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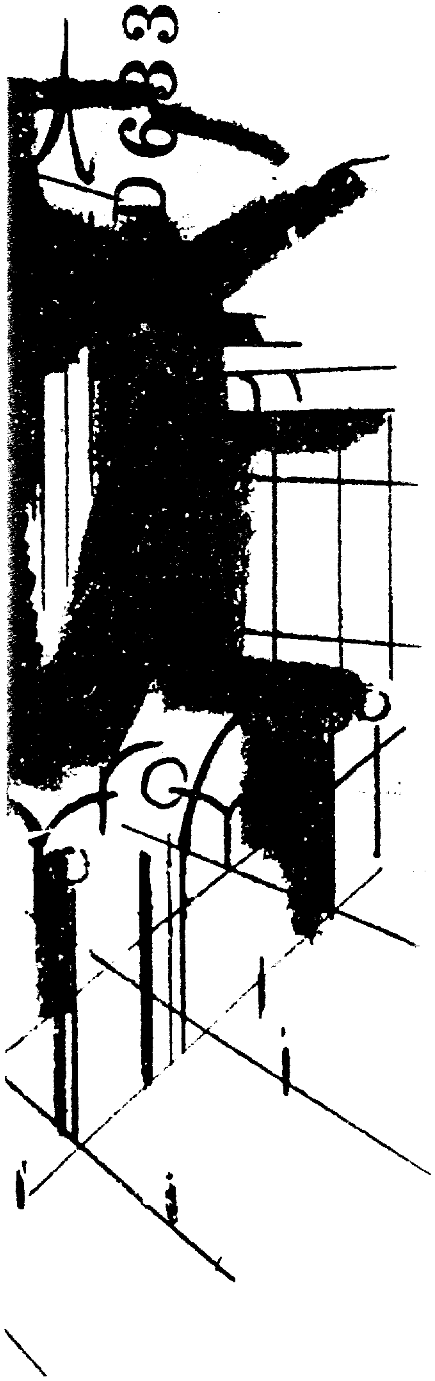
D. S. Smith, et al

California University
Berkeley, California

February 1966



PERFORMANCE OF ANHYDROUS AMMONIA AS A SPARK IGNITION ENGINE FUEL

by

D. S. Smith
E. S. Starkman

Report No. TS-66-2
Contract DA-04-200-AMC-791 (X)
February 1966

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PERFORMANCE OF ANHYDROUS AMMONIA AS A SPARK IGNITION ENGINE FUEL

by

D. S. Smith

E. S. Starkman

Technical Report No. 6

Under Contract
DA-04-200-AMC-791(X) Ammonia Fuel
Army Material Command
R & D Directorate
Chemistry and Materials Branch

Project Director
E. S. Starkman

February, 1966

33240

SUMMARY

The operating characteristics of anhydrous ammonia as a fuel for a spark ignition engine were determined and compared, experimentally and theoretically, to those of iso-octane, the reference hydrocarbon fuel. Experimental data obtained over a wide range of parameters--compression ratio, engine speed, manifold pressure--displayed no grossly divergent fuel characteristics for ammonia compared to iso-octane.

Anhydrous ammonia was demonstrated to operate successfully as a fuel for spark ignition engines. Principal requirements are that it be introduced into the engine in the vapor phase and be partly dissociated to hydrogen and nitrogen ($NH_3 \xrightarrow[\text{catalyst}]{\text{heat}} NH_3 + H_2 + N_2$). Dissociation was accomplished in a temperature controlled catalytic dissociator using pelletized activated iron catalyst. The optimal weight concentration of hydrogen to insure against power loss is approximately 5 per cent.

The maximum theoretically possible indicated output using ammonia vapor when adjusted for 5 per cent hydrogen dissociation is about 75 per cent of that with hydrocarbon. The presently obtained maximum experimental level is 72 per cent at a compression ratio of 10.

Specific fuel consumption, both by theory and experiment, is twofold at maximum power and 2-1/2 fold at maximum economy when using ammonia as a replacement for hydrocarbon.

Spark timing for maximum performance must be advanced slightly for ammonia but sensitivity to spark timing is very little greater than with hydrocarbons.

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I. INTRODUCTION

Investigation of the operating characteristics of anhydrous ammonia as a fuel for spark ignition engines and comparison with a reference hydrocarbon, iso-octane, was conducted under Contract DA-04-200-AMC-791(X) AMMONIA FUEL, Army Material Command, R & D Directorate, Chemistry and Materials Branch. The general objective was to extend the area of applied information relative to successful substitution of gaseous, predissociated ammonia for hydrocarbons as a fuel for spark ignition engines. The reasons for renewed interest in ammonia as a fuel and a review of early success in its application are presented in the SAE literature.(1)* The experimental investigation reported here was initiated in August of 1964 and completed in August of 1965.

The goal of the experimental work, carried out in a single cylinder CFR engine, was to point the way to successful operational application and at the same time to do this with the minimum of engine modification. Thus, the investigation was pursued using nominal and existing engine compression ratios, normal ignition systems, atmospheric or throttled manifold pressures and ammonia dissociation methods well within practical capabilities of a military application.

The purpose of this report is twofold; to present the comparative experimental results and to compare the experimental results with the previously published theoretical predictions(2) which were based on thermodynamic calculations of the relative performance to be expected when operating a spark ignition engine on gaseous ammonia and iso-octane.

* Numbers in parenthesis designate references.

II. EXPERIMENTAL FACILITIES AND EQUIPMENT

The CFR engine test facility and a schematic diagram of the test system for gaseous ammonia are shown in Figures 1 and 2. They provide a general orientation to the physical arrangement of the test facilities and show the significant flow processes.

1. Engine and Dynamometer

A CFR single cylinder engine [3] of 3-1/4" x 4-1/2" dimensions, the primary research apparatus, was used for obtaining all of the experimental data. The engine was well suited for this combustion research because of its small fuel requirement and the ease with which the test variables could be changed, controlled, and observed. Cooling jacket temperatures were fixed by evaporation and recycle of coolants--a jacket temperature of 152°F was maintained for iso-octane by the use of methanol, ethylene glycol was used to maintain a jacket temperature of 354°F for ammonia. The usual "supercharge method" instrumentation was provided in order to facilitate close control and accurate measurement of air pressure and flow rate as shown in Figure 3. Engine speed was monitored by an electronic counter actuated by a magnetic pick up.

The engine was connected to a Sprague electric dynamometer capable of a wide speed range for motoring or power absorbtion. The d.c. output from the dynamometer was dumped into a resistance bank with fine adjustment controls for accurate speed control and power output measurements.

2. Ammonia Supply and Metering

A schematic diagram of the ammonia fuel supply system is shown in

Figure 4. Ammonia vapor was drewn from a refillable high pressure bottle maintained in a tempering tank of water. Constant temperature of the bath and replacement of the ammonia latent heat were affected by supplying submerged steam to the water bath. Primary measurement and calibration of ammonia flow were accomplished with volume calibrated sight glass on the ammonia bottle. Continuous flow rate determinations of ammonia were made with turbine flow meter with data read-out from an electronic counter. Suitable regulators and needle valve were installed to reduce the pressure and meter the ammonia flow. The pressure and temperature of the ammonia were monitored at several points in the fuel supply line. Introduction to the engine was by simple standpipe in the intake manifold.

3. Ammonia Dissociator

Partial decomposition of the ammonia vapor was carried out in a 266 cubic inch capacity stainless steel catalyst chamber. This was loosely filled with a pelletized activated iron catalyst. Two kilowatts of electrical resistance element heaters of the globar type were submerged in the catalyst bed.

A limitation of the catalytic dissociator was its very large thermal lag. It was difficult to quickly adjust the temperature or electrical energy supply to a predetermined level of decomposition. Future considerations of similar dissociators should incorporate a more flexible and rapid means of control of the ammonia decomposition. In addition, there was inadequate dissociation capacity to supply the engine with sufficient hydrogen at an engine speed of 1800 rpm for rich mixtures (this condition is discussed extensively in a later section).

4. Iso-Octane Supply and Metering

Iso-octane was supplied to the engine by the standard CFR pump and injector system.

Metering was accomplished by means of an electrically controlled U-tube type meter shown in Figure 5. This type of meter has the advantage of not requiring buoyancy effect corrections as does the ASTM bean balance method. Water in the left leg is balanced at an immiscible inner face by the iso-octane in the right leg. A calibrated weight of fuel is withdrawn from the meter as the water level drops from electrode B to electrode C. The water serves as a conductor between the electrodes and the breaking of the miniscus on the various electrodes actuates the clock and solenoid values thus affecting the operations of timing, draining, and filling.

5. Gas Chromatograph

A Beckman GC-2 gas chromatograph was used to determine the hydrogen dissociation at engine inlet. The sample was withdrawn by hypodermic syringe from the fuel line sample septum shown in Figure 2 and injected into the chromatograph.

For hydrogen detection a stainless steel molecular sieve (Beckman Column 70015) was used. Nitrogen was used as the carrier gas to make the chromatograph sensitive to hydrogen and quite insensitive to ammonia.

The chromatograph was calibrated on the peak height basis using laboratory prepared samples consisting of ammonia, nitrogen and hydrogen in proportions comparable to the range of dissociation products from the catalytic dissociator.

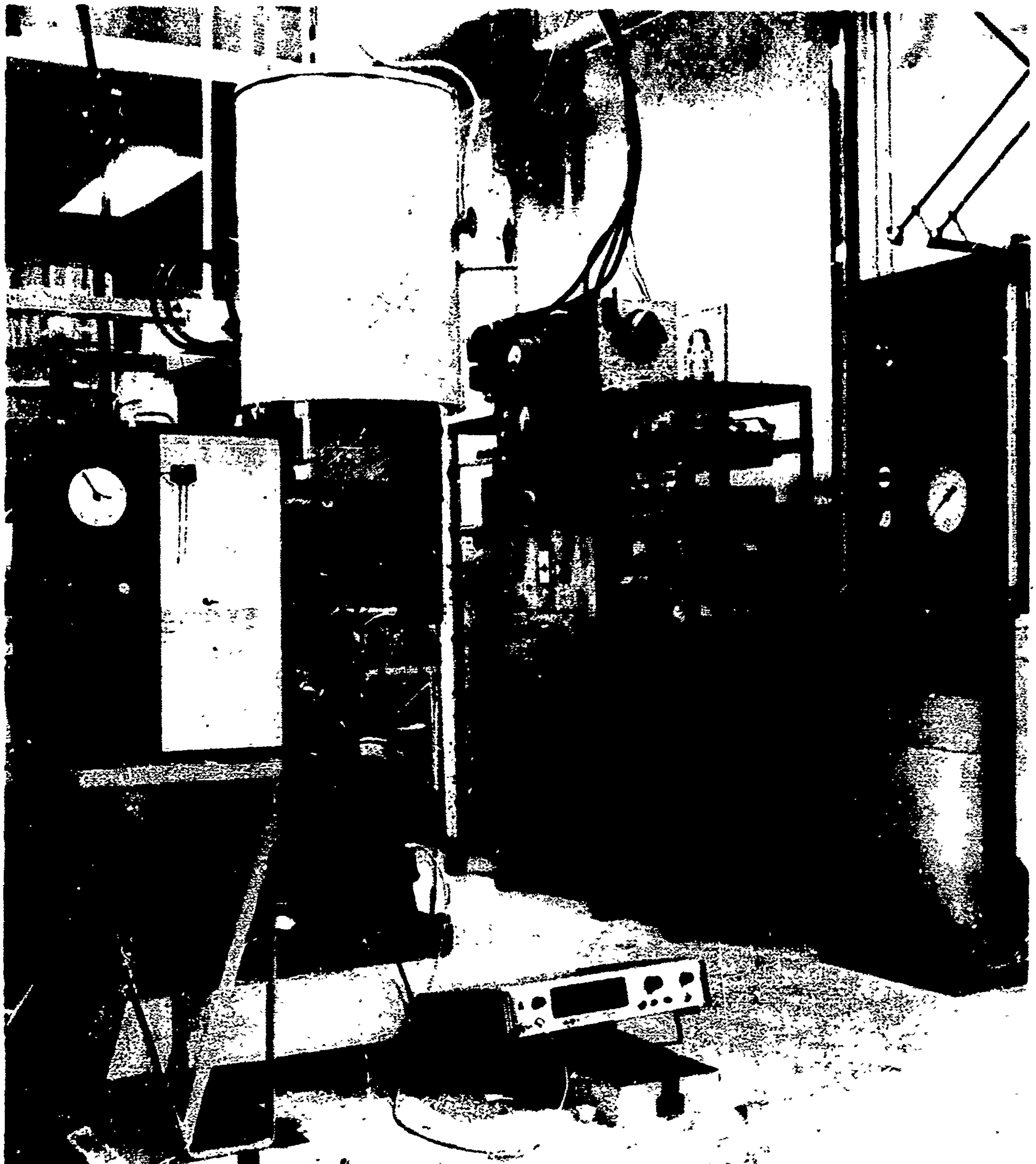


FIGURE 1. CFR ENGINE TEST FACILITY

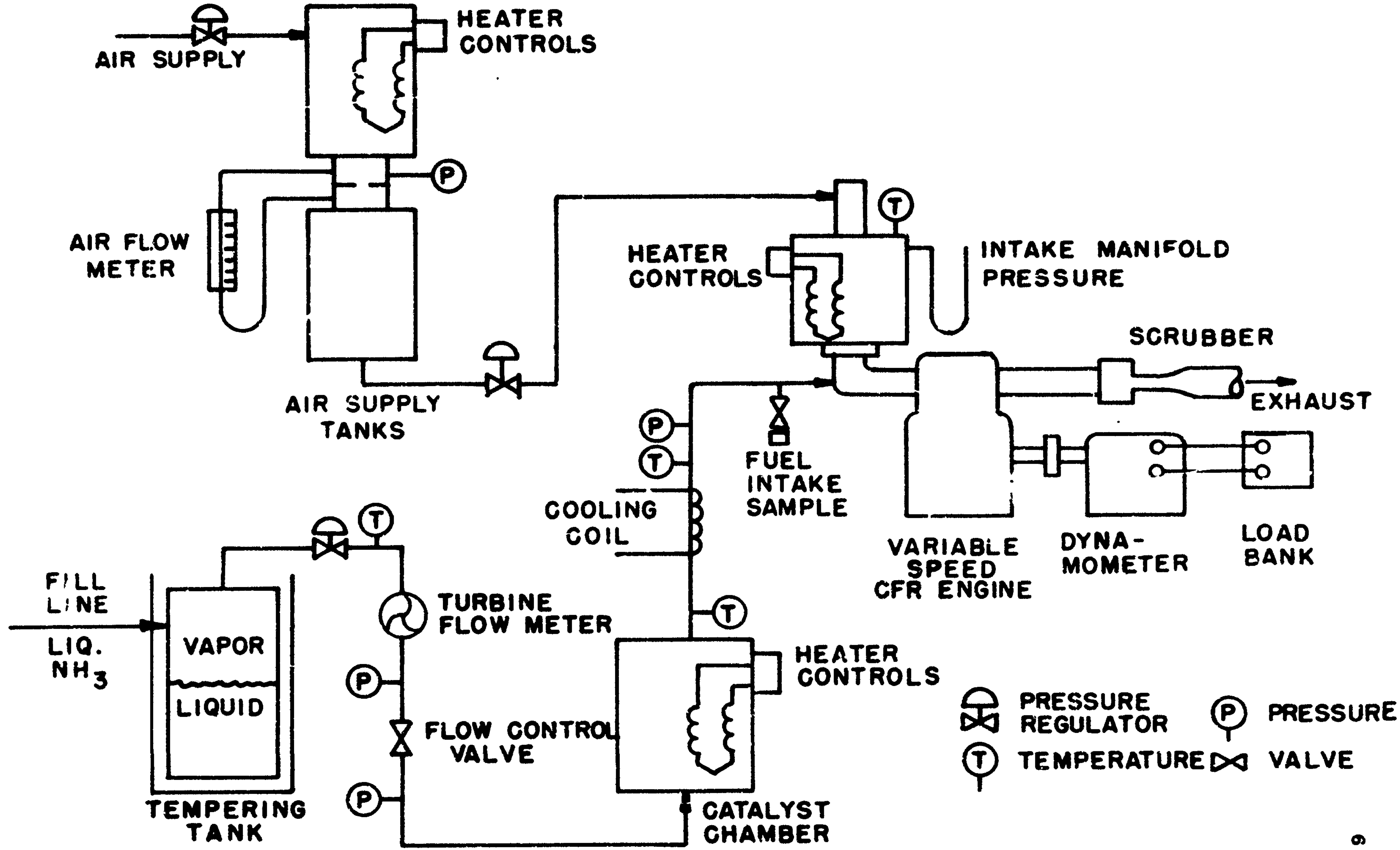


FIGURE 2. SYSTEM SCHEMATIC DIAGRAM - GASEOUS AMMONIA

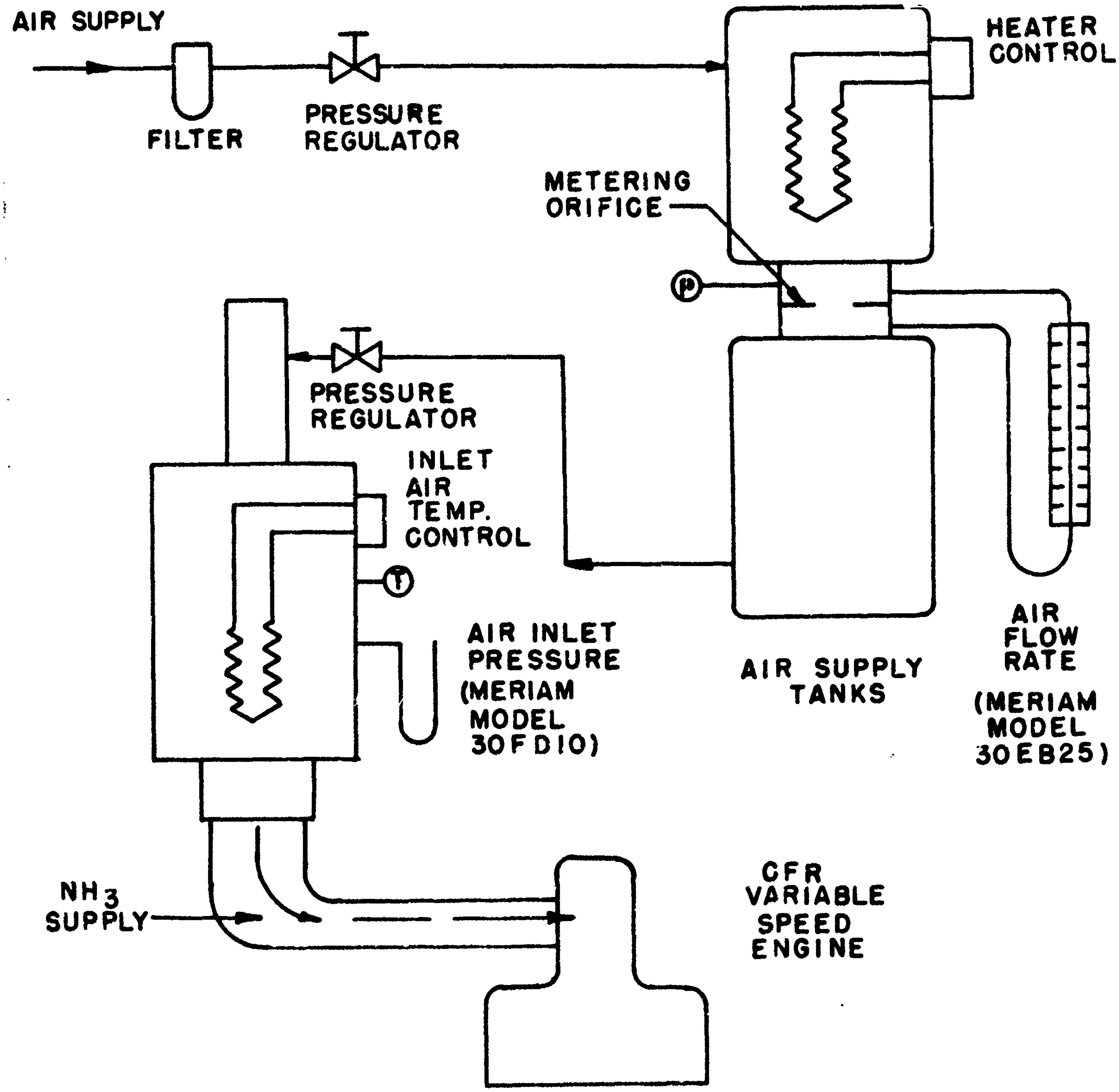


FIGURE 3

SCHEMATIC DIAGRAM OF AIR SUPPLY SYSTEM
TO CFR ENGINE

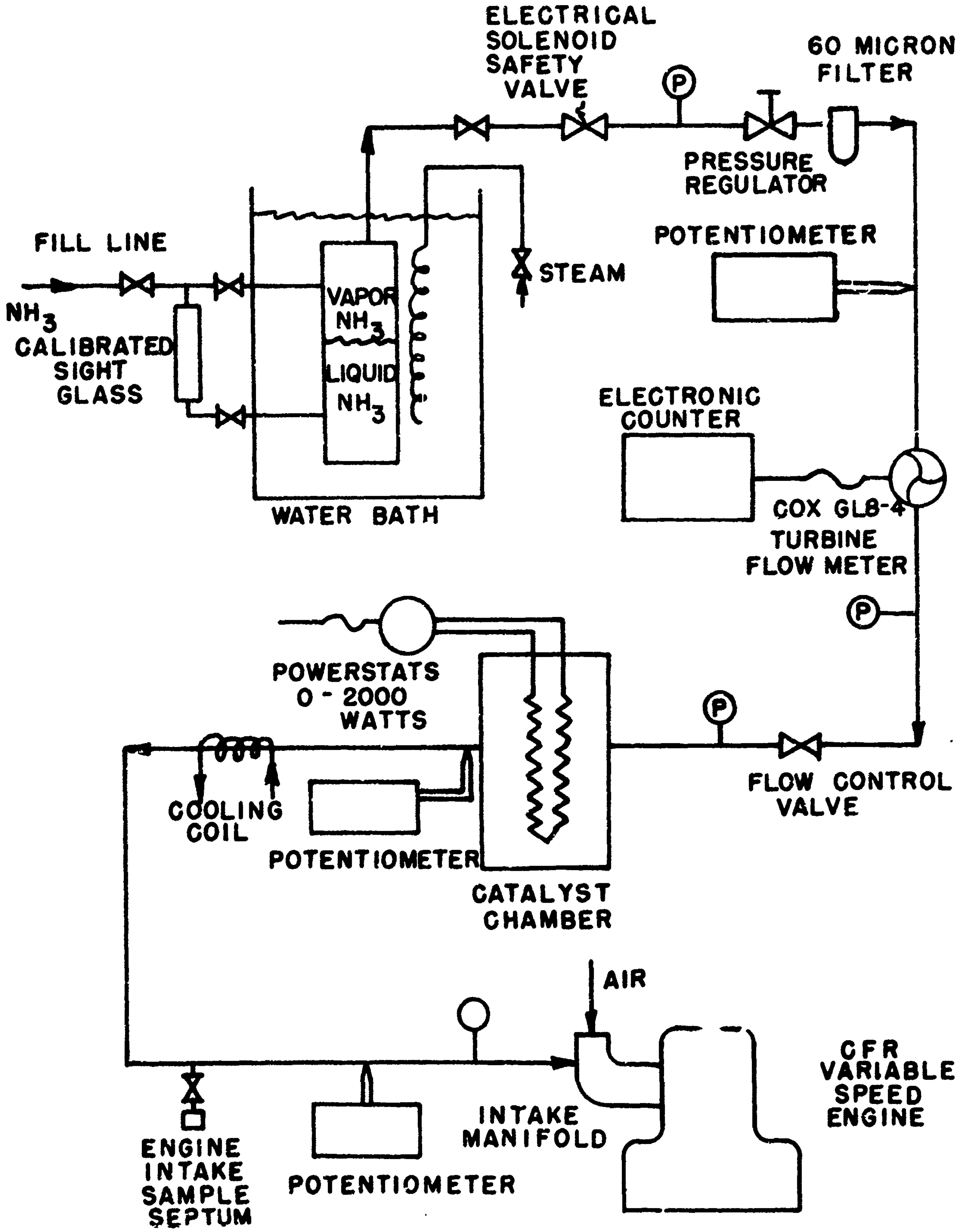


FIGURE 4

SCHEMATIC DIAGRAM OF AMMONIA
FUEL SUPPLY SYSTEM

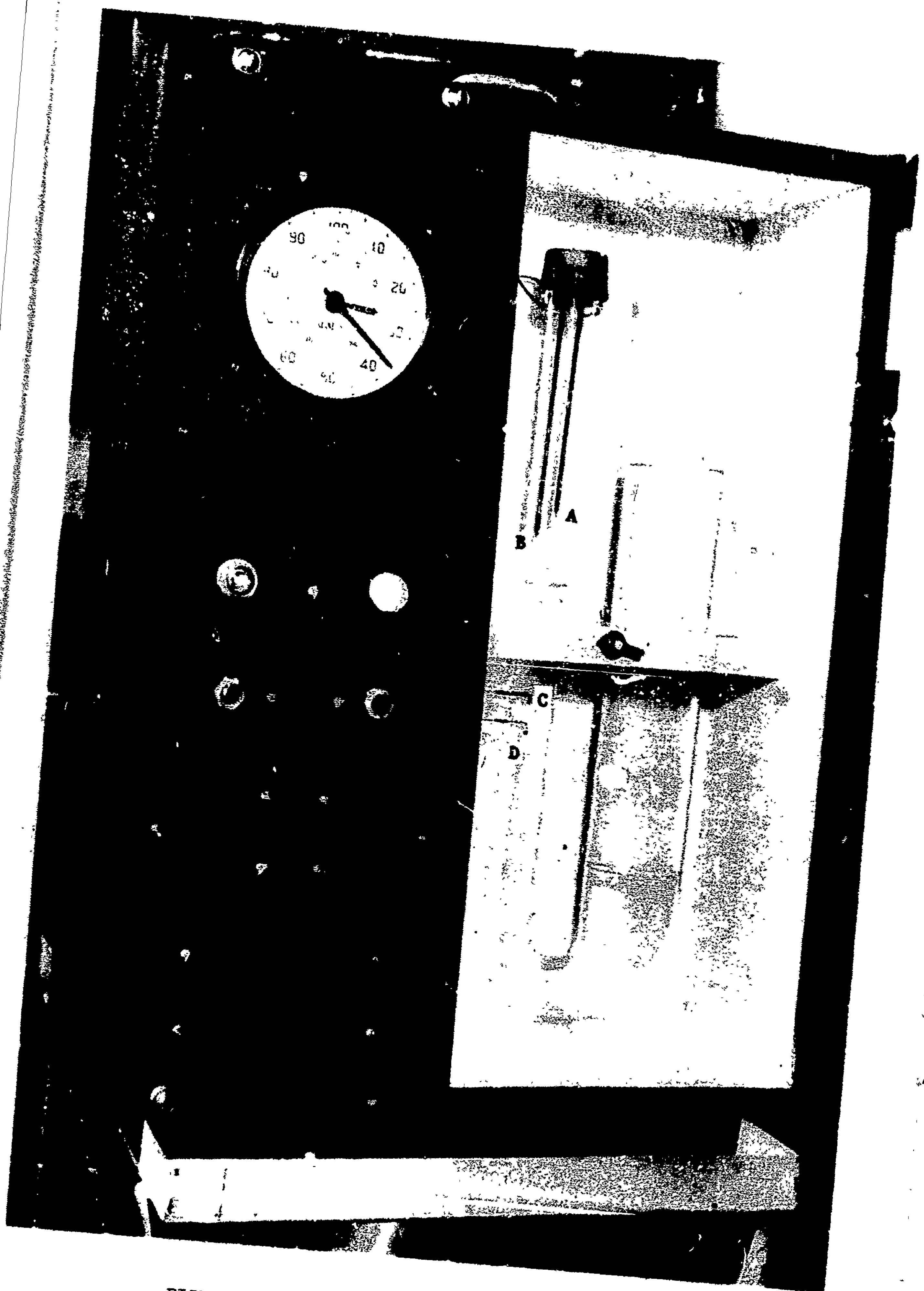


FIGURE 5. LIQUID FUEL FLOW METER

III. EXPERIMENTAL RESULTS

As indicated previously, all experimental results were obtained from a CFR spark ignition engine with gaseous premixed anhydrous ammonia and manifold injected iso-octane as the fuels. The test parameters were compression ratio, engine speed, manifold pressure and for certain tests hydrogen dissociation and spark advance. Iso-octane, C_8H_{18}, was selected as a representative hydrocarbon for the experimental reference fuel. Iso-octane was also used as the model for theoretical calculations[2] comparing ammonia and hydrocarbon fuels. The comparison in this report of experimental and theoretical results are thus compatible.

Throughout this report, fuel-air ratio will be presented in terms of the fuel-air equivalence ratio, ϕ, defined as the actual fuel-air ratio divided by the stoichiometrically correct fuel-air ratio. Therefore, $\phi = 1$ corresponds to a stoichiometrically correct fuel-air ratio. Use of equivalence ratio rather than fuel-air ratio displays the data in better perspective, particularly since ammonia has a chemically correct mixture with 6:1 parts of air by weight and iso-octane has a chemically correct mixture with 15:1 parts of air by weight.

The experimental results were obtained for a cooling jacket temperature of 152°F for iso-octane and 354°F for ammonia. The use of a relatively high jacket temperature when burning ammonia was for the purpose of enhancing ammonia decomposition during the compression stroke. A relatively low jacket temperature was used with iso-octane in order to suppress a tendency to knock at a compression ratio of 10.

Ammonia dissociation and thus hydrogen supply rates were held only as high as was felt necessary to obtain optimum performance at any given operating point. The amount of dissociated ammonia was thus a variable. It is now recognized that the quantity of ammonia which was dissociated was in a few instances marginally low or even below a critical amount. This factor will be considered in a later section at length. Nevertheless, the results shown in Figures 6 through 32 are completely satisfactory for comprehensive and reliable illustration of the important functional relationships.

1. Power Output

Indicated mean effective pressure, the universal expression for the power output to be obtained from an engine of given volumetric capacity, is shown as a function of the fuel-air equivalence ratio in Figures 6-11. The parameters are compression ratio, engine speed and manifold pressure. The data for each curve was collected at a fixed spark setting; the spark advance was adjusted for optimum at maximum power output. The range of optimum spark advance for iso-octane was 22° - 43°, and 35° - 55° BTDC for ammonia.

At an engine speed of 1800 rpm the predissociation of hydrogen, due to the limited capacity of the catalytic dissociator, was insufficient for ammonia to produce full power above a fuel-air equivalence ratio of approximately 0.90. The range of hydrogen deficiency is represented by the dashed portion of the curves.

Figures 12-14 present comparative power output data for ammonia and iso-octane as a function of fuel-air equivalence ratio over the range of parameters. Power output for iso-octane is greater than for ammonia at the same test conditions over the complete range of interest of variables and parameters. However, it is also evident from these figures that ammonia has no grossly divergent power performance characteristics from iso-octane and that the ammonia data are consistent and orderly within the limitations of hydrogen deficiency (for an engine speed of 1800 rpm at an equivalence ratio greater than $\phi = 0.90$). Peak power output for iso-octane occurs near $\phi = 1.15$; in contrast, peak power for ammonia is obtained at a mixture ratio which is about 10 per cent lean.

Power output for ammonia, as indicated by Figure 12, is affected to a much greater extent by low compression ratios than is iso-octane. The increase in maximum output for a compression ratio increase from 6 to 8 is 22 per cent for ammonia and 8 per cent for iso-octane. However, an increase in compression ratio from 8 to 10 produced modest increases in power output of nearly the same magnitude - 10 per cent for ammonia and 7 per cent for iso-octane.

Figure 13 shows power output as influenced by engine speed. The dashed portion of the 1800 rpm ammonia curve reflects a subcritical hydrogen supply rate, as previously mentioned. This decline in performance is dramatic as the mixture is richened. Reflected here is the self-generating character of the ammonia decomposition and flame phenomena, or the failure thereof when insufficient precombustion decomposition has been effected. Observation of this same sudden drop off was noted by Cornelius, Huellmantel and Mitchell.[1]

The trend of mean effective pressure with speed is a slightly increasing one, except of course for that portion of the 1800 rpm curve where the ammonia dissociator, because of its size, was unable to provide sufficient dissociation. For ammonia an increasing trend was not expected. Power output for ammonia is highly dependent upon hydrogen dissociation. Some dissociation occurs during the compression process and is dependent upon the time for reaction. Since at higher engine speeds there is less dwell time, less dissociation would be expected to occur in the cylinder. Therefore, the increasing trend for ammonia seems to indicate that the dissociation in the cylinder at these low compression rates is not significant compared to the amount of predissociation or other effects, perhaps related to turbulence and mixing factors.

The curves at the lower engine speeds, without hydrogen deficiency, indicate that the sensitivity of power output to fuel-air equivalence ratio is approximately the same for the two fuels.

Figure 14 shows the effect of manifold pressure on power output for a compression ratio of 8 and an engine speed of 1800 rpm. The increase in maximum power output for ammonia is in direct proportion to the absolute manifold pressure.

Figures 15 and 16 display the variation of maximum power output, relative maximum power output and theoretical power output with compression ratio. Figure 16 shows directly the relative indicated mean effective pressure obtained when using ammonia as a substitute for iso-octane. Also shown, as dashed lines, are the theoretical prediction of relative power output of approximately .80 and the theoretical prediction adjusted for

5 per cent hydrogen dissociation. The predicted output was obtained by comparing the theoretical output for ammonia at $\phi = 0.90$ with that for iso-octane at $\phi = 1.15$ (the respective fuel-air equivalence ratios for maximum output).

The experimentally determined relative output falls about 8 per cent short of the 80 per cent predicted at a compression ratio of 10 and about 17 per cent less for a compression ratio of 6. There are at least two reasons for this disparity between experiment and theory.

The first and most straight forward is related to prior dissociation of ammonia into H_2 and N_2 before introduction into the cylinder. Theoretical calculations did not account for the accompanying increase in volume of fuel and resulting decrease in volumetric efficiency. For example, decomposition of 25 per cent of the ammonia (for 4 to 5 per cent level of H_2 by weight) will increase the total volume of fuel by the same 25 per cent. At a stoichiometric fuel-air ratio this reduces the amount of air inducted by about 5 per cent. The power output will drop off accordingly. This then accounts for a significant portion of the difference between theory and experiment at a compression ratio of 10.

The other factor involved in the disparity between theory and experiment and particularly the improvement with increasing compression ratio is concerned with the reluctance with which ammonia burns. It has already been indicated that hydrogen in the fuel feed improves the combustion process and so does an increase in compression ratio. It is almost certain that decomposition of ammonia takes place during the compression process. The decomposition produces more hydrogen, but also intermediate species. Thus, as compression ratio and temperature of compression are increased, so are hydrogen and intermediate species concentrations.

This later phenomenon is different from that of dissociating the ammonia in a catalytic dissociator prior to induction. With prior decomposition the engine will receive substantially only hydrogen, nitrogen and ammonia. Because of the time factor involved the engine cannot receive intermediate dissociation species which may have an important role in combustion.

Note also that overly decomposing the ammonia prior to introduction into the engine will result in a mixture which is too rich in hydrogen and attendant difficulties with rapid burn and engine roughness. There is undoubtedly an optimum amount of predissociation corresponding to maximum obtainable output and smooth operation. This latter type of optimization has not been carried out to completion under this program. This optimizing procedure will be a function of engine combustion chamber shape in addition to such other factors as speed, compression ratio, engine temperatures, load and other items.

2. Specific Fuel Consumption

The most significant measure of engine economy is specific fuel consumption. Figures 17 - 22 are the basic comparative fuel consumption curves for ammonia and iso-octane. Indicated specific fuel consumption is shown as a function of the fuel-air equivalence ratio. The parameters are compression ratio, engine speed, and manifold pressure. Cooling jacket temperatures, spark advance and regions of hydrogen deficiency are identical to those for the power output curves. The range of hydrogen deficiency is again represented as the dashed portion of the curves.

Figures 23 - 25 present comparative indicated specific fuel consumption data for ammonia and iso-octane as a function of fuel-air equivalence ratio over the range of the parameters. Over the complete range of interest of parameters and variables, for the same test conditions, the indicated specific fuel consumption for ammonia is greater than that for iso-octane. It is also evident from these curves that, as for power output, ammonia has no grossly divergent characteristics from iso-octane and that the ammonia data are consistent and orderly within the limitations of hydrogen deficiency (at an engine speed of 1800 rpm for a fuel-air equivalence ratio greater than ϕ = 0.90). These figures illustrate directly that specific fuel consumption for ammonia is over twice that of iso-octane on comparable operations. They also illustrate that minimum specific fuel consumption with ammonia occurs at fuel-air equivalence ratios which are 20 to 25 per cent lean in contrast to iso-octane for which minimum values occur at mixtures which are approximately 10 per cent lean.

Figure 23 indicates that specific fuel consumption for ammonia is affected to a much greater extent than for iso-octane by low compression ratios. The decrease in minimum specific fuel consumption for a compression ratio increase from 6 to 8 is 22 per cent for ammonia and 8 per cent for iso-octane. However, an increase in compression ratio from 8 to 10 produced modest decreases in indicated specific fuel consumption of nearly the same magnitudes--3 per cent for ammonia and 2 per cent for iso-octane.

Figure 24 shows the effect of manifold pressure on indicated specific fuel consumption. For both fuels, the change in manifold pressure produced only a very modest effect of approximately 2 - 3 per cent on fuel consumption. However, for ammonia the decrease in manifold pressure produced a reduction in fuel consumption while the opposite effect was produced for iso-octane.

The effect of engine speed on indicated specific fuel consumption is shown in Figure 25. The curves for both fuels display, as would be expected, little or no effect of engine speed on fuel consumption. The effect of hydrogen deficiency is again quite noticeable from the ammonia curve for an engine speed of 1800 rpm. The curves at the lower engine speeds, without hydrogen deficiency, indicate that the sensitivity of fuel consumption to fuel-air equivalence ratio is similar for the two fuels.

The variation of actual, relative, and theoretical indicated specific fuel consumption with compression ratio is displayed in Figures 26 - 28. Figure 26 shows a comparison of actual and theoretical fuel consumption for the condition of minimum fuel consumption and maximum power output. At minimum consumption, ammonia must be supplied at two and one-half times the weight flow rate of iso-octane. For maximum power ammonia flow is double that of iso-octane. The experimental results fall remarkably close to the theoretical.

3. Influence of Hydrogen Concentration Resulting From Ammonia Dissociation

Figure 29 contains the results of an investigation into the influences of hydrogen concentration on power output. Hydrogen content of the partially dissociated ammonia stream was determined directly by gas chromatography as previously described in Section II. The figure shows that 4 to 5 per cent hydrogen is necessary to obtain maximum power at any given mixture ratio when running at 1800 rpm, a compression ratio of eight, and manifold pressure of 30" Hg. This set of relationships would be expected to apply generally to other operating conditions.

Hydrogen was supplied to the test engine in the most likely practical manner, by partially decomposing the ammonia supply. This procedure results in a fuel-air mixture which is exactly corresponding in ammonia, hydrogen, oxygen and nitrogen concentrations to that in any proposed practical application which would use ammonia dissociation for hydrogen.

4. Spark Advance

The effect of spark timing on power output for ammonia and iso-octane is shown in Figures 30 - 32.

Figures 30 and 31 show how optimizing spark timing can influence power output over a range of fuel-air equivalence ratios at a compression ratio of 8 and an engine speed of 1800 rpm. For ammonia the spark should be slightly retarded and for iso-octane slightly advanced from peak power optimum for maximum performance at richer mixtures. For lean mixtures, ammonia required no spark adjustment; iso-octane required slightly greater advancement than for rich mixtures. For both fuels the increase in power output with spark adjustment was quite small over a normal operating range of $\pm$ 10 per cent from ϕ at maximum power. Ammonia appears to be slightly more sensitive than iso-octane to spark setting at rich mixtures. For the same engine speed and compression ratio, reduced manifold pressure produced the same power output trends due to spark adjustments.

Illustrated in Figure 32 is the effect of spark timing on power output at the fuel-air equivalence ratio for maximum power output. This data is for a compression ratio of 8, an engine speed of 1800 rpm, and a manifold pressure of 30" of Hg. The optimum timing for ammonia is about 55 degrees before dead center and for iso-octane about 30 degrees at these operating conditions. Both fuels show only a modest sensitivity of the same magnitude to spark setting over a range of optimum $\pm$ 10 degrees. Ammonia experiences about a 5 per cent loss in power over this range compared to a 4 per cent loss for iso-octane over the same range.

5. Engine Oil Analysis

A sample of crankcase oil was analyzed through the courtesy of Chevron Research Company to determine the effect of using ammonia as a fuel on engine oil under normal engine operating conditions. Detailed results of the chemical and spectrochemical analysis are included in the Appendix, Exhibits I and II. There was no indication that the oil was deteriorating from operation with ammonia fuel. However, an abnormally high amount of copper and lead in the oil sample might indicate possible corrosion of a copper-lead bearing.

IV. CONCLUSIONS

1. Ammonia can be used successfully as a spark ignition engine fuel and at presently existing compression ratios, if introduced as a vapor and if first partly dissociated to hydrogen and nitrogen. Under such circumstances little engine modification is necessary other than a means for flow control of the ammonia and adjustment of the spark timing.

2. Maximum experimental power output for ammonia was 72 per cent of that for iso-octane. This result compares favorably with a theoretically predicted output, when adjusted for 5 per cent hydrogen dissociation, of 75 per cent.

3. Specific fuel consumption using ammonia is increased by a factor of 2 over that of hydrocarbon when compared at peak power and 2-1/2 times when compared at maximum economy.

4. Hydrogen concentration in the fuel feed is a critical factor for successful operation on ammonia as fuel. Minimum concentrations appear to be 4 to 5 per cent by weight at intermediate engine speeds of 1800 rpm.

5. Engine performance rapidly falls if less than minimum concentrations of hydrogen are used. This seems to relate to the self-generating character of the ammonia decomposition during the compression and combustion processes.

6. Performance factors such as are influenced by engine speed, spark timing and manifold pressure are not far different with ammonia than with hydrocarbons as long as minimum amounts of hydrogen are inducted as a part of the fuel flow.

FIGURE 6

INFLUENCE OF FUEL-AIR RATIO
UPON POWER OUTPUT

COMPRESSION RATIO - 6
R.P.M. - 1800
MANIFOLD PRESSURE - 30" Hg.

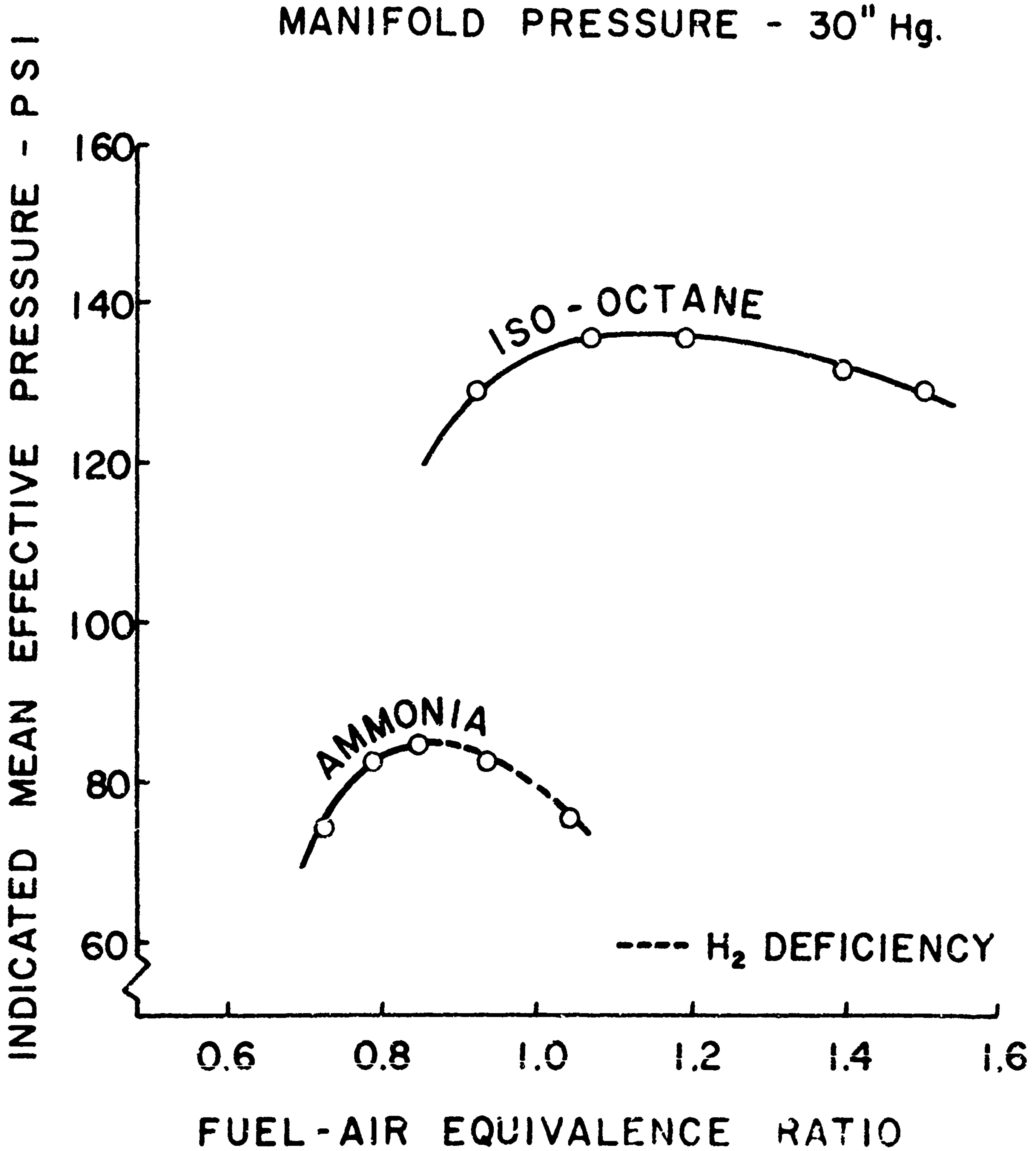


FIGURE 7

INFLUENCE OF FUEL-AIR RATIO UPON POWER OUTPUT

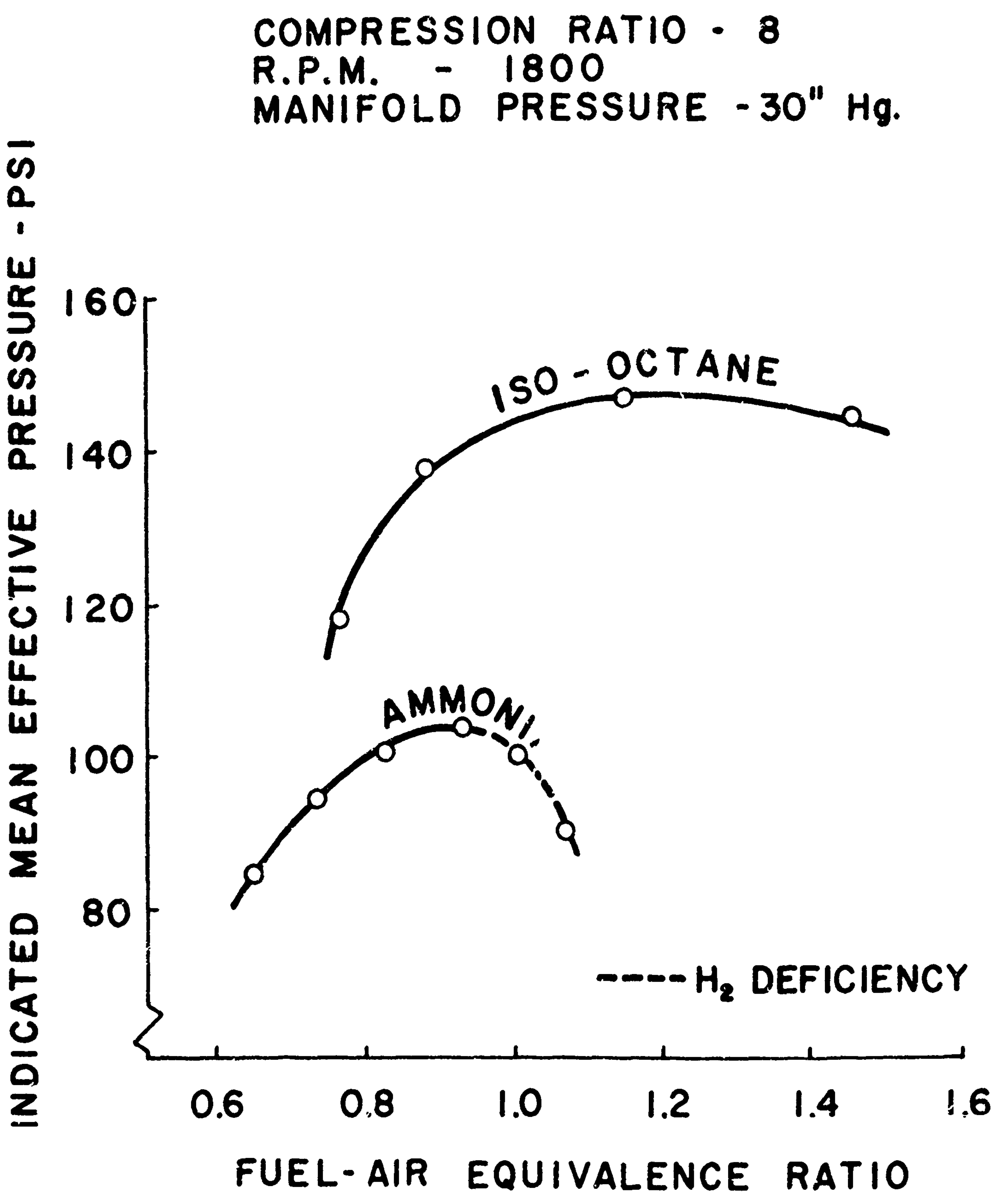


FIGURE 8

INFLUENCE OF FUEL-AIR RATIO UPON POWER OUTPUT

COMPRESSION RATIO - 10
R.P.M. - 1800
MANIFOLD PRESSURE - 30" Hg.

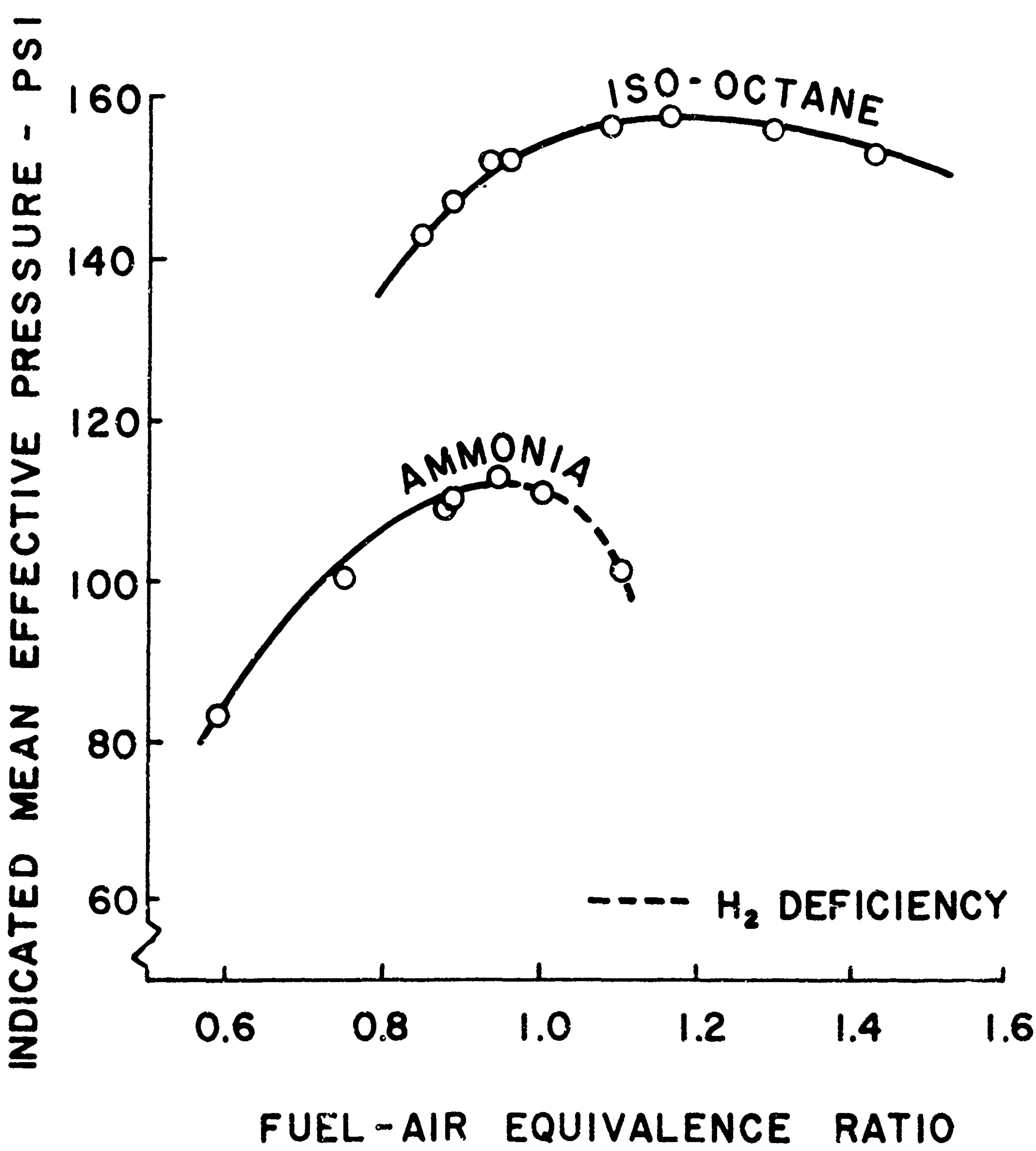


FIGURE 9

INFLUENCE OF FUEL-AIR RATIO
UPON POWER OUTPUT

COMPRESSION RATIO - 8
R.P.M. - 1300
MANIFOLD PRESSURE - 30" Hg.

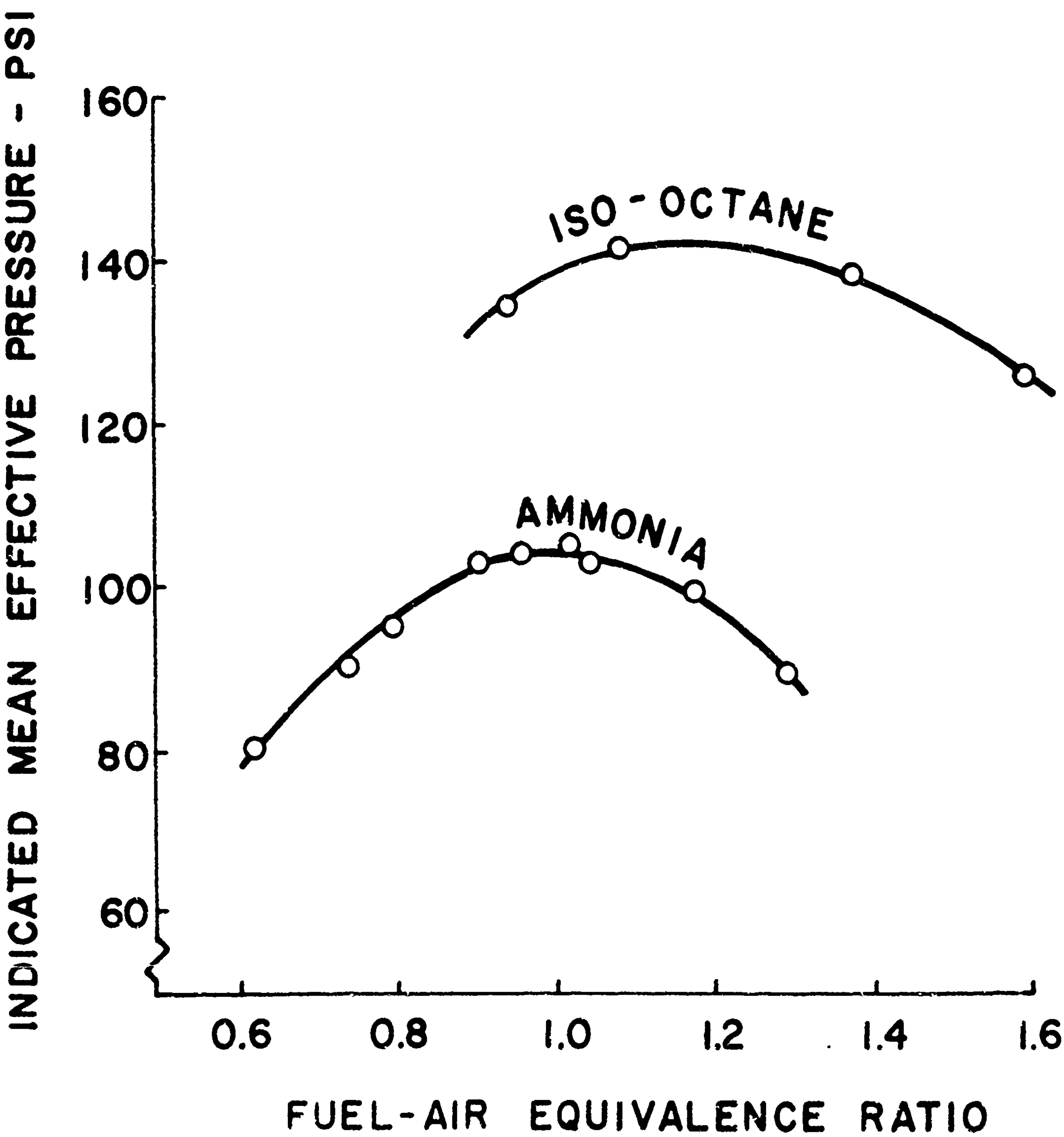


FIGURE 10

INFLUENCE OF FUEL-AIR RATIO
UPON POWER OUTPUT

COMPRESSION RATIO - 8
R.P.M. - 1000
MANIFOLD PRESSURE - 30" Hg.

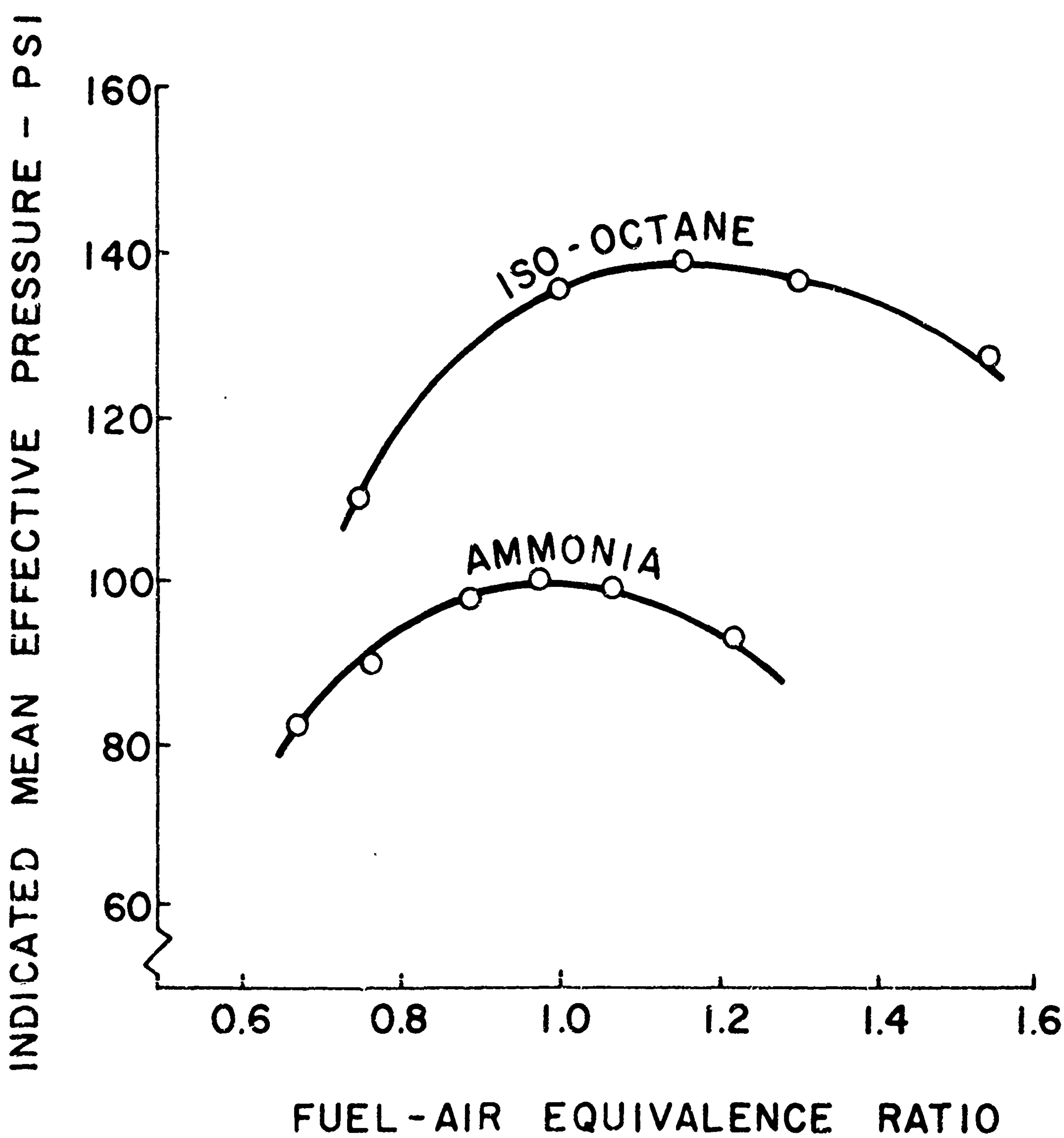


FIGURE 11

INFLUENCE OF FUEL-AIR RATIO UPON POWER OUTPUT

COMPRESSION RATIO - 8
R.P.M. - 1800
MANIFOLD PRESSURE - 21" Hg.

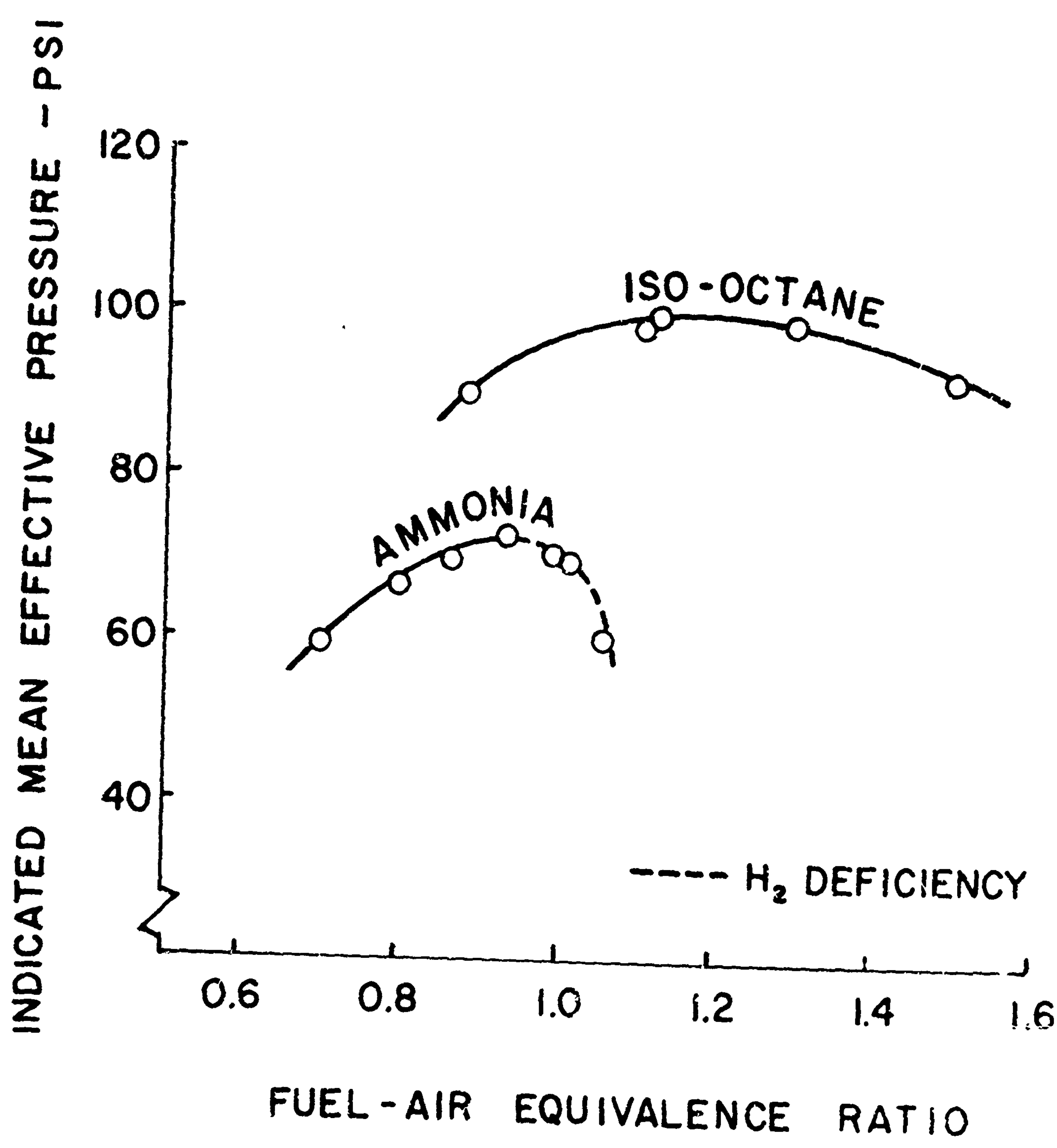


FIGURE 12

COMPARATIVE PERFORMANCE OF AMMONIA AND ISO-OCTANE

R. P. M. - 1800
MANIFOLD PRESSURE - 30" Hg.

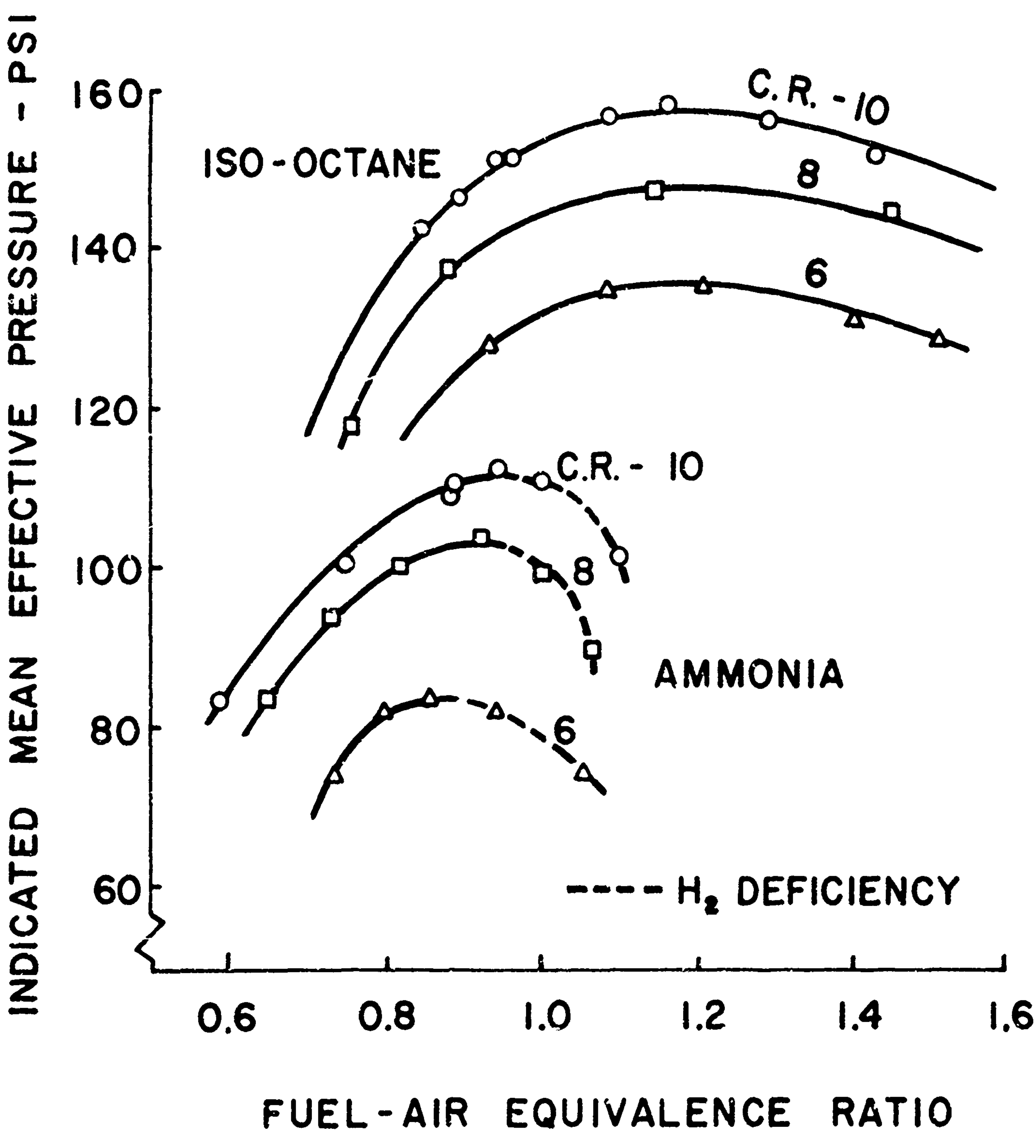


FIGURE 13

EFFECT OF ENGINE SPEED ON POWER OUTPUT

COMPRESSION RATIO - 8
MANIFOLD PRESSURE - 30" Hg.

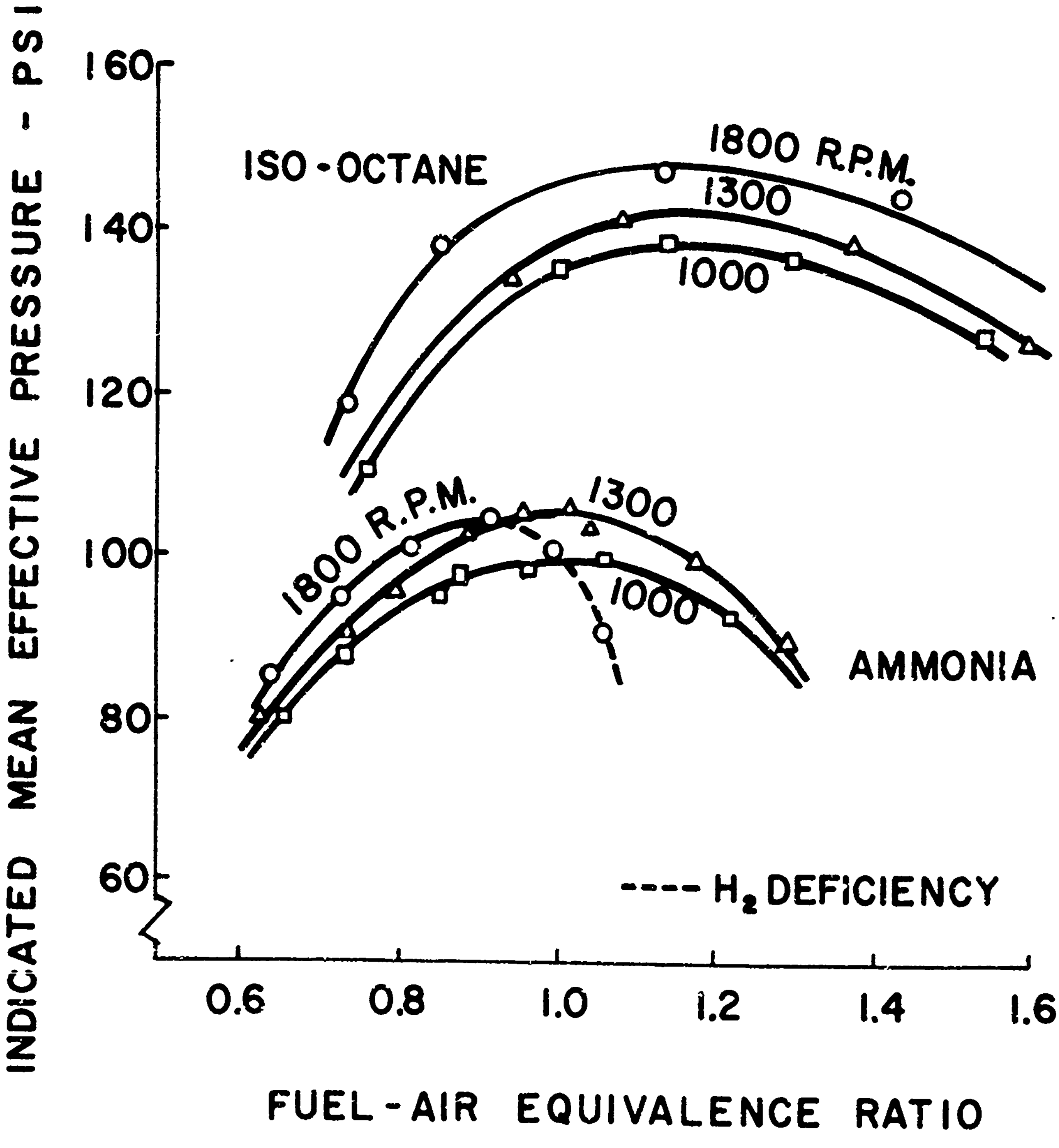


FIGURE 14

EFFECT OF MANIFOLD PRESSURE ON POWER OUTPUT

COMPRESSION RATIO - 8
R.P.M. - 1800

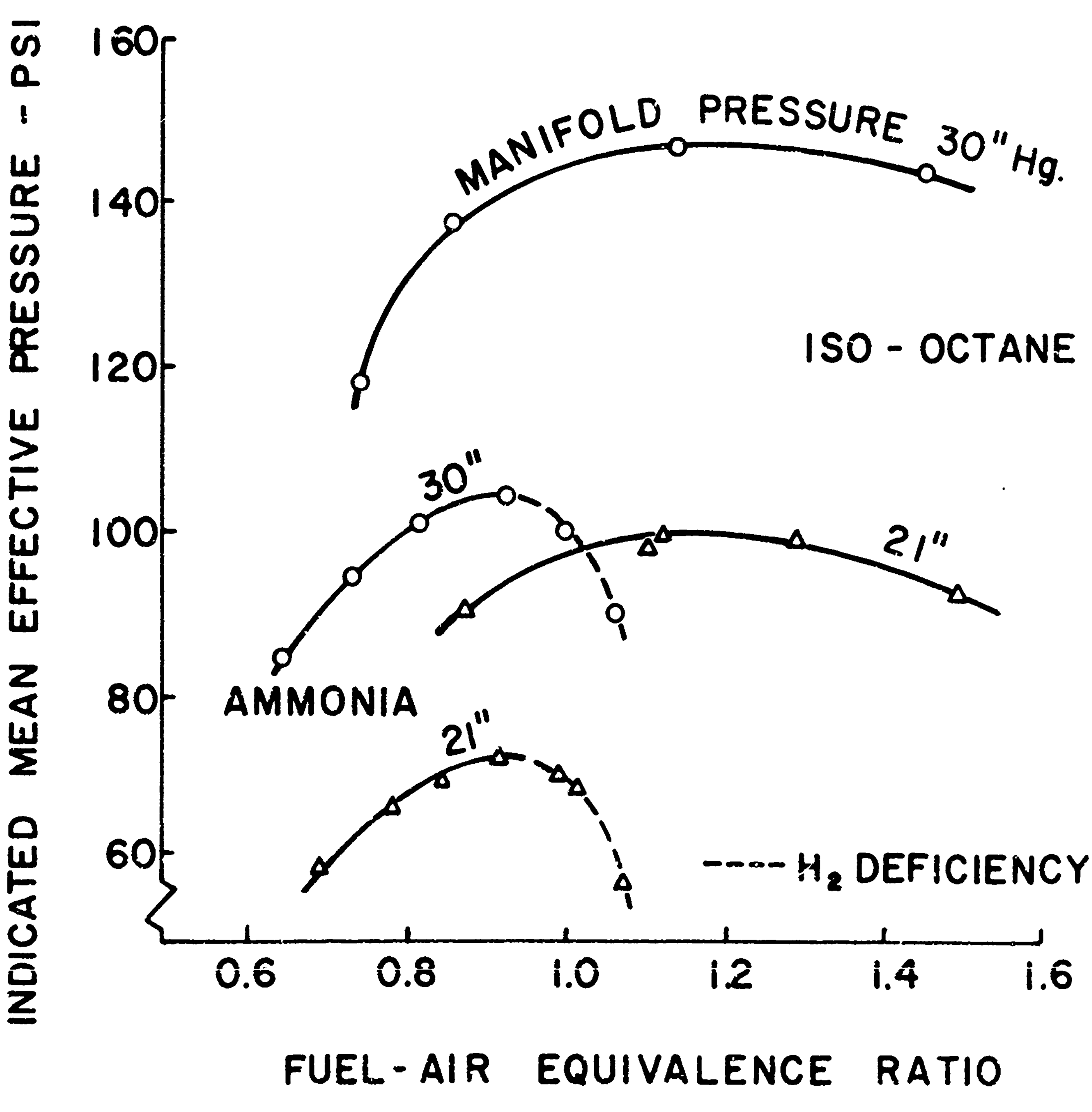


FIGURE 15

VARIATION OF MAXIMUM POWER OUTPUT WITH COMPRESSION RATIO

R.P.M. - 1800
MANIFOLD PRESSURE - 30" Hg.

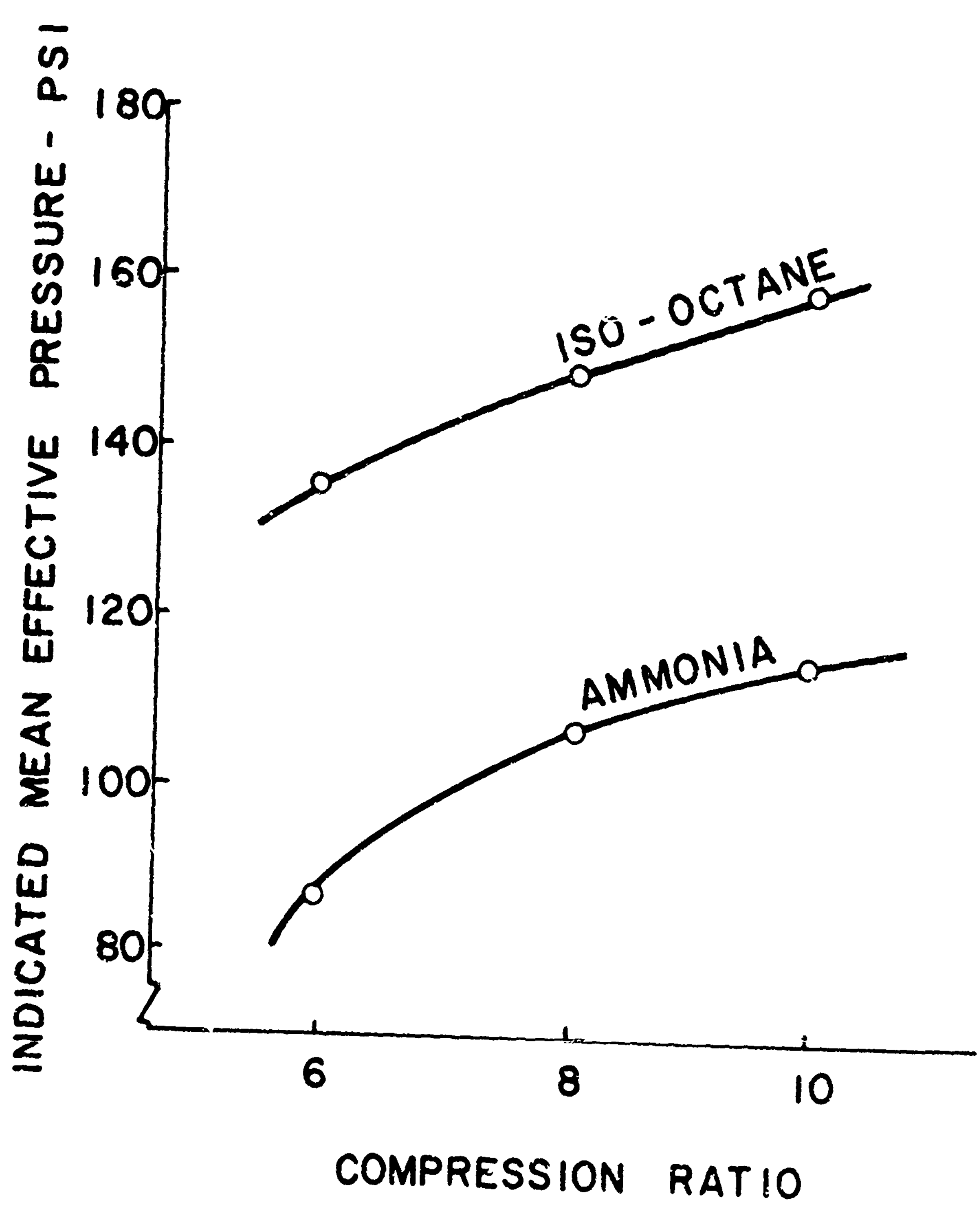


FIGURE 16

RELATIVE MAXIMUM OUTPUT FOR AMMONIA AS COMPARED TO ISO - OCTANE

R.P.M. - 1800
MANIFOLD PRESSURE - 30" Hg.

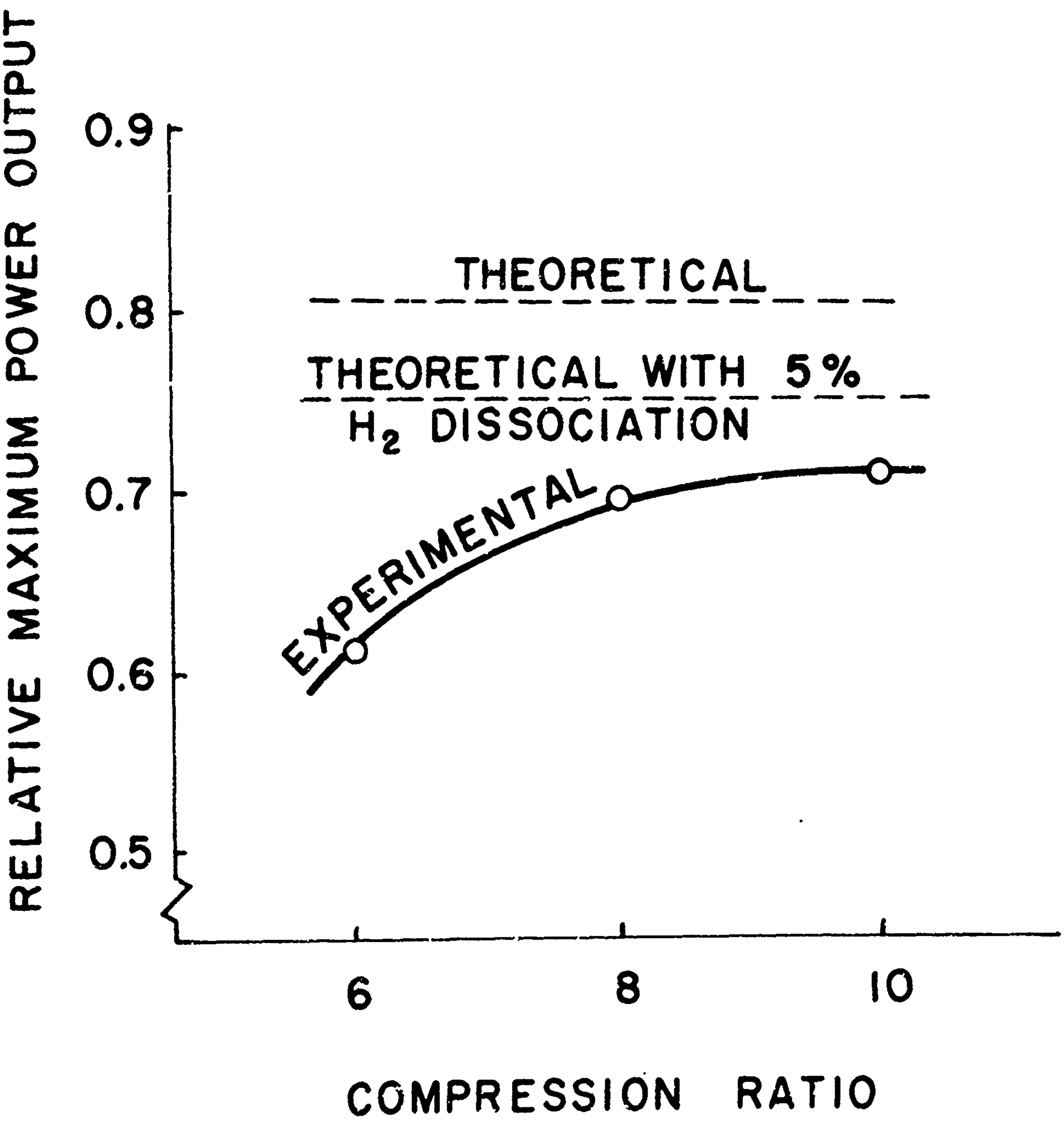


FIGURE 17

INFLUENCE OF FUEL-AIR RATIO UPON FUEL CONSUMPTION

COMPRESSION RATIO - 6
R.P.M. - 1800
MANIFOLD PRESSURE - 30" Hg.

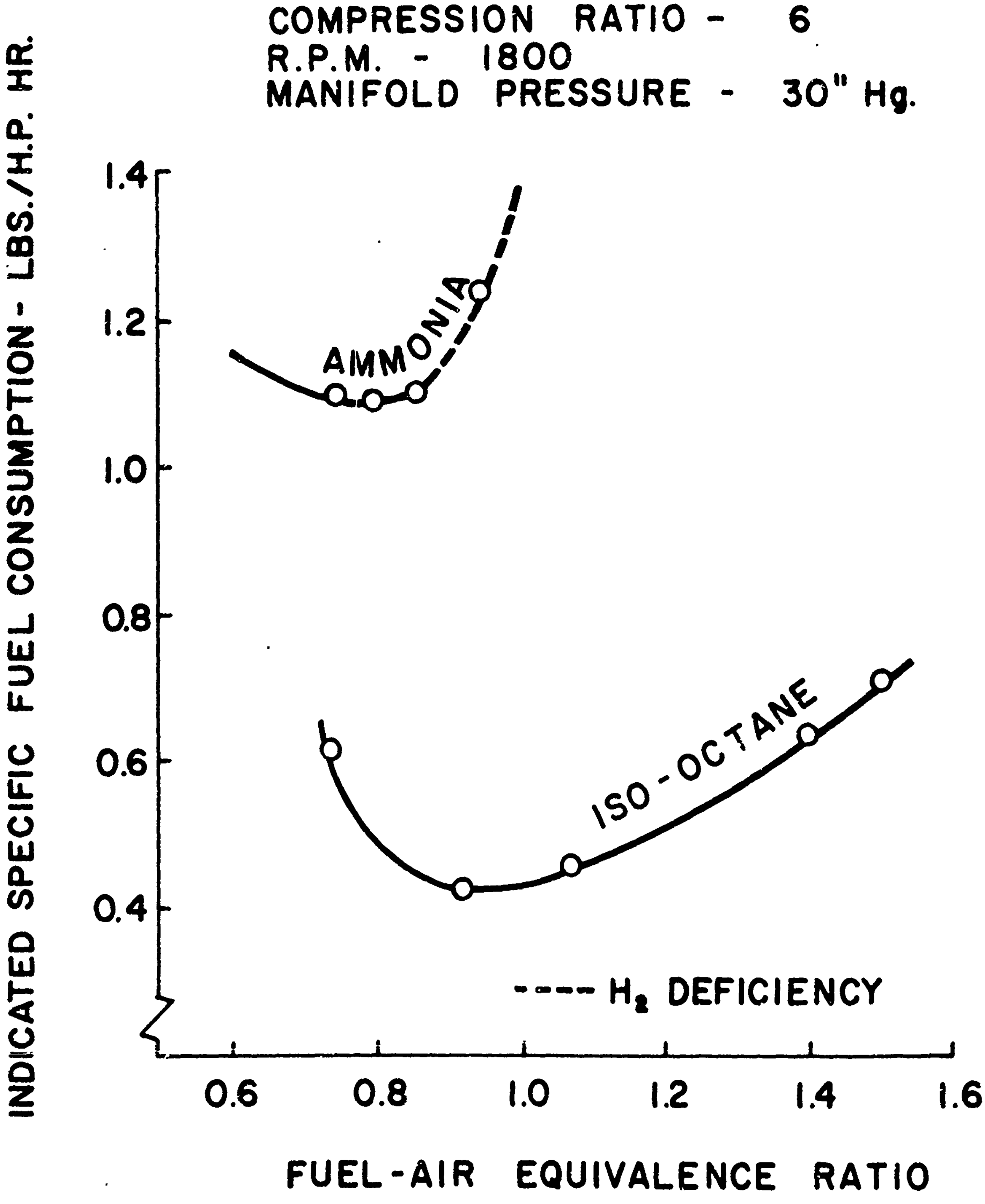


FIGURE 18

INFLUENCE OF FUEL-AIR RATIO
UPON FUEL CONSUMPTION

COMPRESSION RATIO - 8
R.P.M. - 1800
MANIFOLD PRESSURE - 30" Hg.

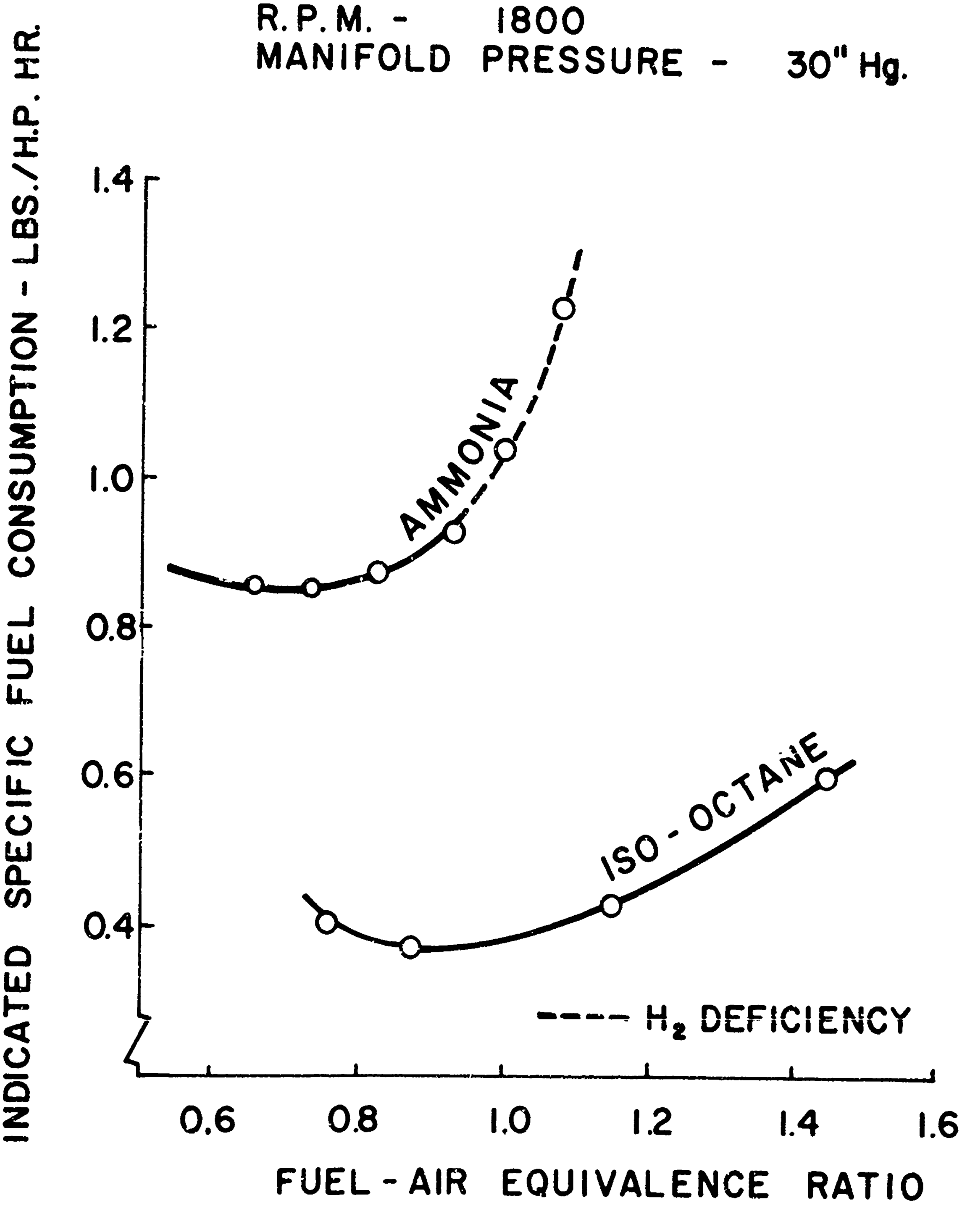


FIGURE 19

INFLUENCE OF FUEL-AIR RATIO UPON FUEL CONSUMPTION

COMPRESSION RATIO - 10
R.P.M. - 1800
MANIFOLD PRESSURE - 30" Hg.

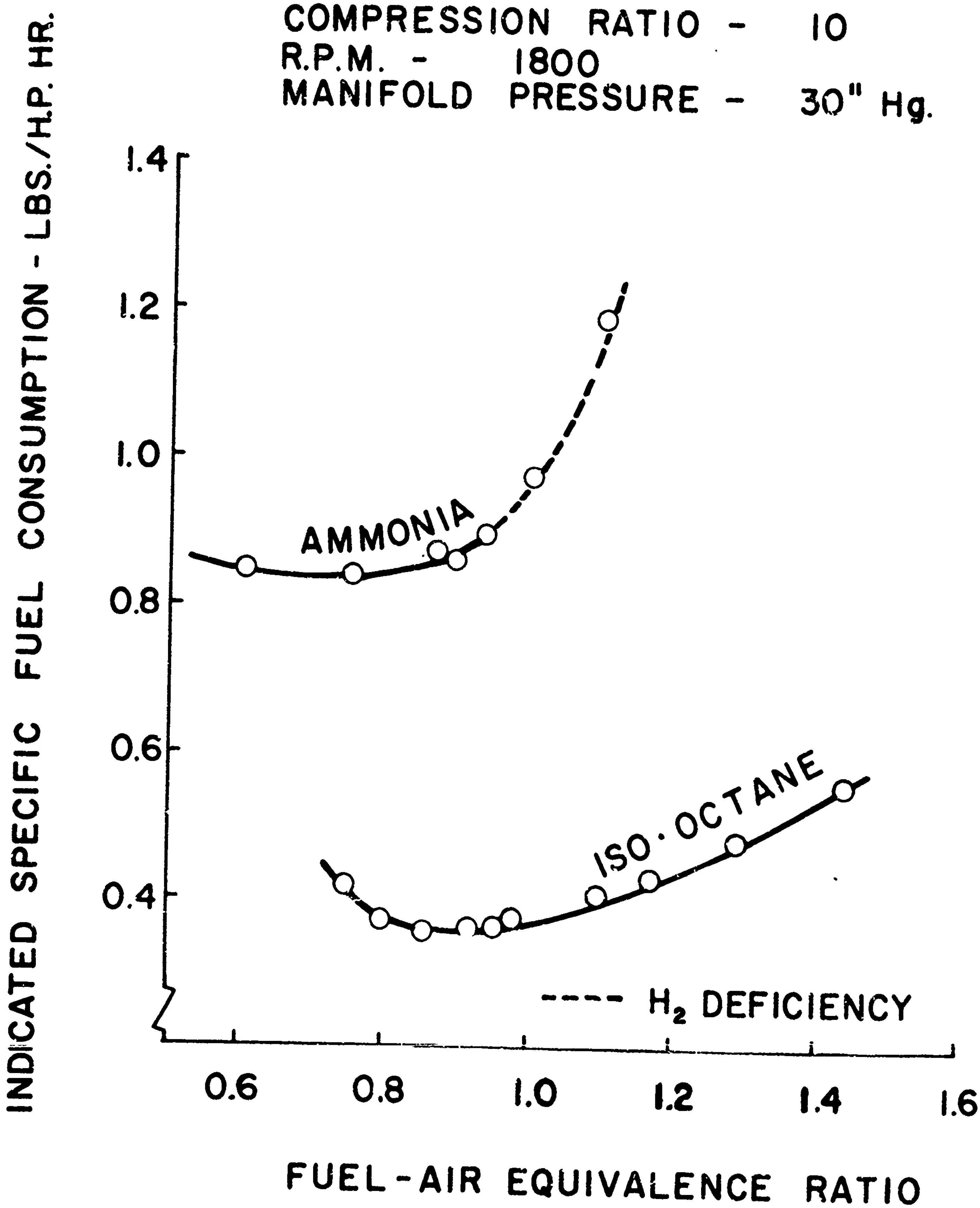


FIGURE 20

INFLUENCE OF FUEL-AIR RATIO
UPON FUEL CONSUMPTION

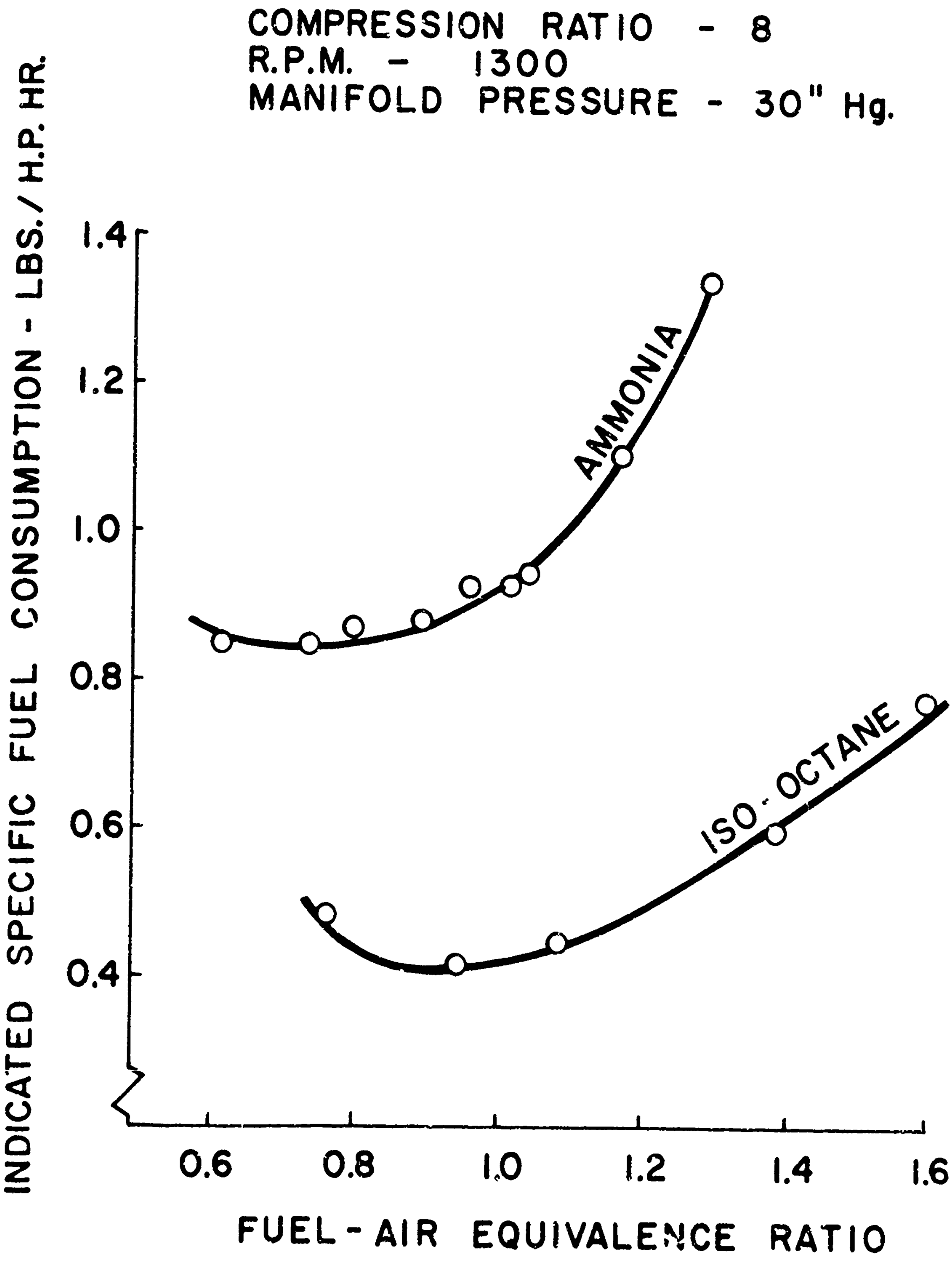


FIGURE 21

INFLUENCE OF FUEL-AIR RATIO
UPON FUEL CONSUMPTION

COMPRESSION RATIO - 8
R.P.M. - 1000
MANIFOLD PRESSURE - 30" Hg.

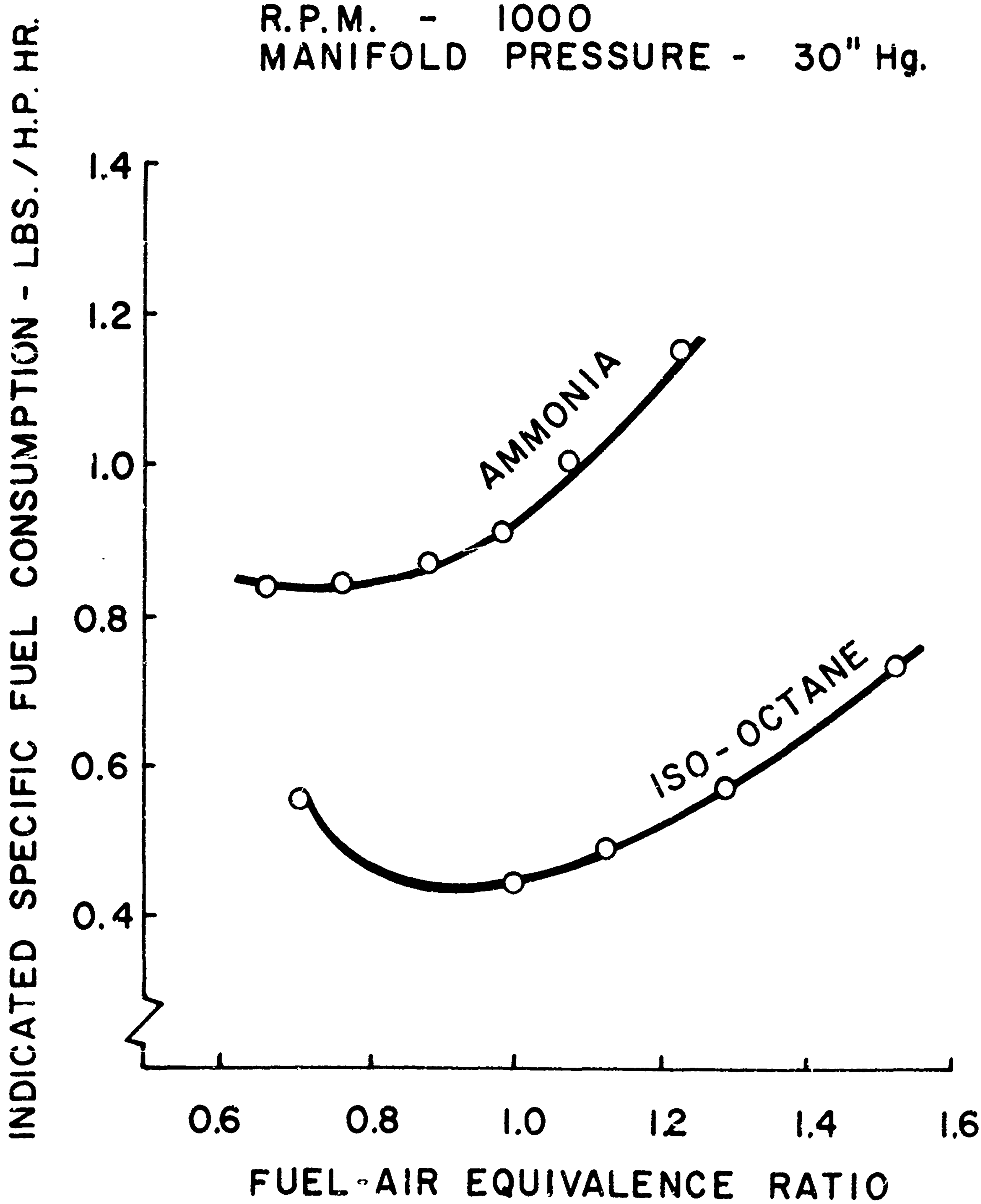


FIGURE 22

INFLUENCE OF FUEL-AIR RATIO UPON FUEL CONSUMPTION

COMPRESSION RATIO - 8
R.P.M. - 1800
MANIFOLD PRESSURE - 21" Hg.

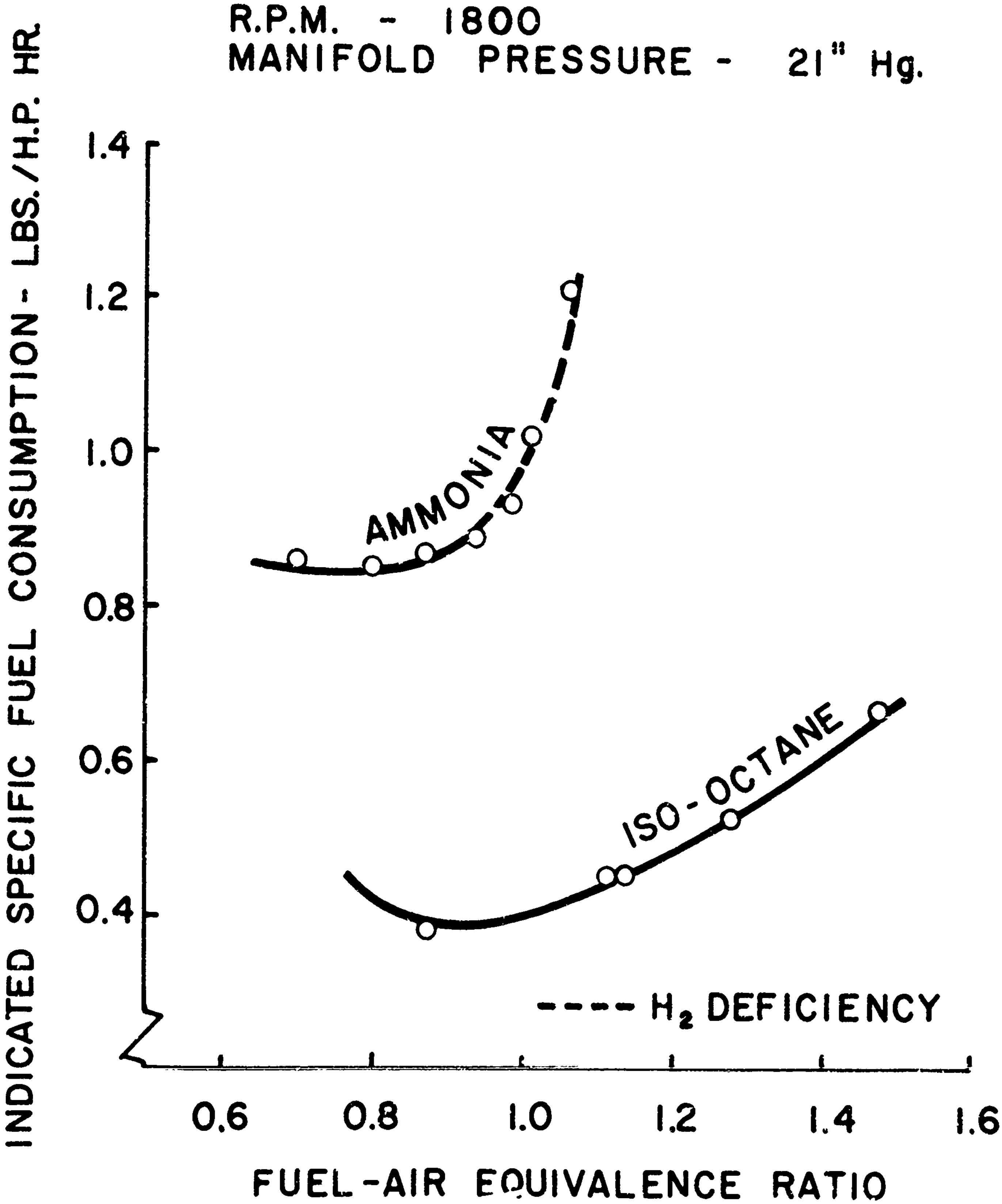


FIGURE 23

COMPARATIVE FUEL CONSUMPTION
AMMONIA AND ISO-OCTANE

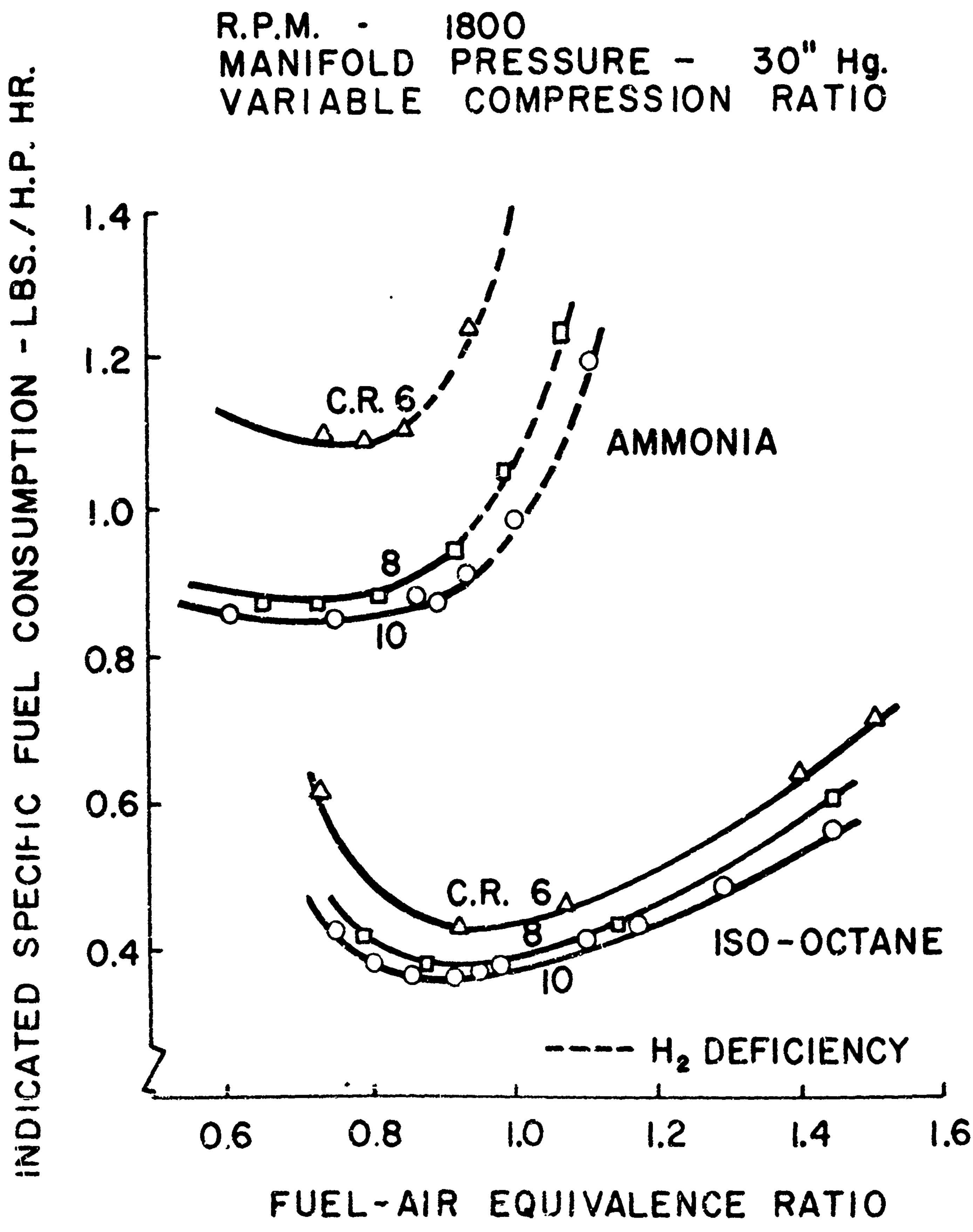


FIGURE 24

COMPARATIVE FUEL CONSUMPTION
AMMONIA AND ISO - OCTANE -
VARIABLE MANIFOLD PRESSURE

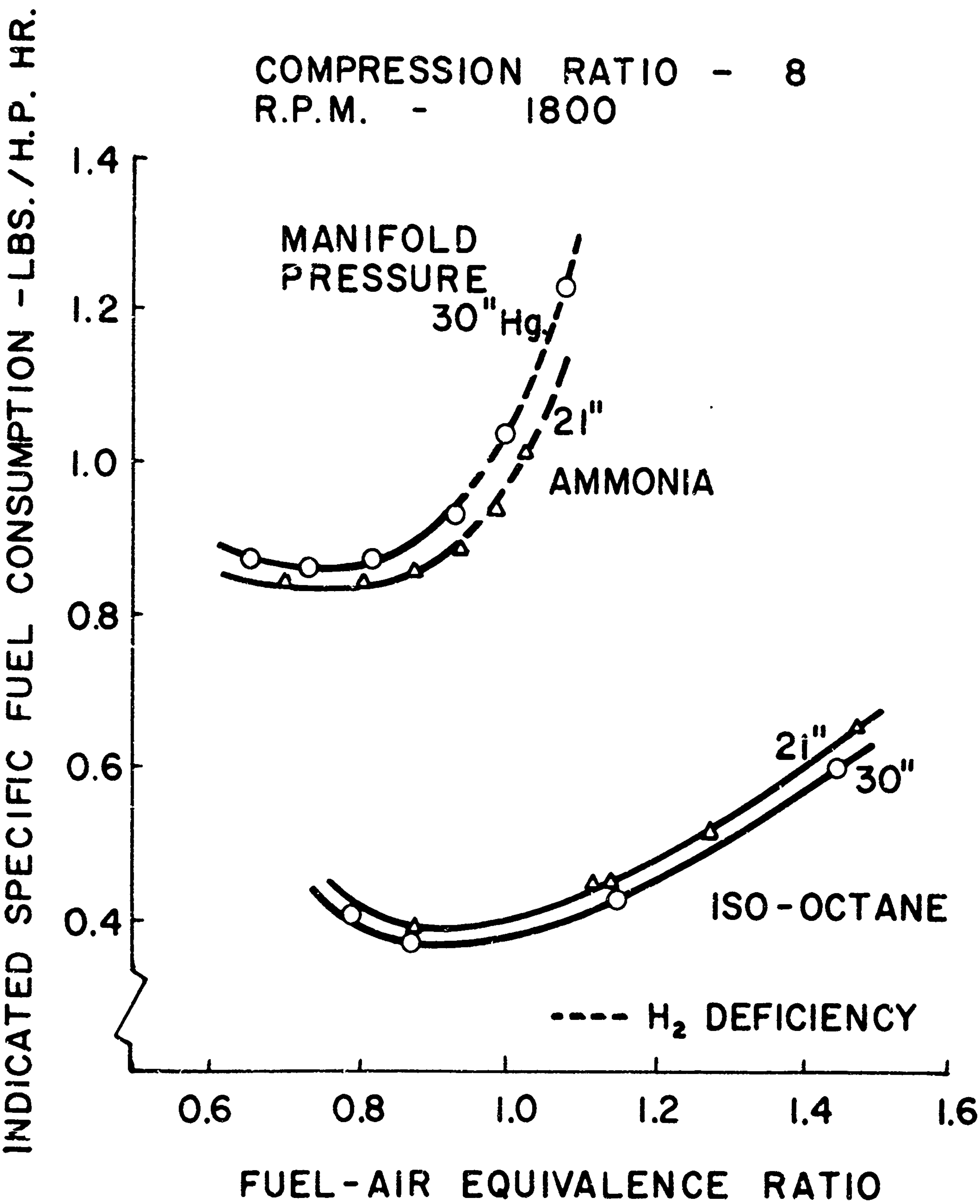


FIGURE 25

COMPARATIVE FUEL CONSUMPTION
AMMONIA AND ISO-OCTANE

VARIABLE ENGINE SPEED
COMPRESSION RATIO - 8
MANIFOLD PRESSURE - 30" Hg.

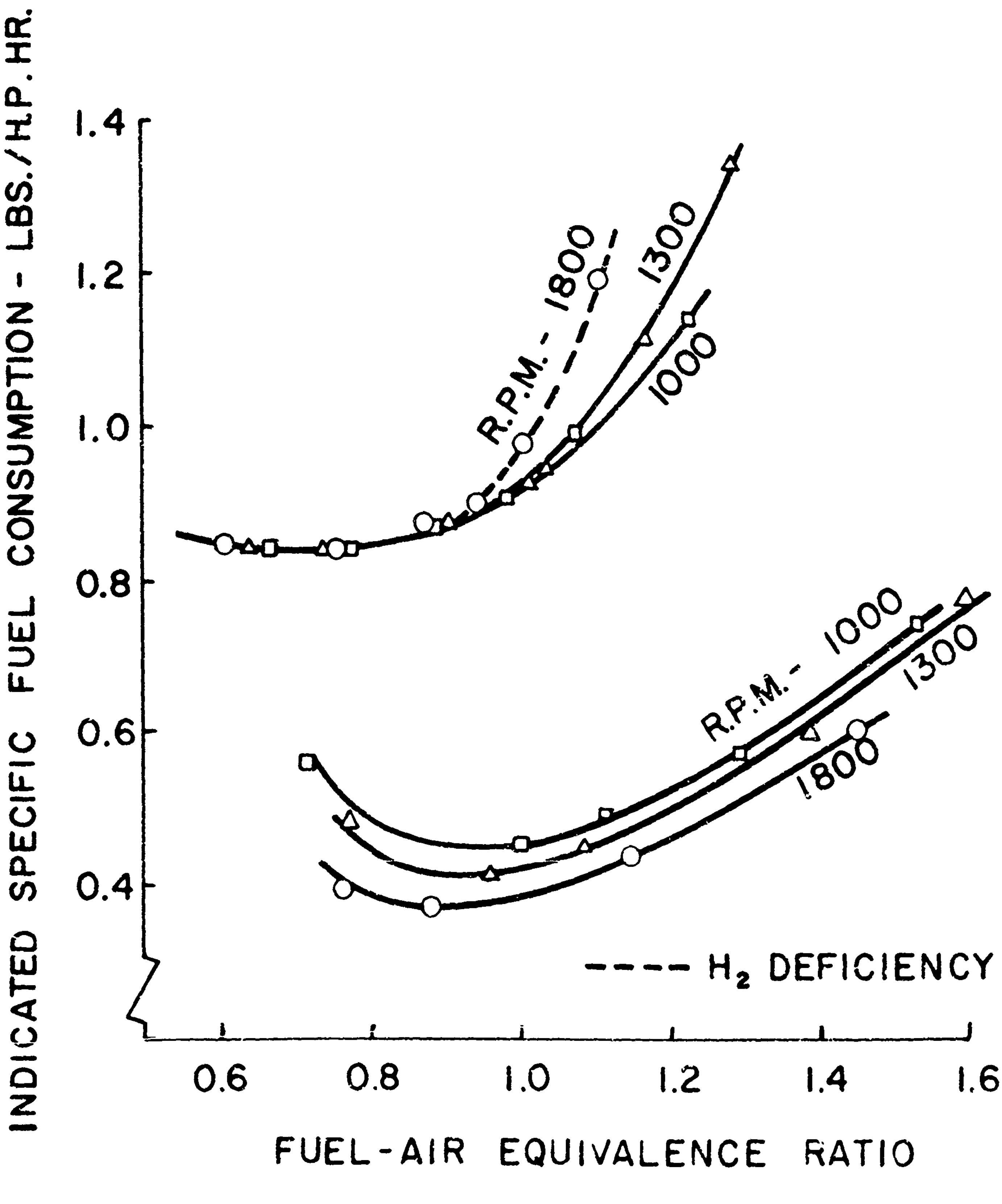


FIGURE 26

FUEL CONSUMPTION OF AMMONIA COMPARED TO ISO-OCTANE

(ISFC AMMONIA / ISFC ISO-OCTANE)

R.P.M. - 1800
MANIFOLD PRESSURE - 30" Hg.

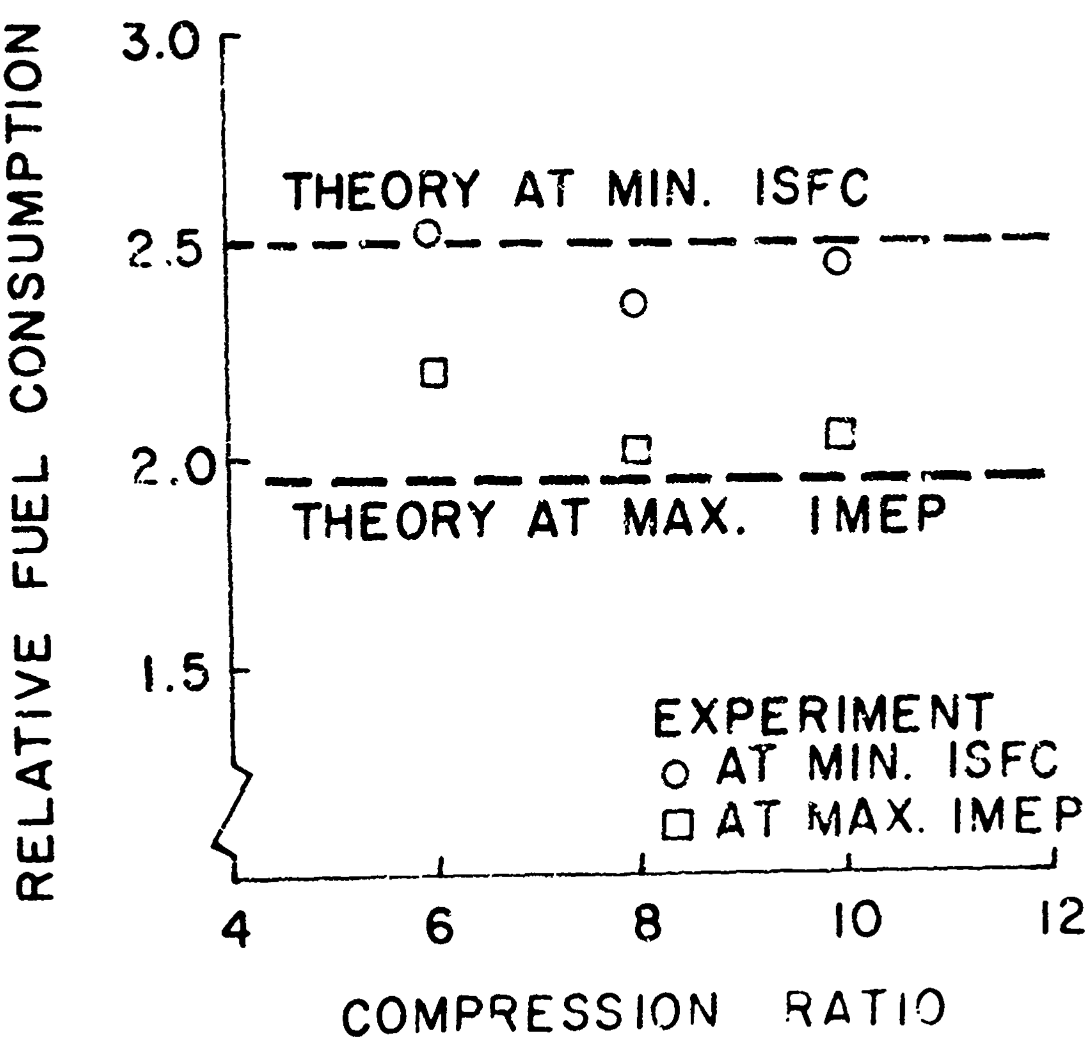


FIGURE 27

INFLUENCE OF COMPRESSION RATIO ON FUEL CONSUMPTION

R.P.M. – 1800
MANIFOLD PRESSURE - 30" Hg.
ϕ = EQUIVALENCE RATIO

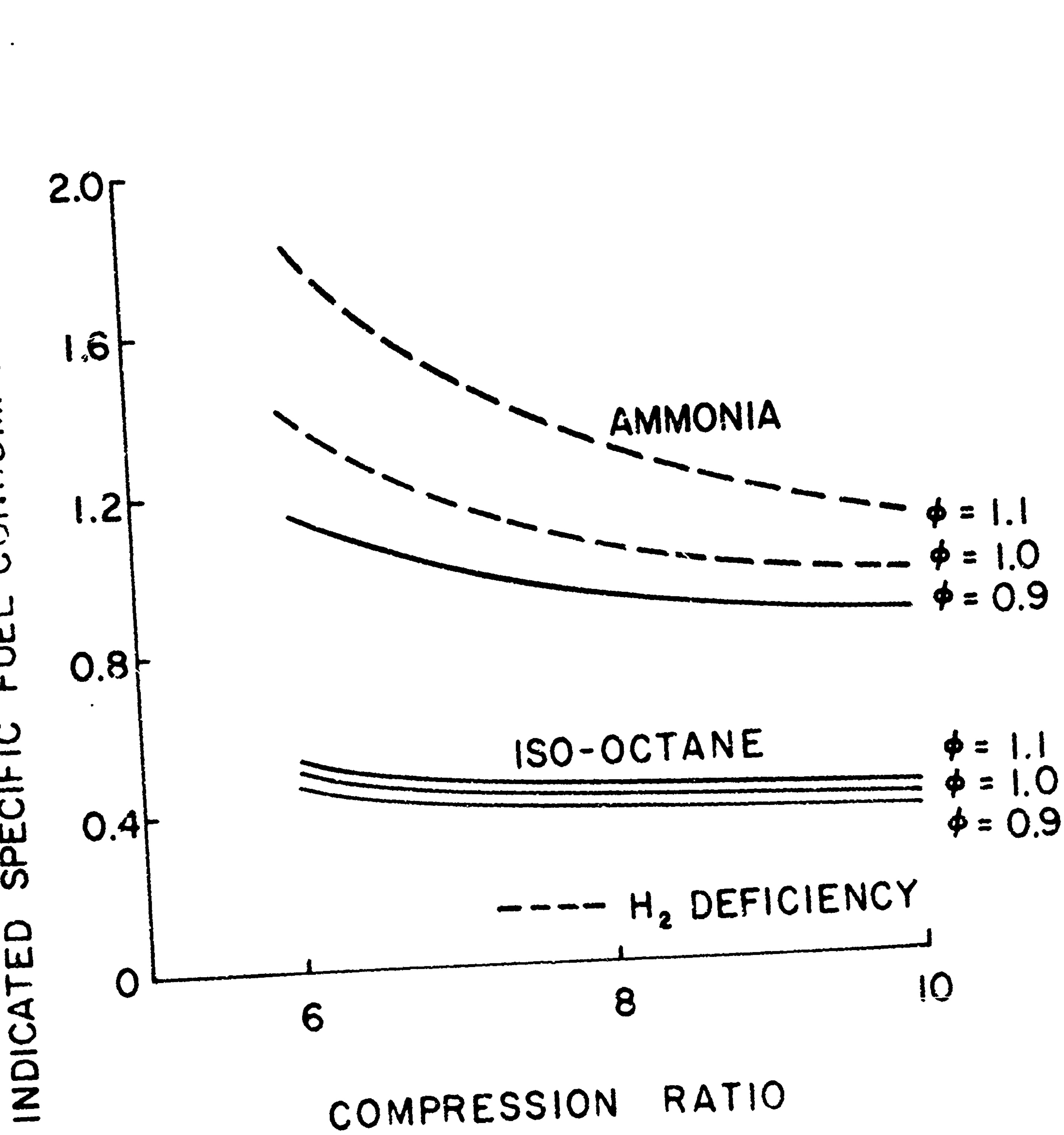


FIGURE 28

INFLUENCE OF COMPRESSION RATIO ON RELATIVE FUEL CONSUMPTION

R.P.M. — 1800
MANIFOLD PRESSURE — 30" Hg.
ϕ = EQUIVALENCE RATIO

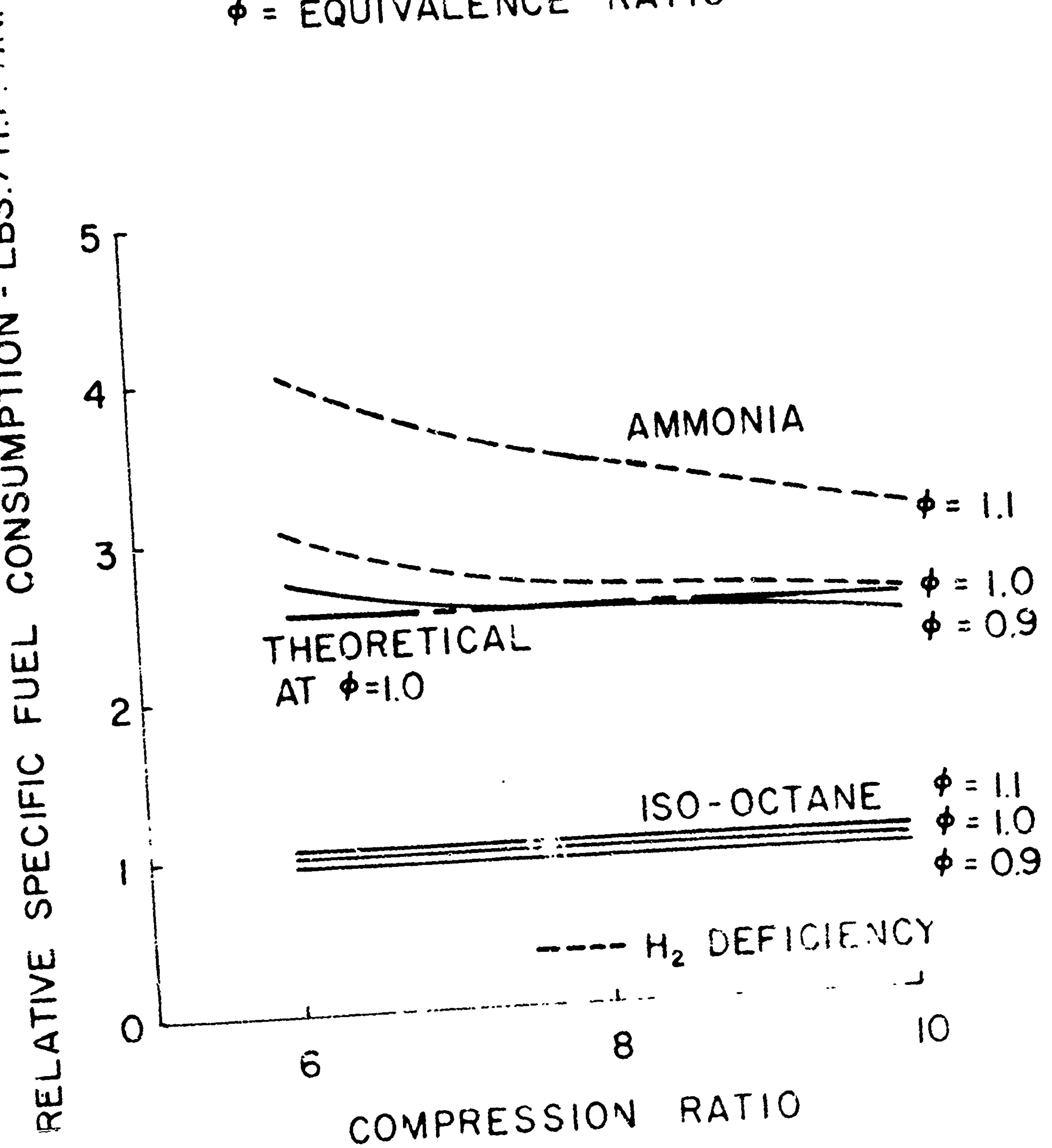


FIGURE 29

EFFECT OF PREDISSOCIATION OF AMMONIA ON POWER OUTPUT

COMPRESSION RATIO - 8
R.P.M. - 1800
MANIFOLD PRESSURE - 30" Hg.

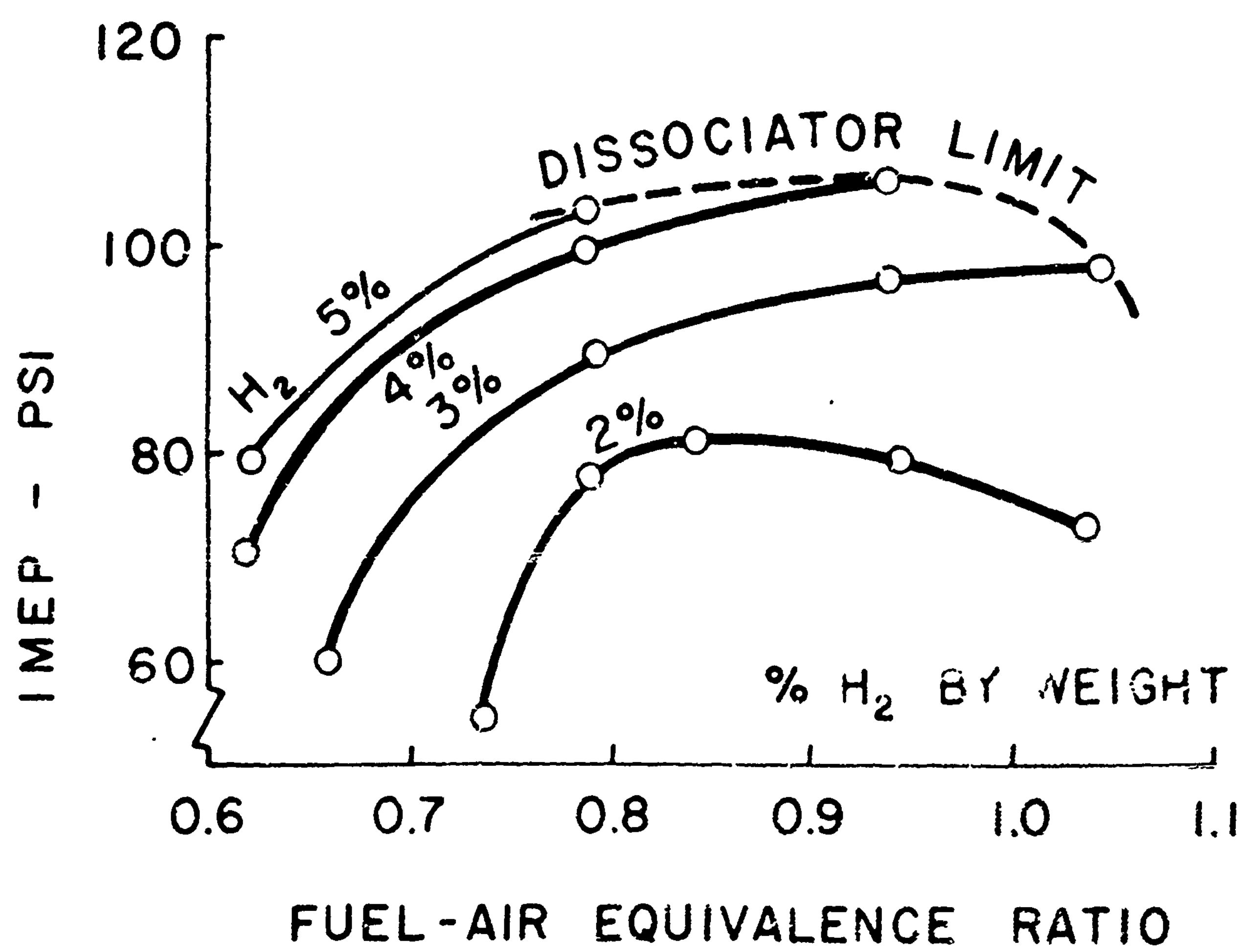


FIGURE 30

INFLUENCE OF SPARK ADVANCE ON POWER OUTPUT AS A FUNCTION OF MIXTURE STRENGTH

COMPRESSION RATIO - 8
R.P.M. - 1800
MANIFOLD PRESSURE - 30" Hg.

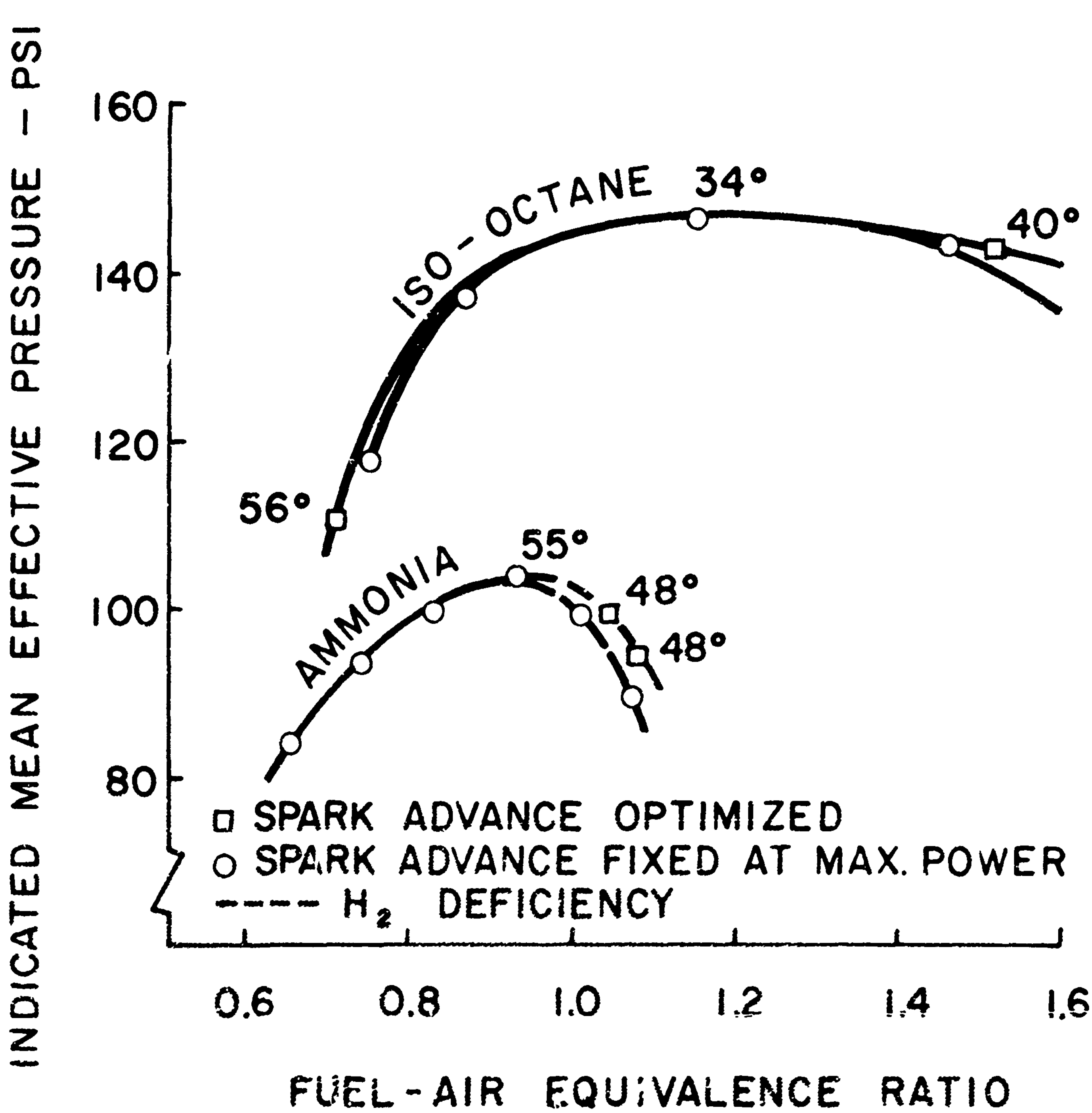


FIGURE 31

INFLUENCE OF SPARK ADVANCE ON POWER OUTPUT AS A FUNCTION OF MIXTURE STRENGTH

COMPRESSION RATIO - 8
R.P.M. - 1800
MANIFOLD PRESSURE - 21" Hg.

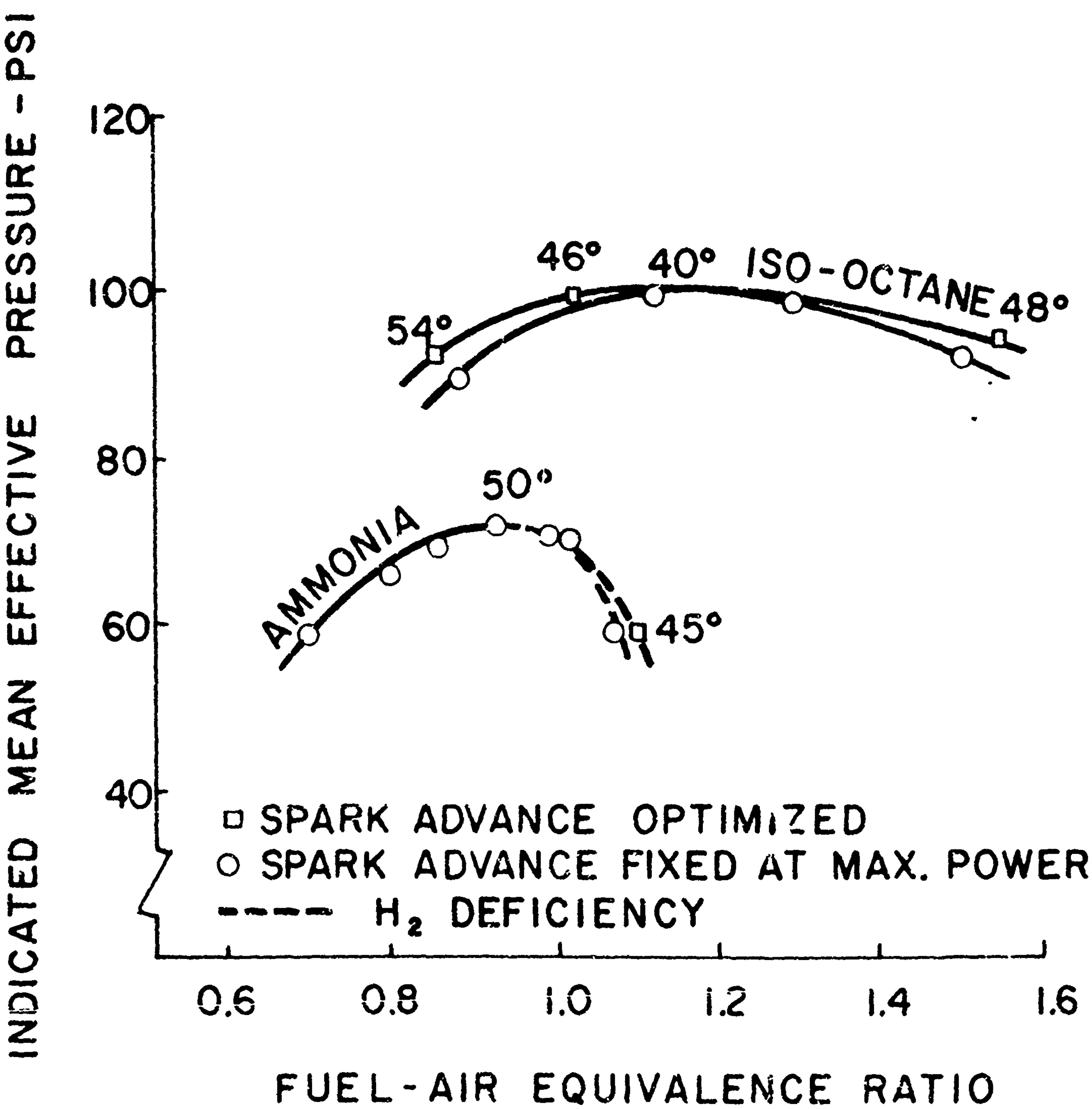
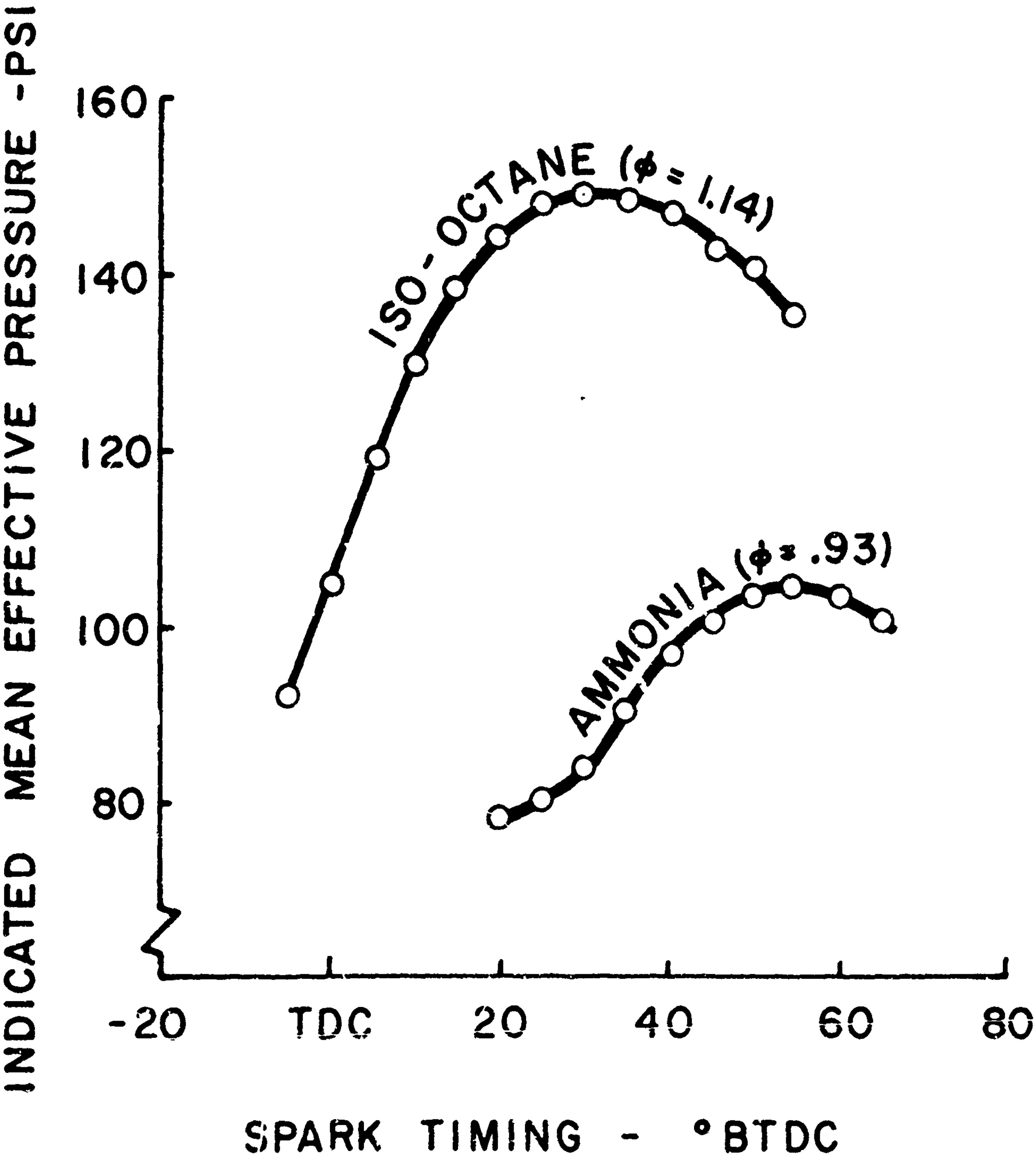


FIGURE 32

INFLUENCE OF SPARK TIMING ON POWER OUTPUT AT FUEL-AIR EQUIVALENCE RATIO FOR MAXIMUM POWER

COMPRESSION RATIO - 8
R.P.M. - 1800
MANIFOLD PRESSURE - 30" Hg.



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2. H. K. Newhall, "Calculation of Engine Performance Using Ammonia Fuel, I. Otto Cycle," University of California, College of Engineering, Office of Research Services. Technical Report No. 2, Ammonia Fuels Studies, 1965.

3. "ASTM Manual for Rating Aviation Fuels by Supercharge and Aviation Methods", American Society for Testing Materials, Philadelphia, Pa., 1958.

ACKNOWLEDGEMENT

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Messrs. G. James, T. Maguire, S. Samuelson and R. Sutton were instrumental in the acquisition of test data. Mr. H. Stewart was responsible for instrumentation and developed the liquid fuel flow meter. Mr. J. DeCosta was responsible for maintenance of the mechanical integrety of the engine and auxillary systems.

APPENDIX

The appendix includes the test results obtained from a chemical and spectrochemical analysis of engine oil and a tabulation of the engine performance test data. The analysis was performed by Chevron Research Company on a gratis basis.

CHEVRON RESEARCH COMPANY
RICHMOND, CALIFORNIA

ANALYSIS OF OIL FROM
CFR ENGINE RUN ON
AMMONIA FUEL

FILE 100.01
SEPTEMBER 1, 1965

Author - N. R. Parkin

Analysis of a sample of oil from a CFR engine run on ammonia fuel at the University of California is shown in the attached table. The total operating time on the oil was 52 hours. The engine ran for 16 hours on isooctane fuel, and ammonia was the fuel for 36 hours. Super RPM DELO Special Oil (SAE 50) was the crankcase lubricant, and botn used and fresh oil samples were analyzed.

The results show a negligible change in gravity, flash point, viscosity, and neutralization numbers which indicate that the oil is not deteriorating during operation with ammonia fuel. The spectrochemical analysis shows an abnormally high amount of copper and lead in the used oil, indicating the corrosion of a copper-lead bearing. Apparently, the use of ammonia fuel is resulting in contamination of the oil with products that are corrosive to copper and lead.

:slc
Encl. - Table

OIL ANALYSES
OIL FROM AMMONIA-FUELED ENGINE

	Used Oil	Fresh Oil
Gravity, °API	25.8	26.1
Flash Point, °F	490	485
Viscosity at 100°F, SUS	1288	1232
Viscosity at 210°F, SUS	97.1	94.4
Neutralization No., ASTM D 664		
pH	7.61	7.22
Total Acid No.	1.6	2.4
Total Base No.	2.3	2.5
Carbon Residue, %	1.00	1.03
Coagulated Pentane Insolubles, %	< 0.01	< 0.01
Coagulated Benzene Insolubles, %	< 0.01	< 0.01
Calcium, %	0.153	0.184 (Typical)
Zinc, %	0.077	0.084 (Typical)
Phosphorus, %	0.089	0.095 (Typical)
Spectrochemical Analysis, ppm		
Aluminum	1.7	
Barium	5.1	
Chromium	0.4	
Copper	51	
Iron	12	
Magnesium	1.7	
Molybdenum	1.6	
Sodium	10	
Lead	130	
Silicon	1.7	
Tin	8.3	

TABLE I

Experimental Data, Iso-Octane

Comp. Ratio	RPM	Manifold Pressure (In. of Hg.)	Fuel-Air Equivalence Ratio (Ø)	Indicated Mean Effective Pressure (Psi)	Spark Advance (Degrees BTDC)
10	1800	30	.80	128	28
10	1800	30	.75	106	28
10	1800	30	.86	143	28
10	1800	30	.91	147	28
10	1800	30	.97	152	28
10	1800	30	1.10	157	28
10	1800	30	1.17	158	28
10	1800	30	1.29	156	28
10	1800	30	1.44	151	28
10	1800	30	.95	152	28
8	1800	30	1.15	147	34
8	1800	30	.76	118	34
8	1800	30	.88	138	34
8	1800	30	1.46	145	34
6	1800	30	1.08	136	43
6	1800	30	1.51	129	43
6	1800	30	.93	129	43
6	1800	30	1.41	133	43
8	1300	30	1.09	143	25
8	1300	30	1.61	124	25
8	1300	30	1.39	139	25
8	1300	30	.95	135	25
8	1300	30	.68	59	25
8	1000	30	1.30	137	22
8	1000	30	1.00	137	22
8	1000	30	1.13	138	22
8	1000	30	.70	78	22
8	1000	30	.75	110	22
8	1000	30	1.54	129	22
8	1800	21	1.13	99	40
8	1800	21	.87	90	40
8	1800	21	1.11	98	40
8	1800	21	1.28	98	40
8	1800	21	1.48	92	40
8	1800	30	1.05	147	34
8	1800	30	1.52	144	40
8	1800	30	1.68	136	46
8	1800	30	.71	110	56
8	1800	30	1.82	131	52

Comp. Ratio	RPM	Manifold Pressure (In. of Hg.)	Fuel-Air Equivalence Ratio (Ø)	Indicated Mean Effective Pressure (psi)	Spark Advance (Degrees BTDC)
8	1800	21	1.02	99	46
8	1800	21	.83	91	54
8	1800	21	1.55	93	48
8	1800	21	1.70	84	53
8	1800	30	1.14	148	34
8	1800	30	1.14	149	30
8	1800	30	1.14	148	25
8	1800	30	1.14	143	20
8	1800	30	1.14	138	15
8	1800	30	1.14	130	10
8	1800	30	1.14	119	5
8	1800	30	1.14	105	0
8	1800	30	1.14	91	5 (ATDC)
8	1800	30	1.14	148	40
8	1800	30	1.14	143	45
8	1800	30	1.14	141	50
8	1800	30	1.14	135	55

TABLE II

Experimental Data, Ammonia

Comp. Ratio	RPM	Manifold Pressure (In. of Hg.)	Fuel-Air Equivalence Ratio (Ø)	Indicated Mean Effective Pressure (Psi)	Spark Advance (Degrees BTDC)	% H_2 Dissoc-iation
10	1800	30	.59	83	53	5+
10	1800	30	.75	100	53	5+
10	1800	30	.88	108	53	3.6
10	1800	30	.89	110	53	3.8
10	1800	30	.95	114	53	3.3
10	1800	30	1.01	111	53	3.1
10	1800	30	1.11	101	53	2.3
8	1800	30	.65	84	55	5+
8	1800	30	.72	93	55	4.8
8	1800	30	.83	100	55	3.5
8	1800	30	.93	104	55	3.7
8	1800	30	1.01	99	55	3.7
8	1800	30	1.08	90	55	3.3
6	1800	30	.74	74	55	4.4
6	1800	30	.80	82	55	4.0
6	1800	30	.86	85	55	4.2
6	1800	30	.95	82	55	3.7
6	1800	30	1.06	78	55	3.1
8	1300	30	.62	80	55	5+
8	1300	30	.74	90	55	5+
8	1300	30	.80	95	55	4.0
8	1300	30	.90	103	55	4.8
8	1300	30	.96	104	55	4.8
8	1300	30	1.03	105	55	4.3
8	1300	30	1.04	102	55	4.4
8	1300	30	1.18	99	55	5+
8	1300	30	1.31	89	55	4.6
8	1000	30	.67	82	35	5+
8	1000	30	.77	89	35	5+
8	1000	30	.89	97	35	5+
8	1000	30	.98	100	35	5+
8	1000	30	1.07	99	35	5+
8	1000	30	1.22	93	35	5+
8	1800	21	.70	59	50	5+
8	1800	21	.80	65	50	5+
8	1800	21	.87	69	50	5+
8	1800	21	.93	72	50	5.0
8	1800	21	.99	70	50	5+
8	1800	21	1.01	69	50	5+
8	1800	21	1.06		50	4.8

Comp. Ratio	RPM	Manifold Pressure (In. of Hg.)	Fuel-Air Equivalence Ratio (Ø)	Indicated Mean Effective Pressure (psi)	Spark Advance (Degrees BTDC)	% H_2 Dissociation
8	1800	30	.93	105	55	3.8
8	1800	30	.93	103	60	3.8
8	1800	30	.93	101	65	3.8
8	1800	30	.93	103	50	3.8
8	1800	30	.93	100	45	3.8
8	1800	30	.93	97	40	3.8
8	1800	30	.93	92	35	3.8
8	1800	30	.93	84	30	3.8
8	1800	30	.93	80	25	3.8
8	1800	30	.93	77	20	3.8
8	1800	30	1.05	101	48	3.7
8	1800	30	1.08	94	48	3.5
8	1800	21	1.06	62	45	4.8
8	1800	30	.79	103	55	5.0
8	1800	30	.62	79	55	5.0
8	1800	30	.94	106	55	4.0
8	1800	30	.79	100	55	4.0
8	1800	30	.62	70	55	4.0
8	1800	30	.94	97	55	3.0
8	1800	30	1.04	98	55	3.0
8	1800	30	.79	89	55	3.0
8	1800	30	.66	58	55	3.0
8	1800	30	.94	78	55	2.0
8	1800	30	.79	78	55	2.0
8	1800	30	.74	54	55	2.0
8	1800	30	.84	80	55	2.0
8	1800	30	1.04	75	55	2.0