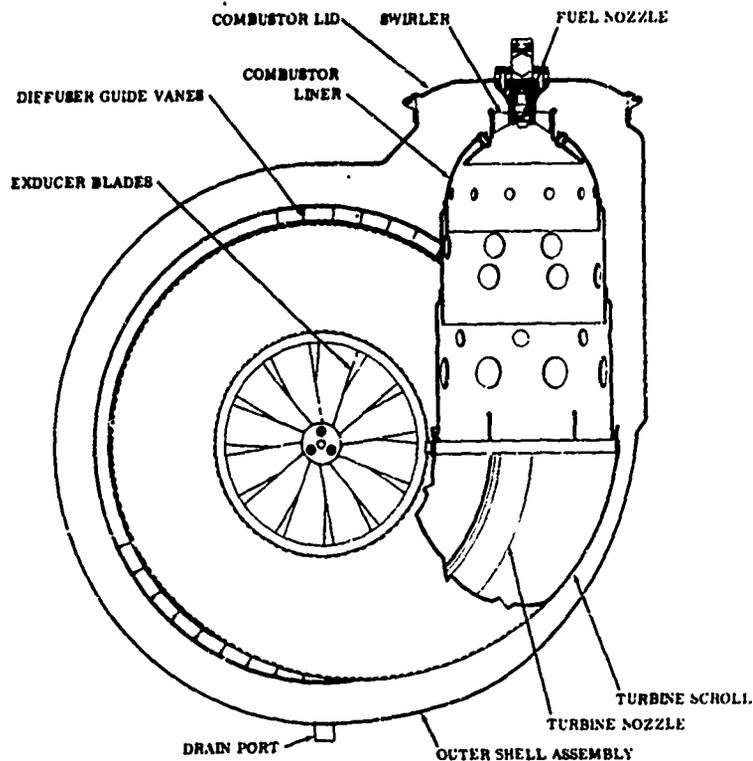


FIGURE 10. T-350 GAS TURBINE ENGINE HYDROCARBON COMBUSTOR

2.1.2 Combustion Systems

2.1.2.1 Hydrocarbon Combustor

A schematic diagram of the standard engine hydrocarbon combustor is shown in Figure 10. Fuel is introduced by a dual-spray, pressure atomizing nozzle in the combustor dome. A primary air swirler is incorporated in the dome. The lower end of the combustor is inserted into the turbine scroll.

The combustor is small in size when compared to the proposed ammonia combustor, with combustion zone velocities of 45 ft/sec at design conditions. The proportions of combustion and dilution air flow were approximately 30 percent injected into the flame zone with the remainder used for wall cooling and dilution.

2.1.2.2 Ammonia Vapor Combustor

Design Criteria

In starting a gas turbine, it is usual to regulate the fuel flow so as to maintain a constant turbine inlet temperature corresponding to the design point full load value.

This will provide the fastest possible acceleration without endangering the engine through over-temperature. Limitations in practice exist through the need to avoid surge and because of inaccuracies in fuel control metering. Examination of Figure 3 indicates another limitation; a considerable shift in the flame stability loop from light-off to full engine speed, such that rich blowout at low engine speed occurs at a much lower turbine inlet temperature than full speed (1000°F as compared to 1950°F). This shift was emphasized by operation during cold day conditions (Fig. 2) and can possibly be explained by a change in the rate controlling mechanism of the flame reaction. Therefore, it would appear that combustor size would be as limited by low speed flame stability requirements (maintenance of a low enough velocity to ensure maximum turbine inlet temperature without flame out) as by high speed requirements.

The tests were not so extensive as to provide precise scaling parameters. On the basis of velocity scaling, a 12-inch combustor would satisfy sea level normal day operation; but on the basis of time to complete combustion, a 10-inch diameter combustor was satisfactory. It was decided, therefore, to fabricate and test 10 and 12-inch diameter combustors, each designed to have identical pressure losses.

Description of Ammonia Vapor Combustor

The vapor burning ammonia combustor was indexed into the existing engine turbine scroll and was retained by the standard V-type clamp used for retaining the cover of the hydrocarbon combustor. The combustion zone was contained within the large diameter section, completely isolated from the dilution section at the plane of the mounting flange (Fig. 11). Combustion air was taken from the compressor discharge plenum and ducted through a butterfly valve into the dome of the combustor. A schematic diagram of the arrangement is shown in Figure 12 and a diagram of the 12-inch diameter combustor is shown in Figure 13. Based upon the combustor rig test experience, wall film cooling was not used since no problems of high wall-temperature had occurred and a hotter wall might be expected to benefit the combustion process.

The variable area air valve was a necessary part of the catalyst combustor system and it was convenient to use it to control the airflow to the flame zone of the vapor combustor. This would allow a constant and optimum air-to-fuel ratio to be obtained and possibly a higher combustion efficiency. As low speed acceleration appeared to be the limitation on combustor size (rich fuel blowout), use of the variable air valve was not thought likely to decrease the necessary combustor size.

The spark plug used was a center electrode, surface discharge type connected to a 20 joule ignition unit with a one spark per second rate. Rig tests (Fig. 4) had shown that this amount of energy could provide lightoff capability to 14 fps. With the 12-inch diameter combustor this velocity corresponds to an engine speed up to 100%.



FIGURE 11. ENGINE ASSEMBLY OF AMMONIA VAPOR COMBUSTOR

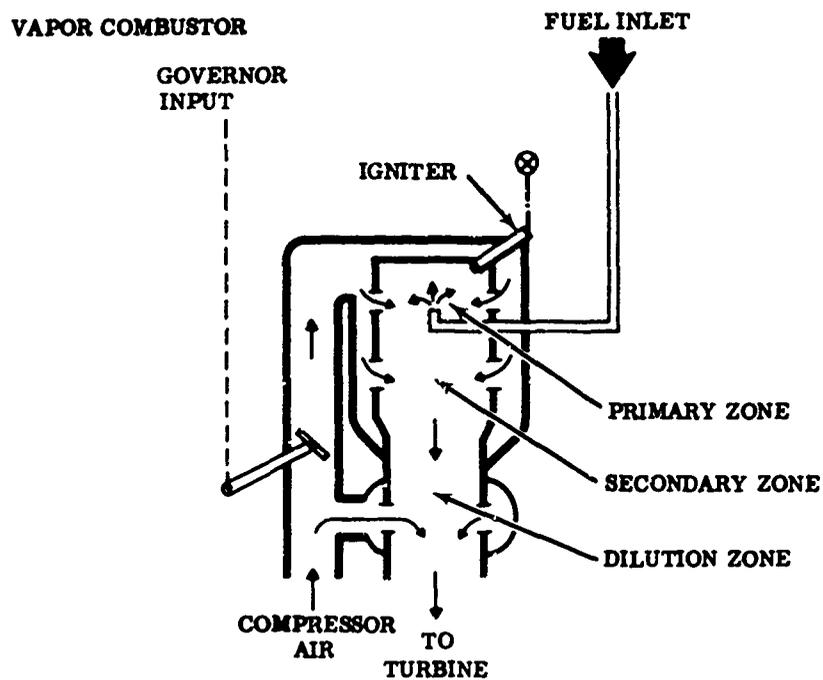


FIGURE 12. AMMONIA VAPOR ENGINE COMBUSTOR CONTROLS

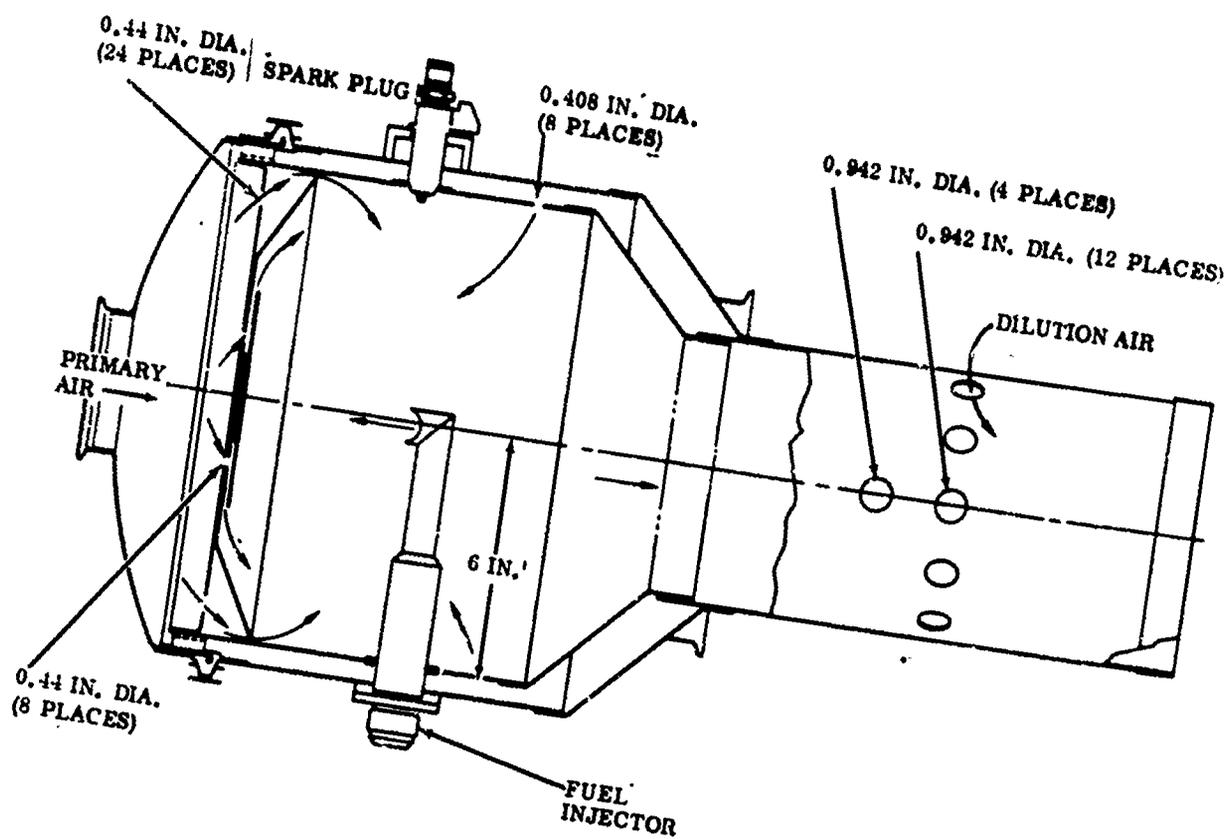


FIGURE 13. AMMONIA VAPOR ENGINE COMBUSTOR

2.1.2.3 Integral Catalyst Combustor

The design of the catalyst combustor for engine tests was essentially based on repackaging the design developed during the Phase I rig tests. The catalyst bed used the same 1/8-inch platinum-impregnated alumina pellets and was essentially the same configuration and design loading. One problem encountered during the rig tests however, was shrinkage of the alumina base of the pellets. This not only caused loss of the catalyst but, more important, it caused hot spots in the catalyst bed due to uneven catalyst distribution. The pellet shrinkage was the result of a phase change of the alumina due to the high operating temperatures. To avoid this, the catalyst pellets were preshrunk by exposure to a 2000°F reducing atmosphere for several hours.

A diagram of the catalyst combustor is shown in Figure 14 and the actual combustor, scroll, and catalyst basket is shown in Figure 15. The combustor mounts directly into the engine turbine scroll and is held in position by the V-clamp normally used for retaining the cover of the hydrocarbon combustor. The most significant change from the rig test combustor was the increase in size of the secondary combustor zone from 5 inches to 7 inches. It was anticipated that this change in size would increase the rich blowout limit of the secondary zone. However, to minimize the length of the

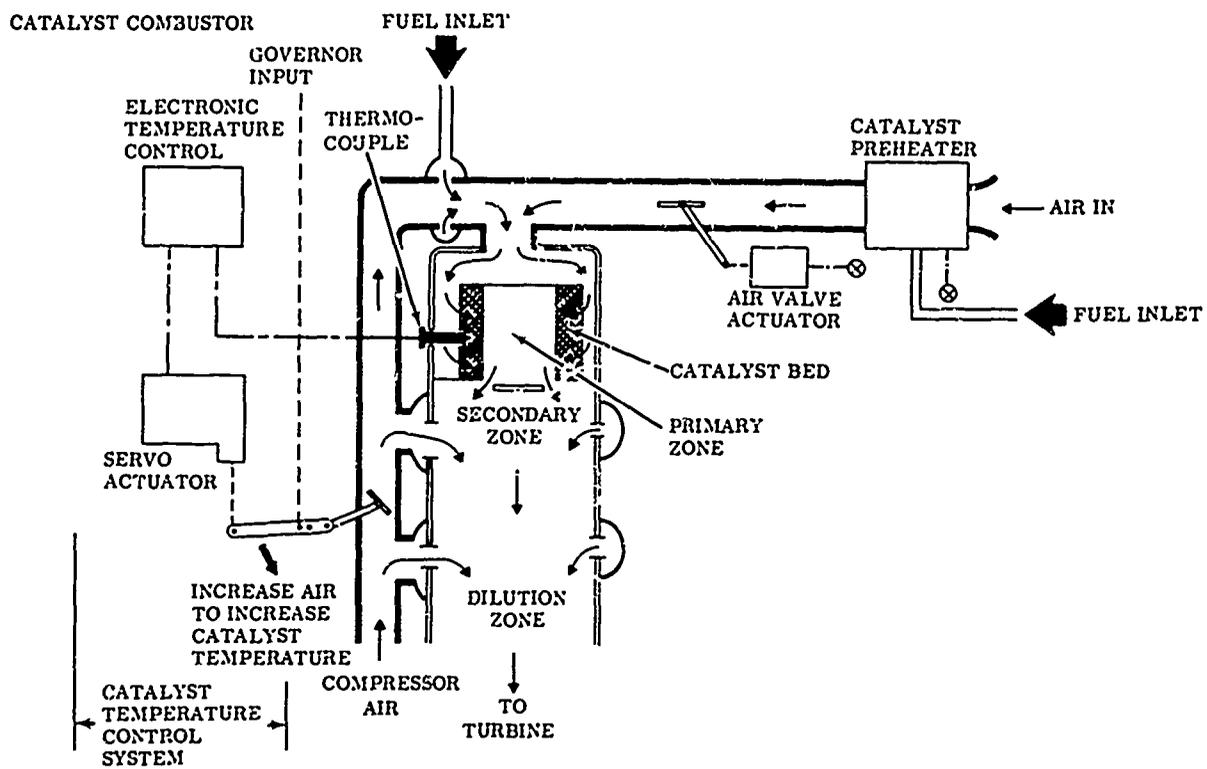


FIGURE 14. CATALYST COMBUSTOR DIAGRAM



FIGURE 15. OXIDATION CATALYST AMMONIA ENGINE COMBUSTOR

combustor, it was necessary to reduce the length of the secondary zone, and introduce all the dilution air at one axial station. Combined primary and secondary air was taken from the compressor discharge plenum and, after passing through a butterfly valve, it was split and the ammonia added to the primary air duct. To minimize the severe catalyst bed distortion experienced during rig tests, the engine combustor was designed with the outer wall made from three separate pieces of Inconel 600 perforated sheet. The sheets were overlapped which allowed free longitudinal movement for expansion. The complete assembly was suspended from the combustor lid.

Combustor pressure loss at full load was 4 percent, an amount dictated by pressure losses in the catalyst, the ducting, and air control valve; not by combustion requirements.

2.1.3 Accessory and Control Systems

2.1.3.1 Design Criteria

The control system requirements were limited to those necessary to provide safe, automatic, engine operation with any one of the hydrocarbon, vaporous ammonia, or catalyst combustor systems tested. Each of the three systems was required to provide automatic starting, acceleration, speed regulation, malfunction sensing, shut-down, and annunciation. The requirement for easy conversion from one combustor configuration to another dictated the extensive use of common components and imposed some restrictions on the specific calibration characteristics of individual system elements. No requirement existed for providing control schedule compensation for environmental conditions, except those prevailing in a typical atmospheric gas turbine test cell.

Every effort was made to utilize standard commercial components, both to ensure maximum reliability and to minimize control development costs. This was achieved, for the most part, by adapting controls used by Solar on other gas turbine engines, or by modifying commercially available components as necessary to provide material compatibility with anhydrous ammonia.

While the delivered engine package was required only to drive a dynamometer load, the test program required that comparative transient performance of the various combustors be evaluated. Provision was made, therefore, for operating the engine either with a dynamometer load, or with a generator and load bank for effecting rapid load transients. The requirement for obtaining transient data was best satisfied by utilizing an electronic governor with dynamic compensation and load sensing. All other tests utilized a conventional hydromechanical governor.

2.1.3.2 Description of The Systems

The fuel and air control systems used for the hydrocarbon or ammonia fueled engine are shown schematically in Figure 16.

Hydrocarbon Fuel System

The hydrocarbon fuel system was adapted from the fuel control system used on the Solar Saturn series engines. Because of limitations in availability of drive pads on the T-350 gear box, fuel was supplied by an electric motor-driven pump rather than by an engine-driven pump. Acceleration fuel was scheduled by the same type of metering valve used on the Saturn gas turbine. The flow area of the metering valve varies as a function of compressor discharge pressure, and the pressure drop across it is maintained constant by a delta-P valve. An integral relief valve protects the fuel pump from overload. Speed control was achieved by a governor valve which was operated mechanically by a Woodward PSG or EG-3 governor. Differential pressure was maintained constant across the governor valve by an integral delta-P valve which bypasses part of the fuel scheduled by the acceleration limiter, as necessary to maintain constant speed.

Ammonia Fuel System

The ammonia fuel systems were identical for either the vapor or catalyst combustor as shown in Figure 16. Acceleration fuel was scheduled by a Woodward gas metering valve. The flow area of the valve varies as a function of compressor discharge pressure, and the pressure drop across it is maintained constant by a Woodward throttling type delta-P regulator. The Woodward PSG (or EG-3) governor mechanically controlled the gas metering valve on a "least" basis to furnish the required fuel to maintain constant speed. A tank was installed upstream from the gas metering valve to collect any liquified fuel which might accumulate because of condensation. The condensate tank includes a manually controlled, electrical heating element and relief valve so that any accumulated liquid could be boiled off through a vent to the exhaust stack. A solenoid-operated, three-position valve served the dual function of fuel shutoff and purge valve. The entire system could be purged of liquid prior to startup by energizing one of the two solenoids, thus venting metered fuel to the exhaust stack. A check valve was also provided in the vent line to prevent back-flow of air into the fuel system in the event of loss of fuel pressure.

Vapor Combustor Air Control System

Since it was determined that a variable fuel-air ratio control was required on the catalyst combustion system, it was decided that it should also be applied to the vapor combustion system. It was thought that the fuel-air control could be used both

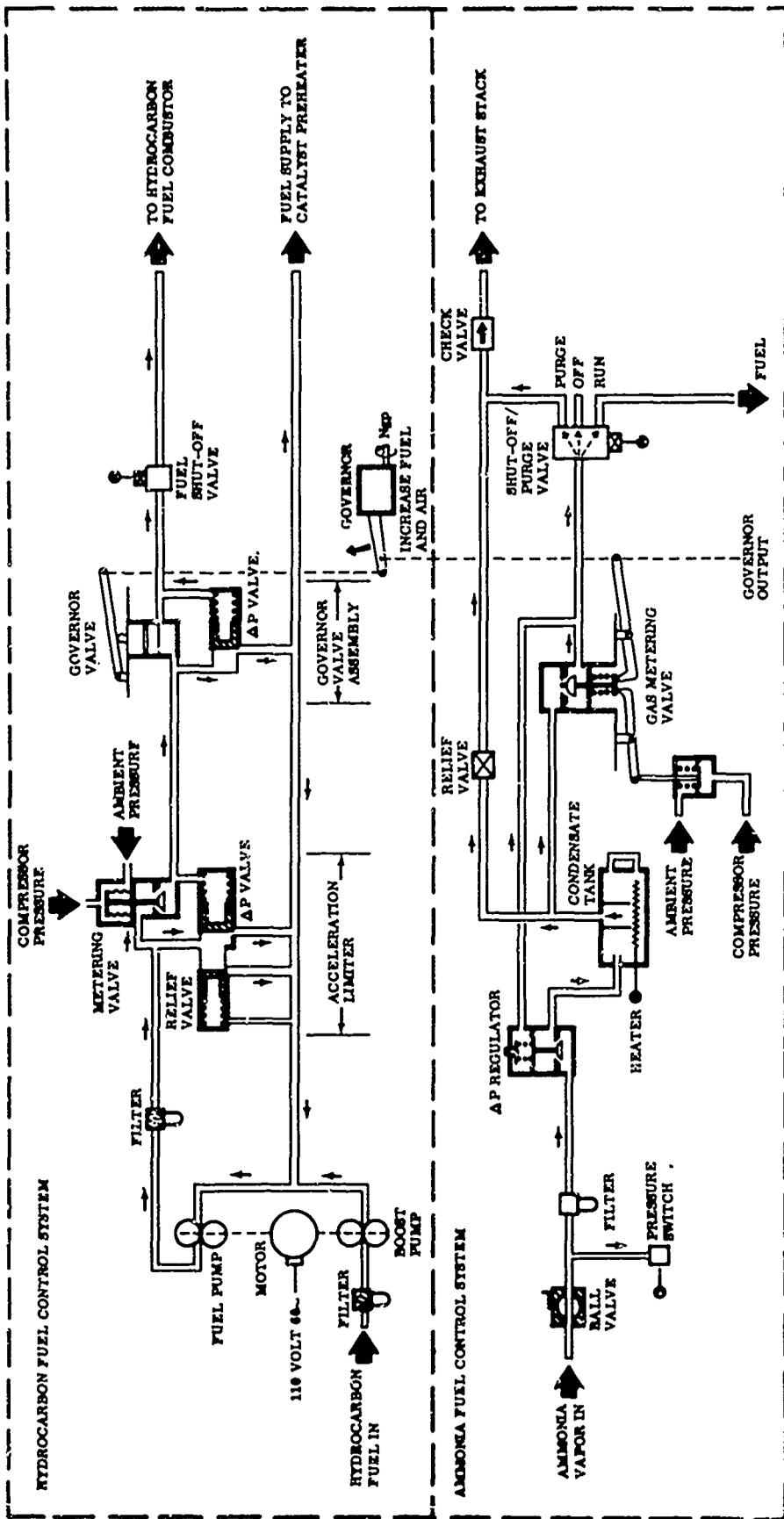


FIGURE 16. FUEL AND CONTROL SYSTEM SCHEMATIC

as a means of minimizing flame-out during load transients and of optimizing combustion efficiency during steady-state operation.

A schematic diagram of the vapor combustor air control is shown in Figure 12. Combustion air was taken from the compressor discharge plenum and directed through a variable butterfly air valve into the dome of the combustor. The control arm of the butterfly valve was linked directly to the engine governor. The arrangement of the linkage was such that when the governor increased the ammonia flow to the engine, it also increased the airflow into the combustor in approximately the same ratio thereby providing an essentially constant combustion fuel-air ratio over the engine load range. During acceleration to rated speed, the governor was in an underspeed condition and consequently held the position of the air valve fixed near the full-open position.

Catalyst Air Control System

While the combustor air control system described previously was adequate for the vapor combustor, the requirements of the catalyst combustor for greater accuracy necessitated closer control of primary zone (i. e., catalyst) temperature. For the catalyst combustor, the electronic catalyst temperature control shown in Figure 14 was incorporated which provided a trimming function to offset temperature errors introduced by schedule tolerances and changes in operating conditions. Two platinum-platinum-13 rhodium thermocouples, selected for long life and fast response, were imbedded in the catalyst bed. The thermocouple output signal was amplified in two stages and compared with a reference voltage, set for 1800°F. The output error signal controlled an electro-pneumatic servo-actuator, powered by externally supplied air pressure and operated through a differential linkage to trim the governor-to-air valve position relationship as necessary to maintain constant temperatures. During rapid load transients, the rapid movement of the governor, to affect both fuel and air flow corrections, prevented large deviations from constant catalyst temperature. Significant errors in temperature, resulting from inaccuracy of the scheduled relationship between fuel and air valve positions or from changes in catalyst performance, were corrected by action of the catalyst temperature trim system.

Starting the catalyst burner required a preheat and preburn period during which the catalyst bed was heated to about 1800°F to obtain automatic ignition of the air-fuel mixture injected during cranking. During this period, the air valve was held in the full-closed position by the return spring on the governor, and warm air (550°F) was provided by a combustion heater to heat the catalyst bed. When the catalyst bed temperature reached 450°F (as indicated by the thermocouples) an automatic switching circuit turned off the fuel supply to the heater, but allowed the heater fan to continue to pump air through the catalyst. At this point, a constant metered supply of ammonia fuel was introduced by opening the fuel shut-off valve. The flow rate was controlled by the metering valve to a value which provided a fuel-air mixture on the air-rich side of stoichiometric. With the catalyst preheated to a temperature of 450°F, oxidation

began to take place on the catalyst surface as soon as the ammonia fuel was introduced and caused a sudden temperature increase of the catalyst bed. The fuel-air mixture was so selected that the maximum steady-state temperature which could result during this preburn phase would be approximately 1950°F. When the temperature reached approximately 1800°F (as indicated by thermocouples sensing catalyst temperature) engine cranking was automatically initiated, the fan was turned off, and the damper valve was closed. Once the engine started to rotate and sufficient hydraulic pressure obtained to operate the governor, the fuel metering valve opened to permit compressor air to enter the primary and secondary zones of the catalyst combustor at the approximate air-fuel ratio necessary to maintain the 1800°F catalyst temperature. Errors from this temperature were corrected by the catalyst temperature trim system, as previously indicated.

2.2 ENGINE TESTS

2.2.1 Test Set-Up

All engine testing was conducted in a standard engine development test cell. The supply of ammonia fuel was from two 1200-gallon, portable, ammonia storage tanks located externally to the test cell. A thermally insulated, 2-inch diameter vapor output line was connected to the top of each tank and interconnected so that both tanks were paralleled. During engine operation, the ammonia tank vapor pressure was maintained at 110 to 140 psig by means of a separate heating system. This consisted of a circulating pump, a natural gas fired heater, a heat exchanger, and the necessary plumbing. Liquid ammonia was drawn from the bottom of the tanks, pumped through the heat exchanger, and back to tank. The ammonia flow to the engine was measured by an orifice, conforming to ASME test code standards, installed at the inlet to the engine combustor. Hydrocarbon fuel, at atmospheric pressure, was supplied to the engine boost pump through a flow meter. Measurement of engine airflow was obtained from a bell-mouth, inlet venturi. Temperatures at the compressor inlet, turbine inlet, and exhaust was measured with conventional chromel-alumel thermocouples. For measurement of engine pressures, all gage lines, fittings, and parts in contact with the corrosive properties of ammonia were stainless steel. King oil was used in manometers sensing ammonia pressure due to the potential chemical reaction of ammonia with water or mercury. The governor and combustor air valve control arms were fitted with Selsyn generators to provide a visual indication of angular position in the test cell control room.

2.2.2 Ammonia Vapor Combustor

Initial problems were experienced in obtaining ignition. Rig tests had shown that use of 20 joules of ignition energy could provide light-off capability to 14 fps. Since this velocity corresponded to an engine speed of 100 percent (using the 12-inch

diameter combustor) it was assumed that the air-fuel ratio was incorrect. Investigation showed that consistent ignition could be obtained if the acceleration control P_{cd} line was disconnected so that the starting fuel flow was fixed. It was also found that the higher this fixed starting fuel flow, the higher the engine speed at the time of ignition. It was possible to consistently start the engine at speeds up to 35 percent (the limit of the starter cranking capability) depending upon the starting flow selected. Lowering the constant starting fuel flow as a means of lowering the ignition speed was limited to a minimum of 14 percent speed; however, ignition could not be obtained below 14 percent at any fixed starting fuel flow.

Based upon the results of the investigation, it was assumed that the ignition problems occurring with the acceleration control P_{cd} line connected were due to an over-rich fuel-air ratio. The control metering characteristics at the ignition speed were approximately:

$$W_f = K_1 P_{cd} + K_2$$

where W_f = metered fuel flow

K_1 and K_2 = constants

P_{cd} = compressor discharge pressure rise

At engine speeds below 14 percent, P_{cd} is essentially zero, and the setting of K_2 determines the starting fuel flow. The change in combustor fuel-air ratio occurring during this initial cranking phase is extremely large. It is assumed that this large change, together with the rapid rate of engine acceleration through the low portion of the speed range and the low spark rate, made it virtually impossible to obtain ignition. Above 14 percent, the combustor fuel-air ratio is controlled at a relatively constant value as a function of the P_{cd} signal to the control. It was assumed, therefore, that failure to light-off at higher speeds was due to the acceleration control maintaining an over-rich fuel-air ratio. The problem was partially solved by shimming the acceleration control to provide a small dead-band to delay the onset of the fuel flow increase due to increasing P_{cd} . However, this particular fix did not completely solve the problem, due primarily to hysteresis in the acceleration control. A solution to provide higher reliability would have required reducing the gain of the acceleration control schedule. The schedule, however, was optimized for operation with the catalyst combustor and any such change would have compromised that systems operation.

Acceleration after ignition proved to be extremely difficult with the 10-inch diameter combustor, and only in isolated instances was it possible to exceed 60 percent speed without flame-out. This was despite the fact that by manual adjustment of

the air valve position it was possible to vary the fuel-air ratio during acceleration. When 100 percent speed was reached, the combustor would usually flame-out as soon as the governor cut back the engine fuel flow. The 10-inch diameter combustor was clearly inadequate; therefore the main effort was concentrated on the 12-inch diameter combustor.

The 12-inch diameter combustor also had a narrow flammability range, although considerable greater than the 10-inch combustor. However, flameout during acceleration or at full-speed under transient load operation was not a problem once the governor and air valve linkage was optimized. Tests were run at full-speed to determine the effects of fuel-air ratio on engine fuel flow. It was found that even at the extreme case between optimum fuel-air ratio and rich or lean limits, the variation in combustion efficiency was extremely small. No explanation for this phenomenon was found. Operation of the engine for 8.3 hours indicated no serious deterioration at normal ambients and power levels. Measurements of combustor wall temperature, however, indicated that for adequate life some wall cooling would be required.

2.2.3 Ammonia Catalyst Combustor

A substantial amount of control system adjustment was necessary before the engine could be satisfactorily started and operated at rated speed. Initial engine starts were made by preheating the catalyst bed to 430°F with a combustion air heater. It was found, however, that portions of the catalyst were not being heated to a sufficiently high temperature to promote a catalytic reaction and, during the preburn phase, large temperature gradients developed. This prevented accurate measurement of catalyst bed temperature, detracted from catalyst bed performance, and introduced the risk of serious damage from overtemperature. These problems were minimized by increasing the preheat temperature to 500°F. The maximum output temperature of the combustion heater however, was 550°F and, with this limited heat capacity, the preheat cycle time was excessive (about 8 minutes), when starting from room temperature. During the preburn phase, ammonia fuel was added to the combustion heater airflow and the catalytic oxidation process raised the catalyst bed temperature to 1800°F. Repeatability of the preburn fuel-air ratio presented occasional difficulty due to hysteresis in the fuel control. During this preburn phase, the catalyst was operated on the fuel-lean side of stoichiometric, in contrast to normal engine operation, which was on the fuel-rich side. If the fuel flow during preburn was too low, the time required to reach the 1800°F starting temperature became excessive; and, due to the fact that the hot gases were being exhausted from the compressor inlet as well as the exhaust, it was possible to overtemperature the aluminum compressor casing. Increasing the fuel flow to speed up the preburn cycle was not without danger however; and if the fuel-air ratio was adjusted to provide too high a stabilized reaction temperature (say 2500°F) the rapid temperature rise could create hot spots sufficient to damage the catalyst. The fuel-air ratio during preburn was set for a stabilized reaction temperature between

1900 and 2000°F. However, the rate of catalyst temperature rise was carefully monitored during the preburn cycle to ensure that the fuel control was operating correctly. The elapsed time of preburn, with the control operating correctly, was approximately 3 minutes.

All the troublesome factors would eventually be eliminated in any control development program and did not appear to indicate a basic problem with the combustor/control concept. Some adjustment of the catalyst temperature trim system gain was necessary to achieve stable operation. Linkage gains were modified in an attempt to match the governor/air valve relationship to provide a rising catalyst temperature with load. Because of the mismatched, non-linearities of the fuel valve and air valve, this was not completely successful. The steady-state performance of the catalyst air-fuel ratio control system was characterized by a slight dip in temperature to about 1750°F at partial load, while the temperature at no-load and at full-load was higher by 50 to 75°F. Modifications to improve the linearity of the air-fuel ratio system over the load range would correct this discrepancy.

The major problems encountered with the catalyst combustion system were governor instability and deterioration of the catalyst bed container. The problem of instability was basically caused by the nonlinear flow characteristics of the butterfly air valve. The slope of the effective area change is very steep when the valve is operating near the closed position, but flattens drastically as the valve is opened further. When the engine was operating at no load, the air valve was nearly closed and the control system gain was excessively high. By running the governor heavily damped, it was possible to prevent limit cycling, but its response to any load change was unacceptable.

The performance of the system was good at full load operation with the butterfly valve half-open. It was possible to operate the governor at a similar gain and with similar dampening to the values optimized for the vapor combustion system. Increasing the catalyst pressure drop would have forced the butterfly valve to open further at engine no-load operation. This was not possible, however, because during acceleration to rated speed, it was necessary for the air valve to open fully to supply sufficient air to maintain the desired 1800°F catalyst bed temperature. The extra air was necessary because of the higher temperature rise required within the catalyst bed resulting from the lower compressor delivery temperature. There seems little doubt that an air valve with more favorable linear opening characteristics would have significantly improved the no-load control system stability.

During the initial engine testing, only three catalyst bed thermocouples were used; one for the temperature control system and two for independent temperature read-out. After a few hours of operation, the temperature spread between the thermocouples increased and the performance deteriorated. This was at first thought to be due to buckling of the inner wall of the catalyst bed, since subsequent reforming of the container and repacking the catalyst corrected the problem. It was assumed that the buckling

had created unequal distribution of the catalyst within the bed and possibly caused some of the thermocouples to be in only partial contact with the catalyst pellets. However, further investigation indicated the thermocouple error was primarily due to leakage of air from the secondary zone around the thermocouples into the catalyst, creating hot spots around the thermocouples. Restrictions were added to the air supply ducting to ensure that the pressure in the secondary air scroll was slightly lower than the primary air delivery passage. By this means, any air and fuel leakage was into the secondary zone and thermocouple errors induced from this source were eliminated.

The configuration of the final design of the catalyst combustor is shown in Figure 17. To avoid distortion resulting from the large temperature gradients experienced during starting and operation, it was necessary for both the inner and outer walls to expand and contract freely in all directions. The outer wall was made from three separate pieces of perforated sheet; each piece was overlapped and bridged by a flexible V-section welded to both adjoining pieces. The complete outer wall slipped loosely into the catalyst bed frame and was allowed over 0.25-inch of free longitudinal movement. The inner wall was the most critically loaded section since it was subjected to the highest temperature and pressure drop. The use of a corrugated section minimized the pressure drop and provided necessary flexibility to withstand the temperature gradients. The inner support rings were necessary to prevent the bowing that resulted from pressure drop. The perforated sheet material was 0.062-inch Inconel 600. However, this material was found to be inadequate for the support rings; the rings were subsequently made from 0.080-inch Hastelloy X, which had a higher yield strength at the 1800°F operating temperature.

The number of thermocouples used for the final design of the catalyst bed was increased to six. After 2 hours of operation, the total spread between thermocouples was less than 50°F at all engine loads. No visible deterioration of the catalyst bed was evident.

The performance of the catalyst deteriorated noticeably at high engine loads, particularly if the control temperature was reduced. Flameout in the secondary zone occurred at a catalyst bed temperature of 1600°F at engine full-load, and at approximately 50°F less at no-load. To minimize the total volume of the catalyst combustor, it was designed to operate at very high loadings. The length of the secondary zone was little more than half of one diameter. It is believed that a small reduction in loading, with some optimization of the secondary zone baffle, would significantly improve secondary zone combustion performance. Such an improvement would allow the catalyst bed to be operated at a lower temperature which would greatly reduce the problem of obtaining adequate catalyst bed container life.

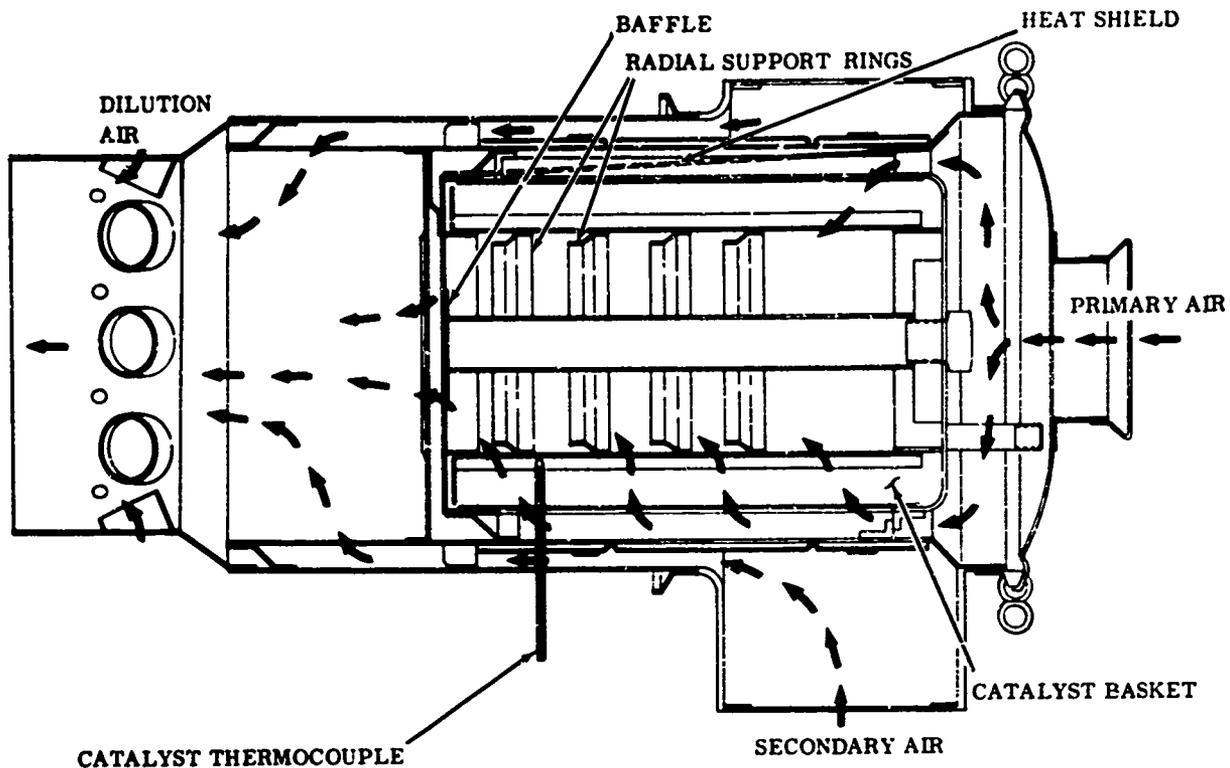


FIGURE 17. FINAL CATALYST COMBUSTOR

2.2.4 Transient Tests

An attempt was made to evaluate the comparative transient performance of the hydrocarbon, vapor, and catalyst combustors by conducting step load changes and observing the speed recovery characteristics of the engine. For this purpose, an electric generator and load bank were installed to replace the dynamometer; the Woodward PSG governor was replaced by a Woodward EG-3C electronic governor, selected for flexibility of dynamic adjustment and because of its provision for load rate sensing.

The original plan was to test all three combustor systems with identical governor settings and to observe the differences in system response under step load changes. Unfortunately, while the test demonstrated that ammonia fueled combustors can give reasonably good transient performance, the data were inconclusive as regards comparative performance. Test results tended more to indicate control response than relative combustor performance. The transient testing of the ammonia vapor and hydrocarbon systems did confirm the theoretically predicated result that the vapor combustor has relatively poor response because of its large size and low velocities. The catalyst combustor should have demonstrated much better response characteristics because of its smaller size. This was not done, however, since the system could not be optimized sufficiently using the same control components as the vapor combustor system.

2.2.4.1 Vapor Combustor

The vapor combustor was installed and step-load changes up to 120 KW were applied. Predicated on no-load performance, the governor was adjusted to the limits of stability. As indicated in Figure 18, the on-load transient speed deviation was 2.8 percent and the off-load deviation was 3.2 percent. Recovery from the off-load transient was sluggish, possibly because of insufficient minimum fuel margin. However, further reduction of minimum fuel could not be made without incurring flameout on rapid load removal. Also, mismatch and non-linearity of the air-fuel ratio control had a significant effect on response of the system, as was born out by later tests of the catalyst combustor.

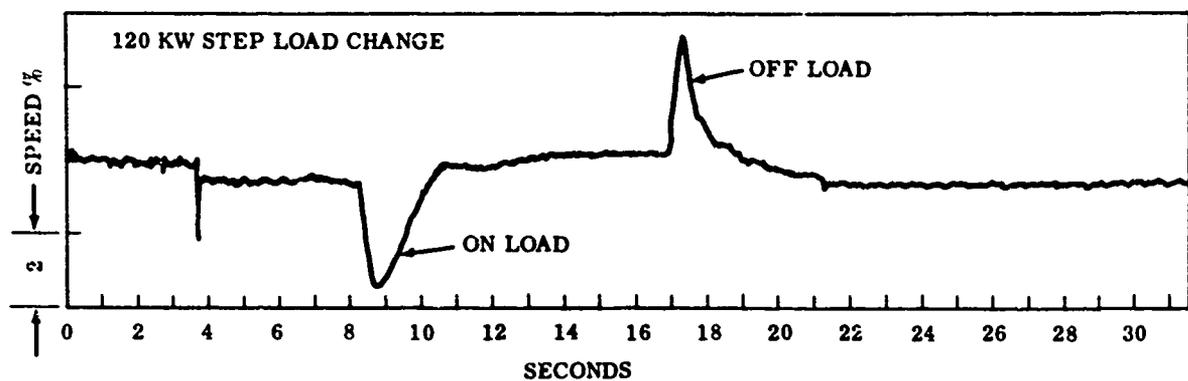


FIGURE 18. RESPONSE CHARACTERISTICS, AMMONIA VAPOR COMBUSTOR

In the ammonia fuel system, the governor operates to reduce fuel by closing the fuel metering valve, which, because of acceleration fuel scheduling requirements, has a highly non-linear contour. This causes a significant increase in fuel flow/governor gain at low flows, and reduces the margin for stability at no-load. Similarly, the butterfly air valve has a non-linear characteristic. If these non-linear features could have been eliminated, it would have been possible to adjust the governor at no-load to a more responsive setting and the transient performance of the vapor burner would have been improved.

As the data in Figure 18 indicate, the governor was not giving true isochronous performance, since a negative steady-state speed droop (from no-load to full-load) of 0.8 percent was observed. This characteristic persisted throughout the test and was considered to be a fault in the governor performance. While the presence of this slight droop clouds interpretation of the data, it is unlikely to have been a significant effect on transient response.

2.2.4.2 Hydrocarbon Combustor

The hydrocarbon system was installed without changing the governor adjustments. Transient response to step-load changes demonstrated significant improvements in speed recovery as shown in Figure 19. On-load speed deviation, with a step of 120 KW, was only 1.7 percent and the off-load deviation was about the same. Recovery time was significantly faster; only one second, compared with two to three seconds for the vapor burner. However, it is not possible to conclude that the improvement was due entirely to the better characteristics of the hydrocarbon combustor. While the tests were conducted with the same governor transfer function, the overall control gain is affected by the hydrocarbon fuel valve and linkage gain as compared to that of the ammonia system. In the hydrocarbon system the fuel flow/generator gain is substantially constant with load.

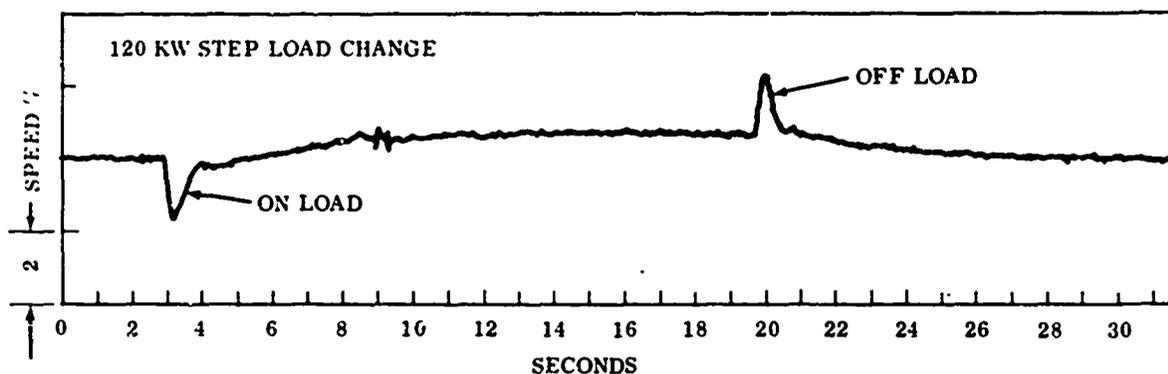


FIGURE 19. RESPONSE CHARACTERISTICS, HYDROCARBON COMBUSTOR

2.2.4.3 Catalyst Combustor

Because of the very high gain of the fuel metering and air control valves near the closed position, it was not possible to completely eliminate the instability at no-load, although the system rapidly became stable with increasing load. Because of this, it was not possible to record comparable system response to step-load transients.

3

ENGINE PERFORMANCE

Engine performance tests were run using the standard engine combustor burning JP-4 hydrocarbon fuel and with the engine converted to operate with the ammonia vapor and catalytic combustors. The tests conducted with the hydrocarbon and ammonia vapor combustors were run over the range 70 to 105 percent corrected engine speed, with loads up to 250 hp. Testing with the ammonia catalyst combustor was limited to operation at 100 percent corrected speed, and due to fuel system problems during the test, loads were limited to 200 hp. All engine data was corrected to sea level, standard day conditions, i. e. , 59°F and 14.7 psia.

The performance of the hydrocarbon burning engine is shown in Figure 20. The level of performance is typical for this engine model. Although the test fuel used was JP-4 with an LHV of 18,700 Btu/lb, all data are shown corrected to diesel equivalent with an LHV of 18,380 Btu/lb. This was done for convenience only since the existing data reduction program included use of diesel fuel.

The ammonia vapor combustor engine performance is shown in Figure 21. The curves show two values of turbine inlet temperature (T_3). The values were obtained by direct measurement and by calculation of work done using the measured value of turbine exhaust temperature, and C_p and γ for the thermodynamic properties of the turbine gases from Figure 22. However, there is a discrepancy in the two values, and it is assumed that the measured temperature is the more reliable. For calculation purposes, the LHV of ammonia vapor fuel was assumed to be 8000 Btu/lb. Engine performance at all speeds is very similar to performance with hydrocarbon fuel.

The performance of ammonia vapor using the catalyst combustor at 100 percent corrected speed is shown in Figure 23. Due to the limited number of data points, the curves are extrapolated from 200 to 250 hp. Again, there is a discrepancy between measured and calculated turbine inlet temperatures.

The performance of the three combustion systems at 100 percent corrected speed is compared directly in Figure 24. All the fuel flows shown in Figure 24 have been reduced to the diesel equivalent by correcting the calorific values for each fuel to reflect equivalent Btu input into the engine. The turbine exhaust temperatures plotted on this curve are all as-measured. The turbine inlet temperature for the JP-4 fuel is calculated from work and measured exhaust temperature because no turbine inlet temperature measurements were taken on this engine. A second procedure for calculating turbine inlet temperature is from the compressor outlet temperature using the calorific

value of the fuel consumed. In the case of the JP-4 hydrocarbon fuel, this method gave excellent agreement with the work calculation. This agreement (to within 1 percent) is quite usual with hydrocarbon fuels because both the calorific values of the fuels and the thermodynamic properties of the combustion products are by now very well known. The validity of this procedure in the case of hydrocarbon fuels has been proven in the past and is confirmed in this case by the good agreement with the standard engine.

For the two ammonia systems, the measured turbine inlet temperature only has been plotted. This is because there was no reason to doubt the validity of the measurements, and there was discrepancy between the temperature calculated from fuel flow and that temperature calculated from work for both the vapor combustor and the catalytic combustor. In the case of the vapor combustor using the calorific value of 8000 Btu/lb, the temperature calculated from fuel flow was in excellent agreement with the measured temperature; but the temperature from turbine work using the thermodynamic properties of Figure 22 (compiled from Ref. 1 and 2) gave a discrepancy of about 2 percent between temperature from fuel and work. In the case of the catalytic combustor, no good agreement could be obtained between the measured temperature and the temperature from fuel flow using the calorific value of 8000 Btu/lb except at the higher loads; the temperature from work was never closer than 2 percent to that from fuel.

Accepting the validity of the temperature data plotted in Figure 24, the most significant difference between the ammonia and hydrocarbon results is the lower turbine inlet temperature at rated load using ammonia fuel. This confirms predictions included in the Phase I report that using ammonia fuel at approximately 2.35 times the rate of diesel fuel would result in cooler turbine temperature for a given horsepower. The gain in horsepower output for the same turbine inlet temperature was about 9 and 13 percent for the vapor and catalyst combustors respectively. The prediction of 10 to 15 percent would undoubtedly have been achieved if the maximum load had been increased slightly to cause an increase in the engine maximum turbine inlet temperature.

At full load, with either ammonia combustor, there is an improvement in diesel equivalent fuel flow from 243 to 237 pph. This reflects an apparent improvement in thermal efficiency of about 2.5 percent.

In viewing the data, there is an obvious difference in the combustion process and the thermodynamic properties of the combustion products of the ammonia fuel, depending on type of combustor. This is indicated by the difference in turbine inlet temperature which is about 80°F higher at no-load, with the catalytic combustor becoming equal to that for the vapor combustor as the load is increased to about full-load. Also, the turbine temperature drop for equal work is greater with the catalytic combustor over the whole load range. The apparent reason for this and for the increased power capability of ammonia fuel for a given turbine inlet temperature is the difference in the thermodynamic properties of the working fluid.

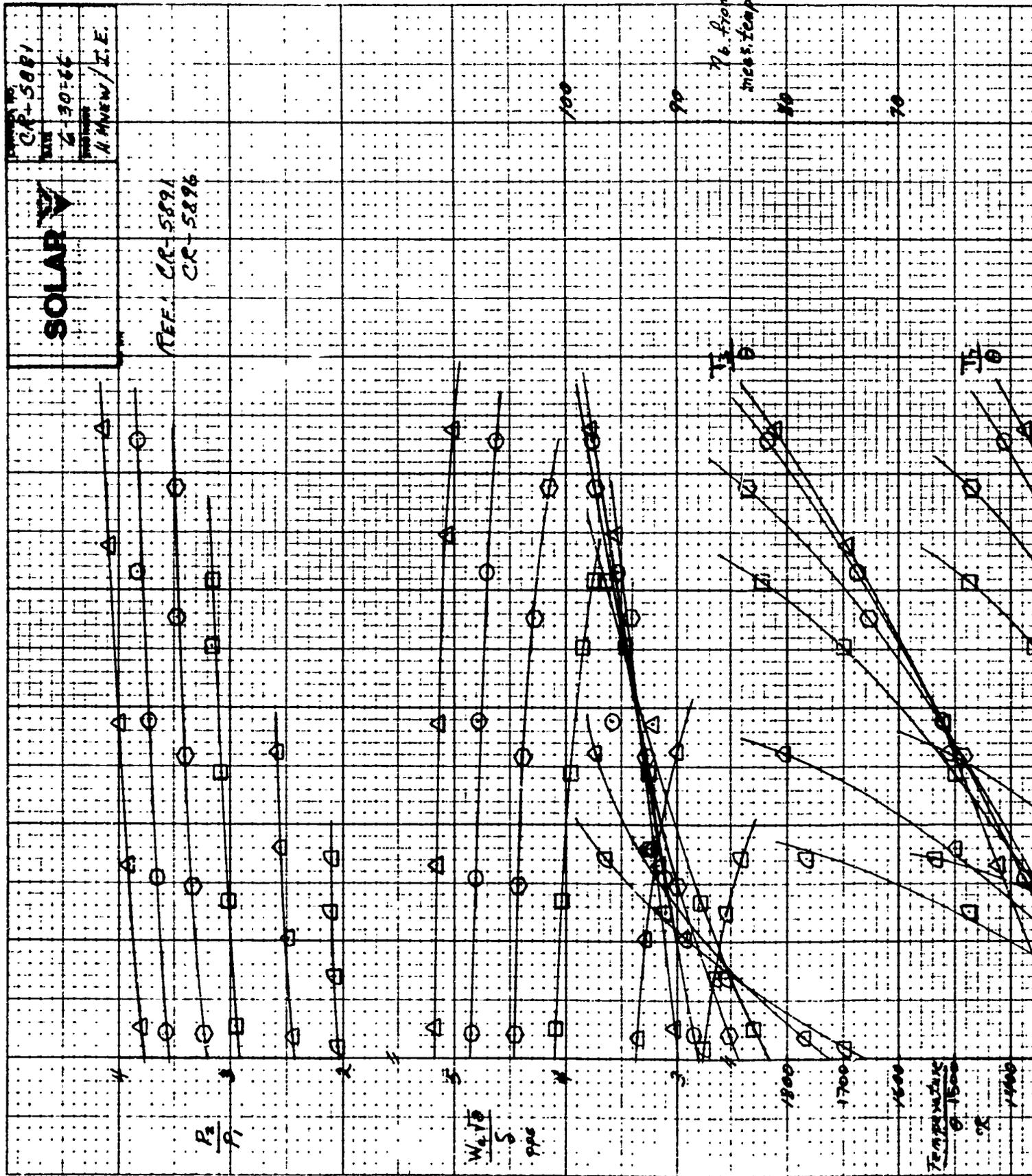
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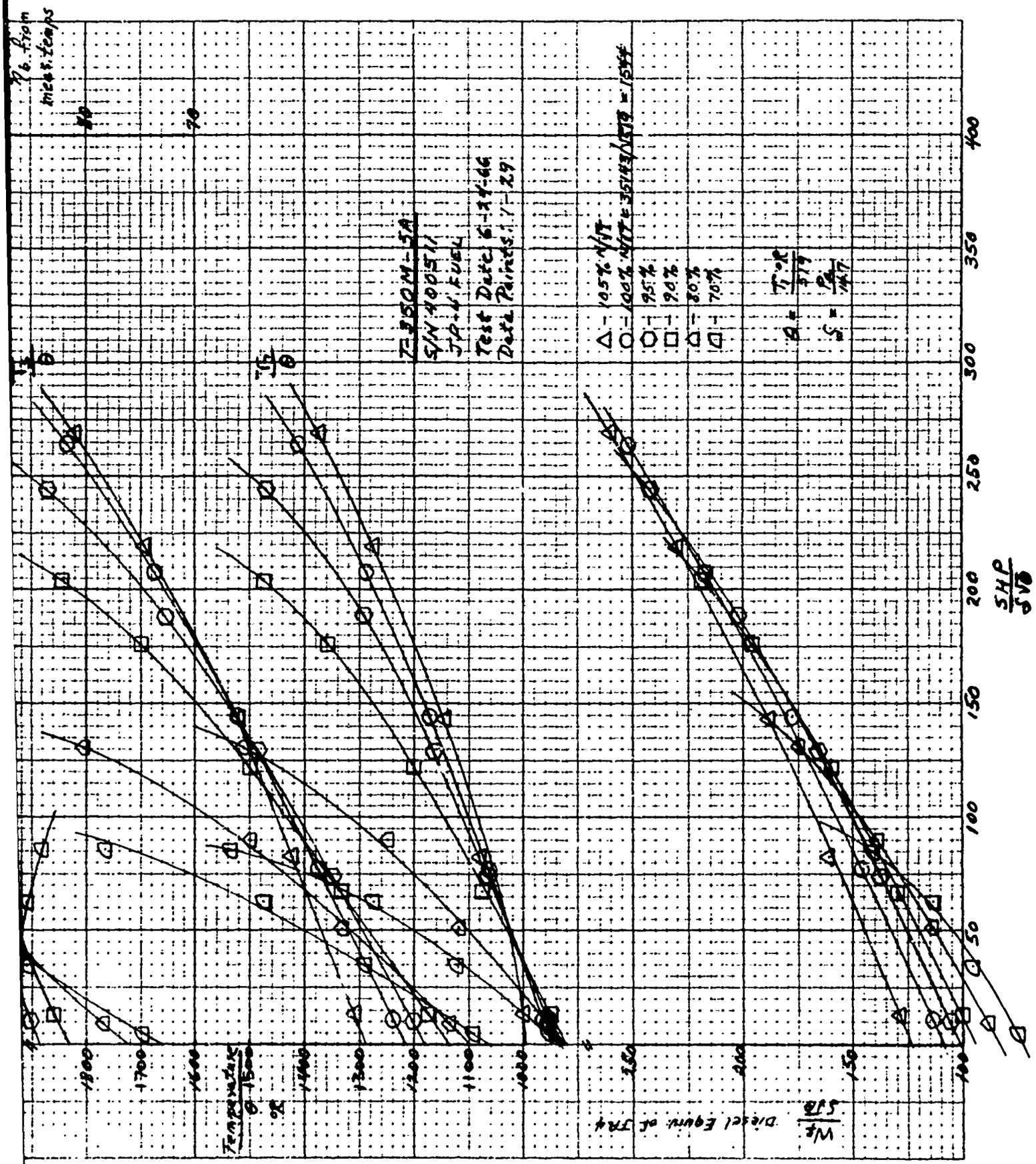
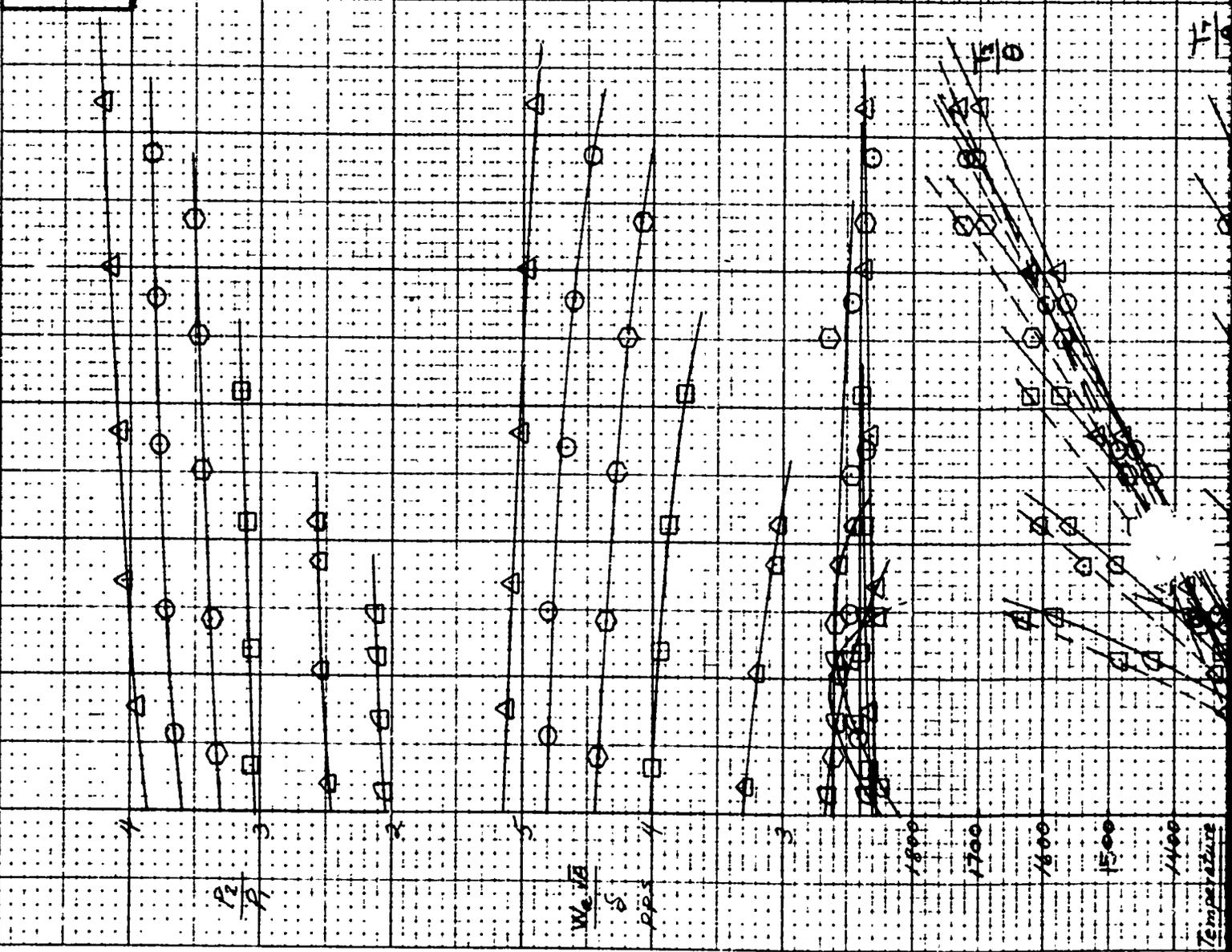


FIGURE 20. ENGINE PERFORMANCE.
HYDROCARBON FUEL

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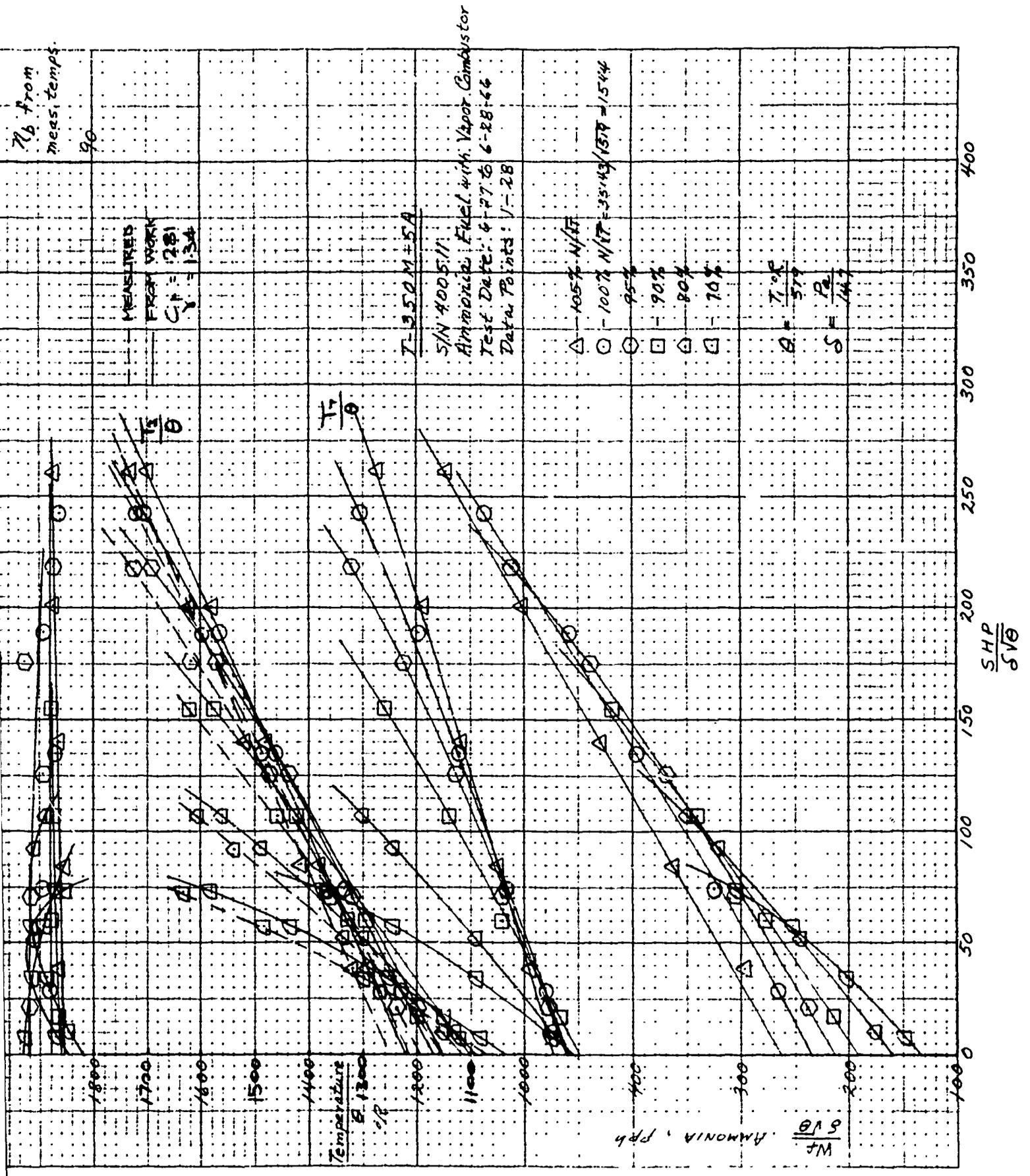


FIGURE 21. ENGINE PERFORMANCE, AMMONIA VAPOR COMBUSTOR

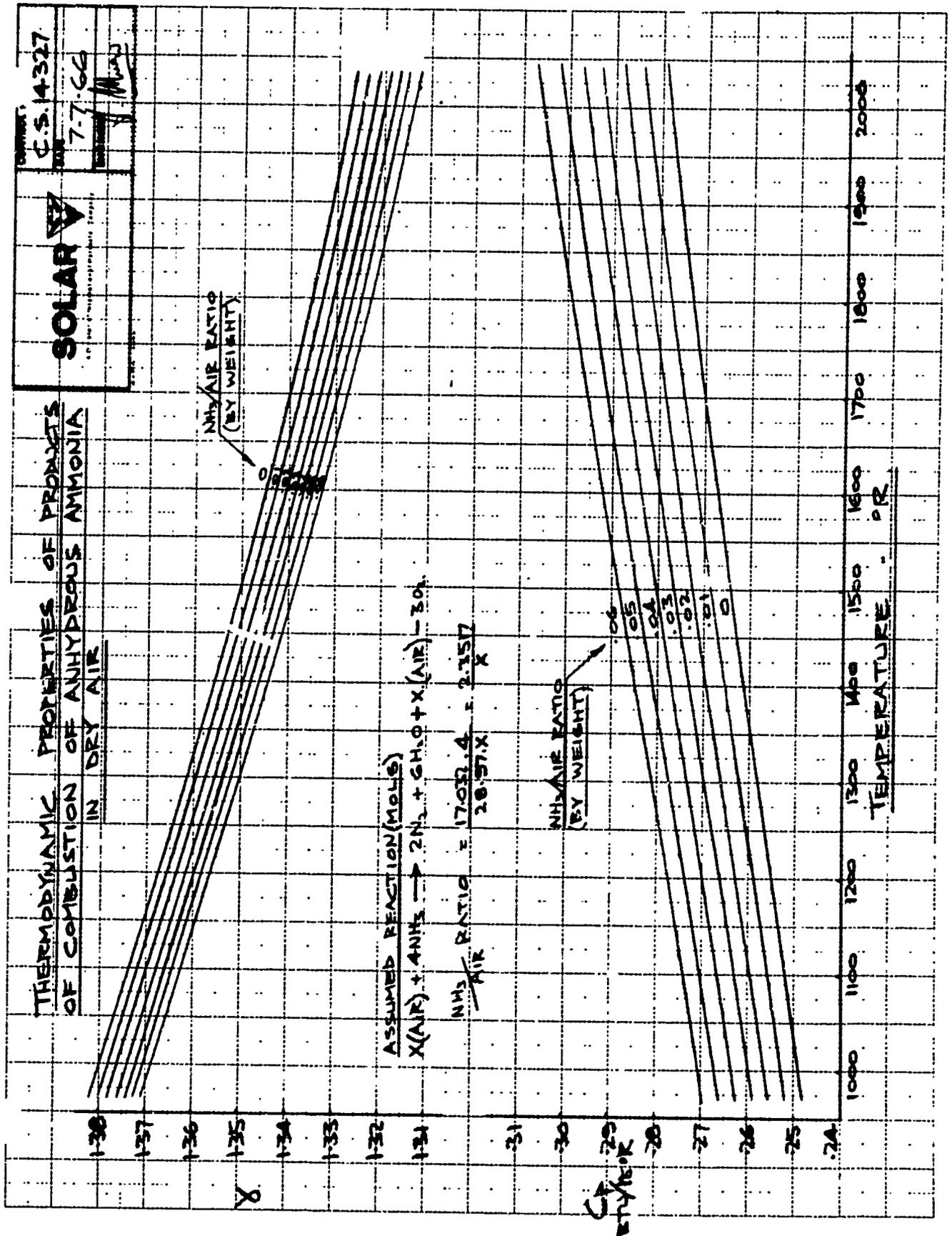
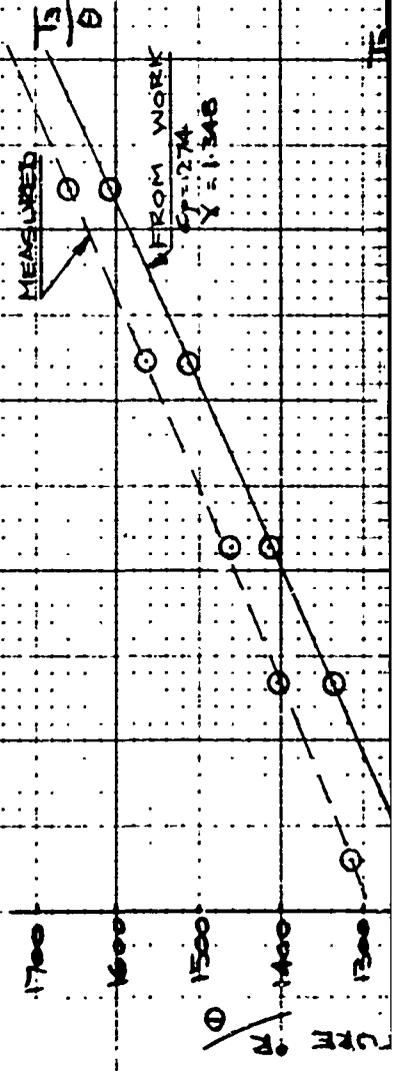
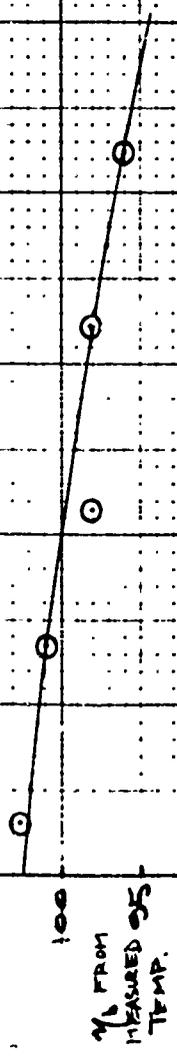
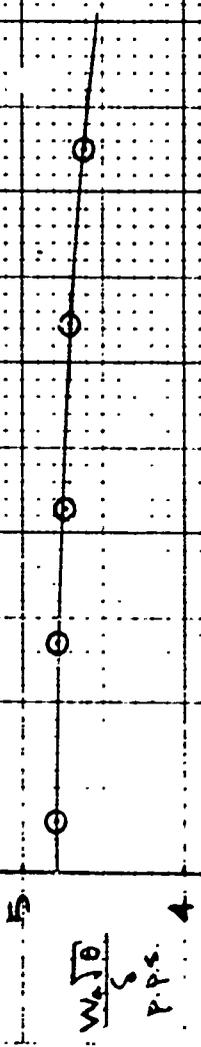


FIGURE 22. THERMODYNAMIC PROPERTIES, ANHYDROUS AMMONIA
 (Data From Ref. 1 and 2)

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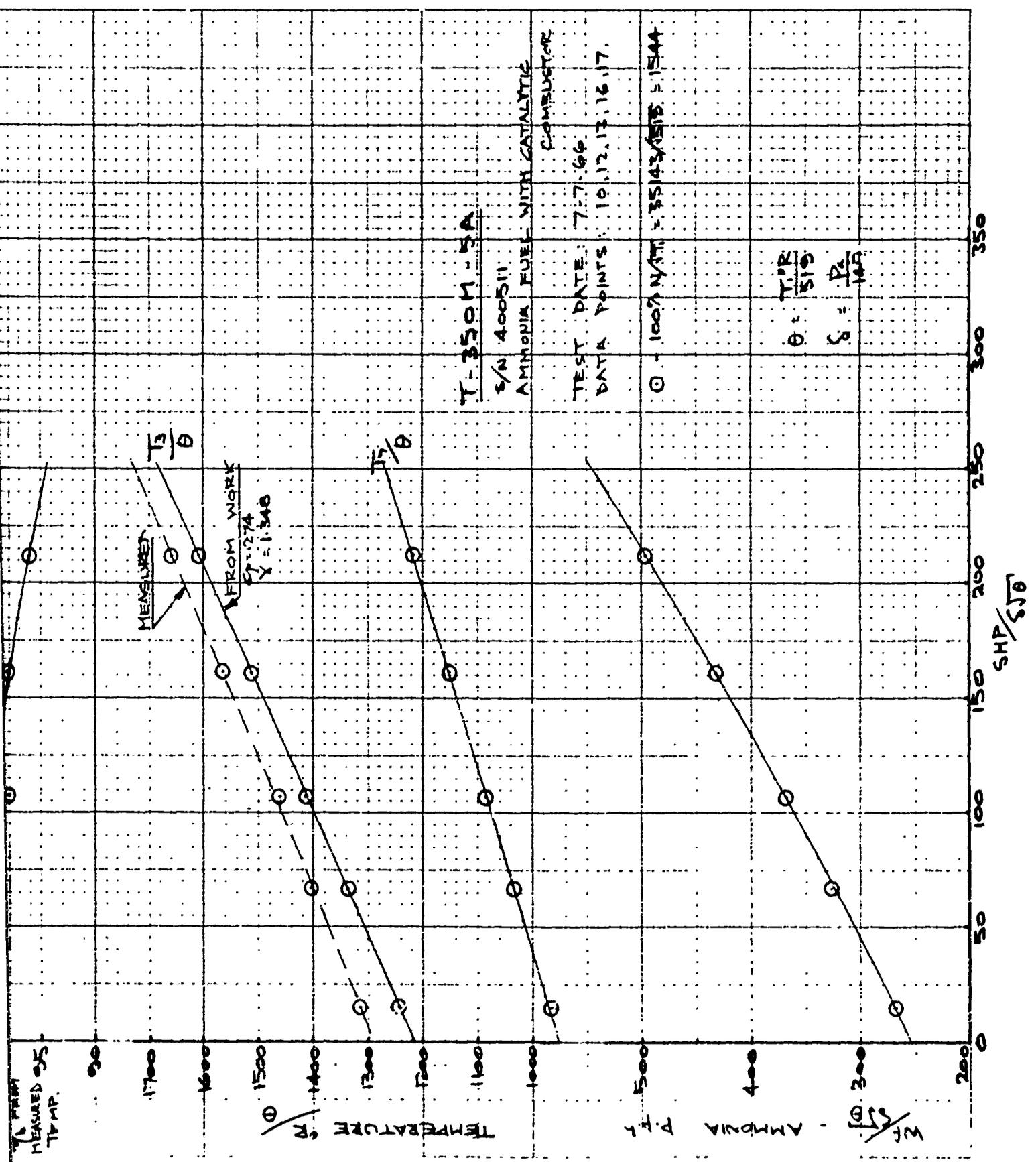
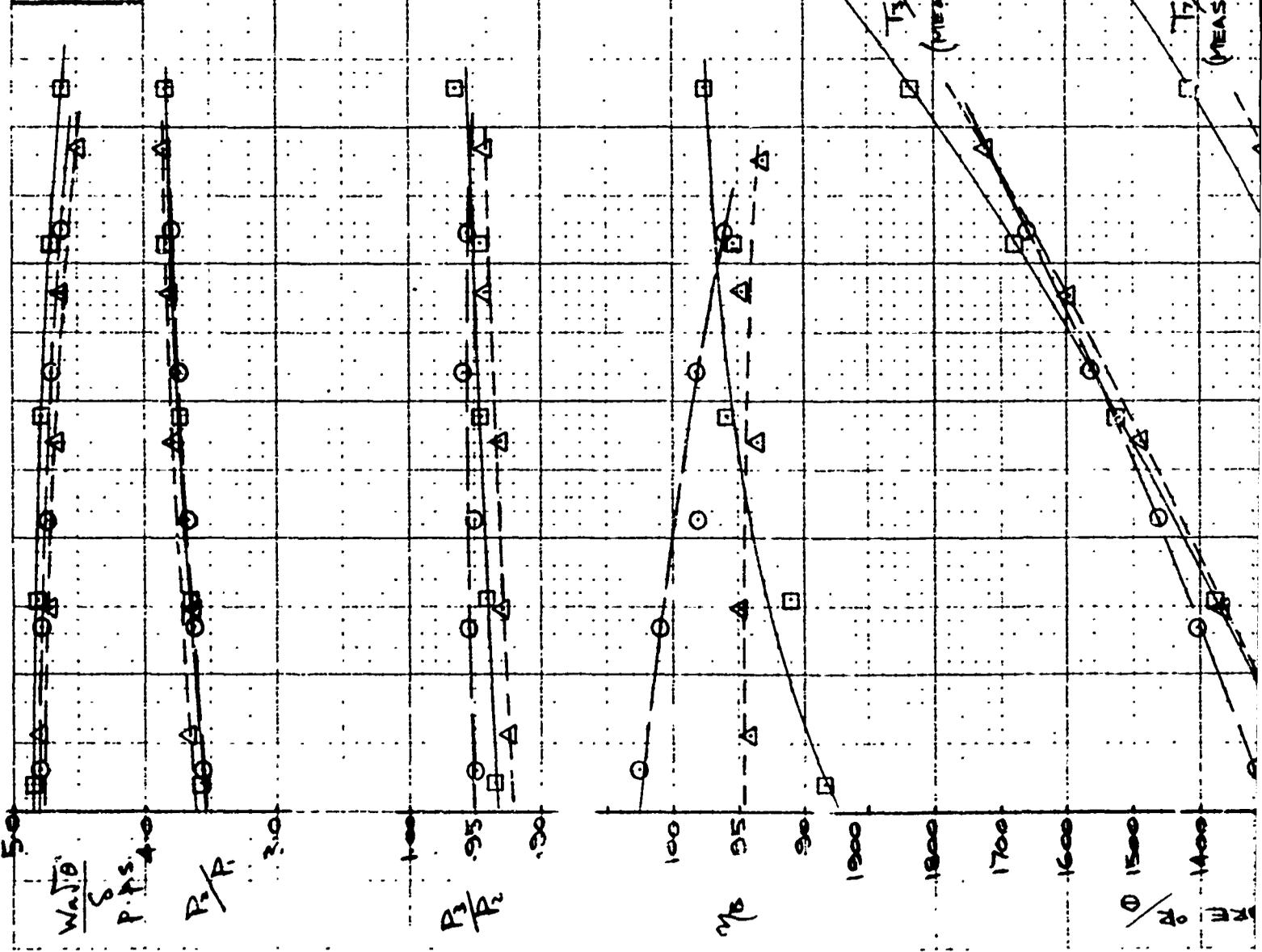


FIGURE 23. ENGINE PERFORMANCE. AMMONIA CATALYTIC COMBUSTOR

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 COMPARISON OF
 PERFORMANCE USING
 HYDROCARBON AND
 AMMONIA FUEL

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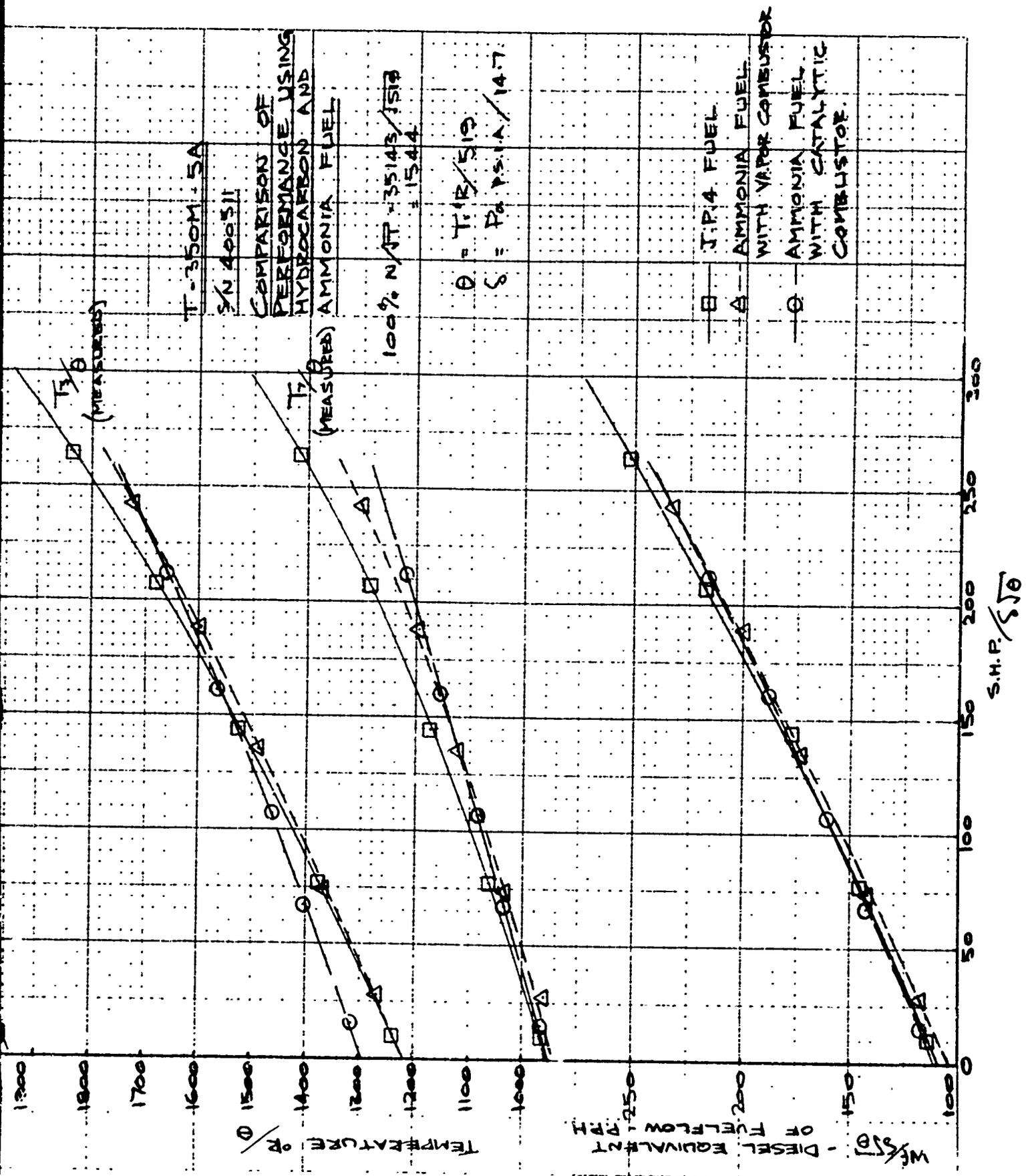


FIGURE 24. T-350 GAS TURBINE ENGINE PERFORMANCE

At 100 percent corrected speed the flow through this engine is controlled by the choked turbine nozzle. The flow parameter at choke is given by the equation:

$$Q_{\text{choke}} = (W_T \sqrt{T_3} / A_N P_3)_{\text{choke}} = \sqrt{\frac{g}{J}} \cdot \frac{1}{\sqrt{C_p}} \cdot \frac{\gamma}{\sqrt{\gamma+1}} \left(\frac{2}{\gamma+1} \right)^{\frac{\gamma+1}{2(\gamma+1)}}$$

Tabulating for each combustor system we get:

	<u>Hydrocarbon</u>	<u>Ammonia Vapor</u>	<u>Ammonia Catalytic</u>
C_p	0.276	0.281	0.274
γ	1.33	1.34	1.348
Q choke	0.523	0.514	0.520
Q choke / (Q choke) hydrocarbon	1.00	0.983	0.994

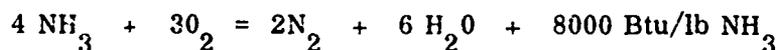
These ratios should be reflected in the test results which may also be tabulated from Figure 24 at the 250 hp point as follows:

	<u>Hydrocarbon</u>	<u>Ammonia Vapor</u>	<u>Ammonia Catalytic</u>
W_a pps	4.65	4.50	4.56
W_f (diesel equivalent) pph	244.0	237.5	237.0
W_f (actual) pps	0.0664	0.1518	0.1516
$W_T = W_a + W_f$ pps	4.7164	4.6518	4.7116
T_3 °R	1792	1740	1730
P_2/P_1	3.82	3.85	3.3
P_3/P_2	0.954	0.942	0.95
$P_3 = 14.7 \times \frac{P_2}{P_1} \times \frac{P_3}{P_2}$ psia	53.6	53.3	53.05
$W\sqrt{T_3}/P_3$	3.722	3.64	3.695
$(W\sqrt{T_3}/P_3) / (W\sqrt{T_3}/P_3)$ hydrocarbon	1.0	0.978	0.993

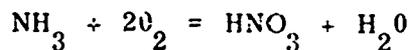
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The preceding agreement between the estimated turbine flow capacities and the actual capacities achieved on the test indicates that the thermodynamic properties of the ammonia combustion products have been reasonably estimated and are different for the two types of combustors. An explanation is also given for the differing compressor matching points along the constant speed line. Thus, the result of the ammonia fuel was to reduce the effective engine capacity by about 3 percent on compressor flow with the vapor combustor, and about 2 percent on compressor flow with the catalytic combustor. Had the turbine nozzle area been opened by corresponding amounts to get equal compressor flows for all three cases, then the two ammonia cases would have matched even cooler at the 250 hp condition. It would be impossible to equalize both compressor massflow and pressure ratio by changing the nozzle area alone because of the differences in combustor pressure loss; therefore, attainment of the full theoretical gain is probably impracticable.

The combustion efficiencies of the ammonia fueled engines are based on an assumed LHV of 8000 Btu/lb. The ammonia vapor combustor gave almost constant combustion efficiency (93.5 to 94.5 percent) over the complete load range; and, taking this, together with the good agreement of the temperature calculated from fuel flow with measured temperature, it can be concluded that the vapor combustor is behaving reasonably close to the chemical equation:



At the no-load end of the catalytic combustor curve, however, the combustor efficiency is above 100 percent and the corresponding measured turbine inlet temperature is some 80°F higher than that for the vapor combustor for almost equal compressor flow and compressor exit temperatures. These facts, taken in conjunction with the poor agreement between measured and calculated temperatures for the catalytic combustor, indicate that the calorific value in the case of the catalytic combustor process is somewhat higher than 8000 Btu/lb. M. Bodenstein (Ref. 3) has shown experimentally that the molecules HNO₂ and HNO₄ are created during the catalytic oxidation of ammonia, and the presence of these large molecules would imply a large calorific value. It is doubtful if HNO₃ can exist in the combustor, but if it could, the reaction:



would give a calorific value of 8680 Btu/lb of ammonia. Similarly the presence of HNO₂ and HNO₄ would give increased calorific values thus explaining the discrepancy at the no-load end of the curves.

A further observation is that for equal quantities of work, the temperature drop through the turbine is greater for the catalytic combustor than for the vapor combustor. This fact, for equal massflow, would imply lower values of specific heat for the turbine gases of the catalytic engine which would occur if oxides of nitrogen were present in these turbine gases. The occurrence of such oxides would not be in contradiction to the formation of HNO_2 and HNO_4 in the combustor because these molecules eventually break down to oxides of nitrogen which then recombine (in the presence of water) to form HNO_3 (nitric acid). The removal of free water molecules from the combustion products would tend to reduce specific heat.

Thus, a possible explanation of the difference in behavior between the vapor and catalytic combustor is the creation of larger molecules of nitrogen compounds in the catalytic combustor with subsequent breakdown and recombination to nitric acid in the exhaust gases. The effect is to increase the calorific value of the ammonia fuel and to reduce the specific heat of the combustion products relative to the values obtained with the vapor combustor. The effect on calorific value reduces as load increases (i. e. as the fuel-to-air ratio increases), there being no detectable difference between vapor and catalytic combustion at the full-load condition. The effect on specific heat, however, perpetuates over the complete range of loading. This hypothesis could easily have been proved or disproved by exhaust gas sampling; the catalytic combustor exhaust gases should produce significantly more nitric acid when bubbled through water than the vapor combustor gases. It is not known if this effect is solely due to the size, shape, and operating conditions of the particular catalytic combustor used on this engine, but the chances are (since this is the common Ostwald process of producing nitric acid) that these effects will occur at some operating condition on any ammonia fueled engine using the catalytic oxidation combustion process.

NOMENCLATURE FOR SECTION 3

<u>Symbol</u>	<u>Item</u>	<u>Unit</u>
A	Surface Area	in ²
C _p	Specific Heat at Constant Pressure	Btu/lb deg F
g	Gravitational Constant	ft/sec ²
J	Mechanical Equivalent of Heat	ft lb/Btu
N	RPM	
P	Stagnation Pressure	psia
Q	Flow Parameter	°R/εsec
T	Stagnation Temperature	deg R
W	Mass Flow Rate	pps or pph
ΔT	Stagnation Temperature Change	deg R
θ	Ratio of Engine Inlet Temperature to 519°R	
γ	Ratio of Specific Heat at Constant Pressure to Heat at Constant Volume	
δ	Ratio of Engine Inlet Pressure to 14.7 psia	
η	Efficiency	

SUFFIXES

<u>Symbol</u>	<u>Item</u>
a	Ambient or Air
b	Burner
c	Compressor
f	Fuel
n	Nozzle
t	Turbine
1	Engine Inlet
2	Compressor Exit
3	Turbine Inlet
7	Engine Exhaust
choke	Choking Condition

4

DISCUSSION

It has been shown that combustion of ammonia is more difficult than hydrocarbon fuels; however, it can be satisfactorily used as a gas turbine fuel providing the combustor size is increased. The larger combustor increases the combustion time and introduces limitations for applications requiring fast response (for example, precise frequency control with gas turbine powered generating sets). The test results do not reflect the optimum response of the ammonia vapor combustor however, since the control system dynamics were not optimized. A valid quantitative comparison of the transient performance of the different combustion systems would require frequency response testing during engine operation. It would then be possible to optimize the control dynamics for each individual system.

To conduct such tests as a part of the program was not possible due to the requirement for interchangeability of the three combustion systems which dictated extensive use of common control elements. Acceleration fuel schedules had to be selected in such a way that both the ammonia vapor and catalyst configurations would perform with the same metering valve contour and pressure regulator setting. Adjustments to the schedule needed to compensate for differences in efficiency were accomplished by modifying fuel and air pressure drop characteristics. The air control valve, which performed acceptably well for the ammonia vapor combustor, was forced to operate nearer the closed position with the catalyst combustor. The excessive valve gain near the closed position created instability and thus prevented any evaluation of the true catalyst system response.

For systems requiring a fast response there is still reason to believe that the catalyst combustion system could be developed to the degree necessary to provide performance comparable with the best hydrocarbon combustion systems.

The performance of the control system was undoubtedly the limiting factor in the engine stability problems encountered and not the response of the combustor itself. However, before applying the catalyst combustion system to any application, a considerable amount of development would be required. This is hardly surprising in view of the fact that the concept is completely different from anything previously attempted with a gas turbine engine. The major effort would necessarily be to achieve acceptable reliability. The most promising approach would seem to be by attempting

to lower the catalyst bed operating temperature. The high operating temperature was dictated by the fact that flame-out occurred in the secondary zone when the fuel-air ratio allowed the catalyst operating temperature to fall below 1550° F. Minor changes made to the secondary zone baffle had little or no effect on this blow-out limit. The conditions for secondary zone combustion, high temperature, and partially decomposed ammonia seem so ideal that it is surprising that such a high temperature was required. If the secondary zone combustion limit could be developed to operate at temperatures of 1000° F, the limiting maximum temperature could be reduced to 1500° F and catalyst reliability would be significantly improved. A wider temperature control range would also simplify control system requirements with a consequent increase in reliability.

For systems where optimum combustor response is not dictated, the ammonia vapor combustor offers an adequate solution. However, the investigation only involved a can combustor, typical of small industrial engines. In aircraft engines such a combustion system is not used because of penalties in weight, bulk, and performance, and the use of an annular combustion system is almost mandatory. The operating conditions of typical annular systems appear more suitable for ammonia operation than the can systems. A comparison of the T-350 engine can and annular combustor design parameters versus the 12-inch diameter ammonia can combustor is shown below:

	<u>Can Combustor</u> <u>(Hydrocarbon)</u>	<u>Can Combustor</u> <u>(Ammonia)</u>	<u>Annular Combustor</u> <u>(Hydrocarbon)</u>
Combustor Cross Section (in. ²)	32.2	113	170
Combustor Volume (in. ³)	34.0	1870	2120

The annular hydrocarbon combustor is considerably larger than the ammonia-fueled can combustor, allowing 50 percent lower velocities and 13 percent greater residence time. This particular annular design is typical of a small industrial or APU engine design. Based upon size alone, it seems probable that many annular combustor engines could be operated on ammonia fuel without any increase in combustor size.

5

CONCLUSIONS

1. Ammonia is a satisfactory substitute for hydrocarbon fuels in a simple-cycle gas turbine engine. Two different combustion systems, using ammonia in the vapor state, demonstrated satisfactory combustion performance in a converted simple-cycle, single can burner, gas turbine engine.
2. Rig test data developed for ammonia vapor combustor performance are satisfactory for use in design of engine combustors. Conversion of single can combustor engines to ammonia fuel operation can be accomplished without major redesign and development effort.
3. The complexity and cost of basic engines using ammonia vapor combustors will not be significantly greater than existing hydrocarbon burning engines. The addition of exhaust heat ammonia vaporizing systems to obtain system self-sufficiency will significantly increase size, cost, and weight.
4. Further development of the ammonia catalyst combustion system is required before accurate assessment of its response characteristics and potential operating reliability can be determined.
5. Power increases in the order of 10 to 20 percent can be extracted from ammonia burning engines relative to hydrocarbon burning engines at similar turbine inlet temperatures. The fuel consumption of ammonia, relative to hydrocarbon at a similar power output, is approximately 2.25 times greater by weight and 2.82 times greater by volume.

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1. Patterson, D. J. and Van Wylen, G. J., "Empirical Heat Capacity Equations For Ideal Gases." Trans ASME, Vol. 85, Series "C" (Aug. 1963) p. 281.
2. Tables of Thermal Properties of Gases. National Bureau of Standards Circular No. 564 (1955).
3. Bodenstein, M., Z. Elektrochem, 47, (1941)

END