CARBURETION AND IGNITION REQUIREMENTS OF A SPARK-IGNITION ENGINE

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Krishna Raju

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This is to certify that the

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presented by

Krishna Raju

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Louis L. Otto
Major professor

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OF A SPARK-IGNITION ENGINE

by

KRISHNA RAJU

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ABSTRACT

This work is intended to evaluate experimentally the carburetion and ignition requirements of a spark-ignition, single-cylinder internal-combustion engine. The selection of a single-cylinder engine for investigation avoided the effects of uneven mixture distribution and at the same time conformed to the general method of approach to this type of problem. The carburetion requirements (air-fuel mixture ratio as a function of air-flow rate) and ignition requirements (optimum spark-advance angle) of an internal-combustion engine must be known to permit the proper calibration of these accessory items of the engine. These requirements are affected directly by engine speed, relative engine load, engine compression ratio, combustion chamber temperature, and entering air temperature. In this investigation, the effect of several different engine loads and engine speeds was determined at constant compression ratio, engine jacket temperature and entering air temperature.

A brief summary of results is as follows: optimum spark advance decreased with increasing load. It also decreased with the reduction of engine speed. However, the decrease of optimum spark advance was less rapid with increasing speed. Maximum mean-effective pressure occurred at the same air-fuel ratio regardless of throttle position. The operation at high relative load produced the lowest brake-specific fuel consumption value and the leanest maximum-economy air-fuel ratio. In addition, the useful range of air-fuel ratio became narrower as the load was reduced. The maximum-economy and maximum-power mixture ratios

intersected at idling points and thus represent the intersections of the two mixture conditions for each constant speed.

The interpretation of the results is as important as the study itself. An extensive survey of the literature revealed the lack of uniform presentation of results of ignition requirements of spark-ignition engines. The method of presentation adopted by Otto (7) was used here as it represented the results of a more acceptable and understandable way.

The experimental results, in general, are in agreement with the results obtained by previous investigators (1, 2, 6, 7, 8, and 10) over a wide range of speeds and loads.

In this investigation the experimental set-up needed, in addition to the basic test equipment, was the design and construction of air-flow and fuel-flow measuring equipment. The accuracy of the test results depend mainly upon the accuracy with which the actual conditions are indicated by the instruments. Instrumentation thus became an important aspect of this investigation. A considerable amount of time was spent on instrumentation; and the collection of data became routine work. With a few more additional items of equipment, the same set-up could be used to further investigate the carburetion and ignition requirements at different entering air temperatures and engine jacket temperatures.

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INTRODUCTION

This work is intended to evaluate experimentally the carburation and ignition requirements of a spark-ignition, single-cylinder internal-combustion engine. The carburation requirements (air-fuel mixture ratio-as a function of air-flow rate) and ignition requirements (optimum spark-advance angle) of an internal-combustion engine must be known to permit the proper calibration of these accessory items of the engine. These requirements are affected directly by engine speed, relative engine load, engine compression ratio, combustion chamber temperature, and entering air temperature. In this investigation the effect of several different engine loads and engine speeds were determined at constant engine-jacket temperature, compression ratio, and entering-air temperature.

An enormous amount of theoretical and experimental research has been done on the subject of combustion. In spite of this effort, however, knowledge of the mechanism of flame propagation remains still incomplete and one must depend on experimental evidence rather than on theory in examining those factors which govern the character and the speed of flames in gaseous mixtures (9).

In a spark-ignition engine the flame progresses in a turbulent mixture. The turbulence or state of agitation of the mixture, produced by the induction process and the piston movement, increases the rate of burning, and the following relation can be written for the velocity of the flame (15)

$$U_T = U_n \sqrt{1 + B \left(\frac{U'}{U_n} \right)^2} \quad \text{where}$$

U_T is the effective velocity of the turbulent flame, U_n the laminar flame velocity, U' the turbulent intensity, and B a constant introduced by Shelkin and presumed to be of the order of unity. The first step in the prediction of the flame velocity in the spark-ignition engine is the calculation of the laminar-flame velocity taking into account the variation of initial temperature of the mixture during flame propagation. There are a large number of approximate equations which approach the problem of flame propagation from various aspects: thermal conductivity including chemical reaction kinetics, diffusion of chain carriers of oxidation reaction, modification of pressure drop in the flame. Only one model, the Semenoff thermal model, has up to this time been used in engine calculations. In this theory all the hydrocarbon molecules of the fuel are supposed to be cracked to carbon monoxide and hydrogen in the neighborhood of the flame front, so that the oxidation reactions are limited to the union of oxygen and carbon monoxide or hydrogen in presence of water vapor.

The general approximate relation may be written

$$U_n = \frac{2 \lambda_f \int_0^{T_f} W \, dT}{d_o \, \rho_o \, C_p \, (T_f - T_o)} \left(\frac{n_1}{n_2} \right)^m \left(\frac{\lambda}{DC_p} \right)_f^m \quad \text{where}$$

λ_f is the thermal conductivity at the flame temperature; W the reaction rate (molecules reacting per cu. cm. per sec.); ρ_o the density of the mixture at initial temperature (grams per cu. cm.); ρ_f the density of the mixture at flame temperature; d_o fuel concentration at the initial temperature (molecules of fuel per cu. cm. of mixture); C_p the mean specific heat T_o to T_f (cal. per gram deg. K); C_{pf} the specific heat of flame temperature (cal. per gram per deg. K.); D the diffusion coefficient at flame temperature (sq. cm. per sec.); T_o the initial temperature of the mixture; T_f the computed temperature of the equilibrium flame (deg. K.); n_1/n_2 moles of reactants per molecule of products from the stoichiometric equation; and m the molecularity of flame reaction.

As the temperature and pressure change during the flame propagation, this period has to be divided into small fractions of constant initial temperature and pressure, in order to evaluate the laminar flame velocity for these conditions. Such calculations were made first by Karpoff and the resulting data for the laminar-flame velocity were reasonably close to the values given by actual experiment on an engine.

The different theories of flame propagation are useful in the prediction of the relative flame-speed variation with the parameters such as initial temperature, oxygen concentration, etc.

Until now a single-cylinder indicator or a stroboscopic card has been used to give only qualitative information on the influence of parameters such as the spark advance, the mixture strength, or the speed of the engine. Ricardo divides the whole combustion process into two quite distinct stages (13). The first is assumed to be a chemical process controlled by the effect of temperature and pressure on the acceleration of oxidation. The second is considered as pure mechanical evolution. Experiments made more recently by von Hoogstraten seem to prove the possibility of separating with good accuracy the main-combustion phase and the delay period. The delay period depends on mixture strength, temperature and density. Russian investigators (14) introduced a third stage corresponding to the completion of combustion. The first stage is felt to be controlled by the rate of chemical reaction and by small-scale turbulence. In the second stage with a high degree of turbulence, these authors think that the chemical factor can be neglected and the combustion rate depends upon the large-scale turbulence. The completion of combustion in the third stage is governed by the chemical factor.

The burning rate of the homogeneous air-fuel mixture in a spark-ignition engine is presumed to be controlled by the spark setting. Unfortunately, this control is only apparent and has an action on the mean diagram but not on each separate cycle (15). The spread of the pressures for the same crankshaft position is generally assumed to be produced by scattering of spark setting. It is considered also that this instability increases with the speed of the engine in the same way as the spark position. Experimental evidence also proves that the spark setting is not entirely responsible for the combustion rate dispersion. The chemical nature of the fuel seems to have some action on the pressure spread during the normal combustion. The burning rate of fuel is an important factor in all kinds of combustion (15).

Abnormal combustion, known also as detonation, is a violent, explosive reaction of oxygen and fuel which causes a high momentary pressure unbalance in the engine. This pressure differential gives rise to a pressure wave which travels through the combustion chamber at a velocity dictated by the density of gases and which, upon striking the combustion chamber walls, is reflected, with a resulting frequency of the detonating wave depending upon the physical dimensions of the chamber. By lowering compression ratios, lower temperatures were achieved with detonation eliminated but efficiency was sacrificed. Finally, the higher the temperature of the unburned mixture and the longer the time it is held at high temperatures, the more severe will be the detonation (11).

Extensive research in recent years has shown that early chemical reactions precede knock in an internal combustion engine. These reactions take place in the air-fuel mixture ahead of knock. These are generally ref rred to as precombustion or preflame reactions.

The chemical reactions preceding knock can be divided into two classes: the first type includes those reactions which have been called precombustion reactions and which are characterized by the evolution of relatively large quantities of light and heat with consequent rise in pressure but which do not result in complete combustion. The second type includes those which can not be detected by any large scale energy release and are referred to as small-scale or precool-flame reactions.

The first type of reactions, mentioned above, is characterized by the thermoneutral formation of pro knock compounds. The second class of reactions which occur at somewhat higher temperatures and are accompanied by radiation and by the evolution of considerable quantities of heat, does not increase the knocking tendency of an air-fuel mixture (16).

In a spark-ignition engine, flame speed which governs the normal combustion process, is affected by several operating variables: mixture ratio, initial pressure, initial temperature, residual gases, humidity, engine speed, and spark advance. In our present investigation, the effect of mixture ratio, engine speed, spark advance and the relative concentration of residual gases on combustion process has significant importance.

Effect of fuel-air ratio on flame speed. The work done by many investigators has proven that flame speed increases as fuel-air ratio is increased from a very lean mixture up to about the best power mixture ratio and then decreases (9). The decrease of flame speed on either side of the most rapid burning mixture can be attributed to the mechanical and thermal barrier action caused by an excess of inert nitrogen or of fuel vapor.

Effect of engine speed. The flame speed increases rapidly with engine r.p.m. This effect was first measured by Marvin and Best, although it had been well known that the spark-advance required in engine did not increase nearly in proportion to r.p.m. The increase of flame speed with increasing engine speed is undoubtedly due to the marked effect of turbulence, which has been noted in many bomb experiments and in engines when the turbulence was varied independent of speed.

Until experimental technique developed to the point where the progress of the flame would be observed in the cylinder of an engine in operation, it was believed that the turbulence might be of such a nature as to break up the flame front and scatter separate flame nuclei throughout the cylinder volume. Actual measurements of flame movement show that is not the case. The increase of flame velocity may be due to the effectiveness of small vortices in accelerating the transfer of heat and energy-rich molecules to the unburned gases or to an increase in the area of the flame front or both.

In an engine the chief operating variable affecting the turbulence is inlet velocity which is roughly proportional to piston speed at wide open throttle. Without the increase in flame velocity caused by increasing turbulence, spark-ignition engines would not run at the very high piston speeds used in some present-day engines (9).

Effect of spark advance. The variation of flame speed with spark timing is probably a pressure effect, the maximum flame speed occurring when the piston reaches top center in about the middle of the combustion process.

Residual gases. Studies have indicated that an increase in the residual gas content decreases flame speed, which agrees with the results obtained from combustion bombs (9). Residual gases are almost completely inert. They act as a mechanical and thermal barrier and, consequently, diminish the diffusion rate of active molecules and the conduction and radiation of heat energy from actively burning portions to the unburned mixture ahead of the flame. An increase in the proportion of residual gases in the overall mixture will cause a decrease in the rate of flame propagation.

In addition to studies of flame travel and pressure, chemical sampling and spectroscopic observations have been made in investigating the combustion process. Chemical sampling has not yielded any important contributions to our knowledge of combustion process, although it has confirmed the theory that most of the combustion takes place in or near the flame front.

Spectroscopic research has yielded interesting data on combustion temperature and on the process of transfer of heat to the cylinder walls by radiation (12).

THE PROBLEM

A spark-ignition engine follows the following cycle of events: induction of a previously prepared air-fuel mixture into the cylinder, compression of the mixture to a high temperature and pressure, ignition of the compressed mixture at some predetermined time and point and the subsequent orderly burning of the entire mixture, expansion of the combustion products to a lower temperature and pressure, release and partial expulsion of the products of combustion to make room for the induction of the next fresh charge. The above cycle of events include two very important fundamental characteristics which formed the subject matter for this investigation. The first of these is the preparation of the mixture of air and fuel prior to its induction into the cylinder, and the second is the ignition of the compressed mixture at a predetermined time in the cycle and at a predetermined point within the combustion chamber (7).

The first characteristic necessitates the incorporation into the engine of a device which will prepare air-fuel mixtures of proper proportions, and distribute them to the various cylinders of the engine. The design, operation and adjustment of this device require a knowledge of the air-fuel requirements or the carburetion requirements of an engine.

The second characteristic requires the need of a device which will raise to its ignition temperature a small localized portion of the mixture at the proper location and at the proper predetermined time.

The design, operation and adjustment of the ignition device require a knowledge of the ignition timing requirements of an engine (7). It is the purpose of this paper to investigate experimentally these two fundamental requirements of a spark-ignition engine.

Considerable work has been done on carburetion and ignition requirements of a multi-cylinder spark-ignition engine by various investigators in the past. In a multi-cylinder engine, the manifolding system usually encounters difficulty in furnishing the same mixture ratio to each of the several cylinders. Hull and Parker (5) found the distribution error in a four cylinder engine increased at low-power output. In the final adaptation of the manifolding system to a given engine, a compromise is made between the requirements of the individual cylinders of the engine and the limitations or modifications imposed on these requirements by deficiencies of the distribution system and of the fuel used. It is the purpose of this paper to present and discuss results obtained in an experimental investigation of a single-cylinder engine. The selection of a single-cylinder engine for investigation avoided the effects of uneven mixture distribution and at the same time conformed to the general method of approach to the problem. The major part of the investigation was instrumentation and the interpretation of the results in a suitable form. An extensive survey of literature revealed the lack of uniform presentation of results of ignition requirements of a spark-ignition engine. The method described by Otto (7) was adopted here. An unpublished manuscript prepared by Dr. Otto greatly helped as a guide throughout this study.

Prior to the beginning of each test run the following information was recorded: date, name of engine, fuel used, manometer zero and size of orifice used in the air chamber.

The first step was to open the water tap for circulation of water through the condenser. The battery was then connected, fuel supply line kept on and the electric fuel pump started.

The dynamometer was then used as a motor to turn the engine over. As soon as the engine was turning over, the ignition switch was turned on while the throttle was kept near the idling position. The mixture control was set rich during this period. As soon as the engine fired continuously the field strength in the motor was increased and the motor used as a generator driven by the engine. The load on the dynamometer was then adjusted by the rheostat to the approximate speed desired.

Load was determined at the full-throttle opening and closed-throttle position for a constant speed. From the difference between these two loads, the different loads were determined: namely, full, three-quarter, one-half, one-quarter, and no load for that particular speed. In every case, optimum spark-advance for maximum power was used.

Fuel flow was varied by turning the needle valve in smaller increments. As mentioned above, the spark advance was brought to optimum point before any readings were taken.

The runs were made at 600, 1,200, 1,800, and 2,400 revolutions per minute and at full, three-quarters, one-half, one-quarter, and no load conditions for different air-fuel ratios in each case with 96 octane fuel.

TABLE 1

RUNNING LOG OF AIR FLOW - FUEL FLOW - SPARK ADVANCE

CHRISTIE SINGLE CYLINDER

2400 RPM

BEAM LOAD LBS.	SPARK ADV. °BTC	BHP	B.M.E.P.	B.S.F.C.	AIR FLOW #/HR.	FUEL FLOW #/HR.	A/F
14.2	54	6.815	67.6	1.24	70	8.43	8.31
15.0	52	7.20	71.5	0.910	70	6.55	10.7
14.3	52	6.86	68.1	0.776	70	5.32	13.15
13.0	53	6.25	62.0	0.747	70	4.67	15.0
11.4	59	5.47	54.4	0.753	70	4.12	17.0
9.3	65	4.46	44.3	0.810	70	3.61	19.4
11.5	56	5.52	54.75	1.28	60	7.06	8.5
12.2	54	5.85	58.10	0.951	60	5.56	10.8
11.6	54	5.57	55.25	0.862	60	4.80	12.5
10.5	58	5.05	50	0.834	60	4.21	14.25
8.4	64	4.04	40	0.857	60	3.46	17.35
6.2	69	2.98	29.58	1.032	60	3.08	19.5
9.2	57	4.42	43.85	1.285	50	5.68	8.81
9.8	56	4.70	46.85	0.993	50	4.67	10.7
9.5	56	4.56	45.25	0.926	50	4.22	11.84
8.6	57	4.13	41.0	0.908	50	3.75	13.3
6.7	63	3.22	31.9	0.925	50	2.98	16.8
5.1	70	2.45	24.29	1.07	50	2.62	19.1
5.3	60	2.54	25.21	1.77	40	4.49	8.9
6.1	59	2.93	29.05	1.28	40	3.74	10.7
5.9	60	2.84	28.1	1.15	40	3.28	12.2
5.3	62	2.54	25.21	1.13	40	2.87	13.94
4.3	65	2.06	20.45	1.22	40	2.51	15.9
3.2	73	1.548	15.28	1.43	40	2.21	18.1

No Load Air Flow = 30 #/hr. 96 octane fuel.

TABLE 2

RUNNING LOG OF AIR FLOW - FUEL FLOW - SPARK ADVANCE

CHRISTIE SINGLE CYLINDER

1800 RPM

BEAM LOAD LBS.	SPARK ADV. °BTC	BHP	B.M.E.P.	B.S.F.C.	AIR FLOW #/HR.	FUEL FLOW #/HR.	A/F
18.3	50	6.59	87.1	1.05	58	6.9	8.4
19.3	48	6.95	91.9	0.766	58	5.32	10.9
18.8	48	6.75	89.5	0.696	58	4.71	12.3
17.4	49	6.26	82.9	0.649	58	4.06	14.3
14.9	51	5.36	71	0.639	58	3.42	16.95
11.9	59	4.29	56.6	0.695	58	2.98	19.45
14.5	54	5.22	69	1.12	48.5	5.85	8.3
15.5	52	5.57	73.8	0.799	48.5	4.45	10.9
14.9	53	5.36	71.0	0.740	48.5	3.97	12.2
13.4	53	4.82	63.8	0.701	48.5	3.38	14.35
11.1	56	4.0	52.9	0.700	48.5	2.80	17.3
8.5	63	3.06	40.5	0.789	48.5	2.41	20.1
10.8	56	3.88	51.5	1.18	39.0	4.59	8.5
11.5	55	4.14	54.75	0.872	39.0	3.61	10.8
10.9	55	3.92	52.0	0.821	39.0	3.22	12.1
10.1	56	3.64	48.1	0.789	39.0	2.87	13.6
8.2	58	2.96	39.01	0.801	39.0	2.37	16.5
6.1	68	2.2	29.02	0.896	39.0	1.97	19.8
6.7	59	2.42	31.9	1.37	29.5	3.32	8.9
7.3	58	2.63	34.8	1.04	29.5	2.73	10.8
7.2	58	2.59	34.3	0.962	29.5	2.49	11.8
6.7	60	2.42	31.95	0.921	29.5	2.23	13.2
5.6	64	2.02	26.64	0.901	29.5	1.82	16.2
4.2	75	1.51	20.0	1.08	29.5	1.63	18.1

No Load Air Flow = 20 #/hr.

Bad valve bounce at 1800 rpm.

TABLE 3

RUNNING LOG OF AIR FLOW - FUEL FLOW - SPARK ADVANCE

CHRISTIE SINGLE CYLINDER

1200 RPM

BEAM LOAD LBS.	SPARK ADV. °BTC	BHP	B.M.E.P.	B.S.F.C.	AIR FLOW #/HR.	FUEL FLOW #/HR.	A/F
20.4	41	4.9	97.2	0.940	40	4.60	8.7
21.0	40	5.05	100	0.714	40	3.60	11.1
20.7	41	4.97	98.6	0.635	40	3.16	12.6
19.1	43	4.58	91.0	0.556	40	2.65	15.1
16.9	47	4.05	80.5	0.573	40	2.32	17.2
14.8	51	3.56	70.5	0.596	40	2.12	18.85
16.3	47	3.91	77.6	0.925	33	3.62	9.12
16.7	46	4.0	79.5	0.743	33	2.97	11.1
16.5	47	3.96	78.6	0.685	33	2.71	12.15
15.2	50	3.64	72.5	0.610	33	2.22	14.84
13.1	53	3.14	62.5	0.605	33	1.90	17.4
11.4	58	2.74	54.3	0.625	33	1.71	19.3
10.9	53	2.62	52	1.15	26.5	3.01	8.8
11.4	51	2.74	54.3	0.870	26.5	2.38	11.2
11.0	51	2.63	52.4	0.800	26.5	2.11	12.54
10.2	53	2.45	48.6	0.755	26.5	1.85	14.3
9.1	58	2.21	43.4	0.737	26.5	1.63	16.25
7.1	62	1.70	33.8	0.830	26.5	1.41	18.80
7.3	57	1.75	34.8	1.27	19.5	2.22	8.8
7.6	55	1.84	36.2	0.956	19.5	1.76	11.1
7.2	55	1.73	34.3	0.875	19.5	1.51	12.9
6.1	58	1.46	29.0	0.876	19.5	1.28	15.23
5.2	63	1.25	24.8	0.93	19.5	1.16	16.8
4.2	70	1.01	20.0	1.04	19.5	1.05	18.6

No Load Air Flow = 12.5 #/hr.

TABLE 4

RUNNING LOG OF AIR FLOW - FUEL FLOW - SPARK ADVANCE

CHRISTIE SINGLE CYLINDER

600 RPM

BEAM LOAD LBS.	SPARK ADV. °BTC	BHP	B.M.E.P.	B.S.F.C.	AIR FLOW #/HR.	FUEL FLOW #/HR.	A/F
19.6	29	2.36	93.4	0.966	19.0	2.28	8.35
20.3	28	2.44	96.6	0.714	19.0	1.74	10.9
20.0	29	2.40	95.25	0.610	19.0	1.46	13.0
19.1	33	2.30	91.0	0.557	19.0	1.28	14.84
17.2	38	2.06	82.0	0.554	19.0	1.14	16.7
14.4	42	1.73	68.5	0.614	19.0	1.06	17.9
15.9	37	1.91	75.75	1.00	15.5	1.92	8.1
16.5	36	1.98	78.5	0.724	15.5	1.43	10.85
16.2	37	1.945	77.1	0.627	15.5	1.22	12.7
14.9	42	1.79	71.0	0.570	15.5	1.02	15.2
12.7	47	1.525	60.5	0.590	15.5	0.90	17.2
10.5	52	1.26	50.0	0.675	15.5	0.85	18.2
12.0	43	1.44	57.2	0.993	12.0	1.43	8.4
12.2	42	1.465	58.15	0.785	12.0	1.15	10.4
11.9	44	1.43	56.7	0.670	12.0	0.96	12.5
9.9	52	1.19	47.15	0.630	12.0	0.75	16.0
8.5	55	1.02	40.5	0.695	12.0	0.71	16.9
7.7	60	0.925	36.63	0.714	12.0	0.66	18.2
7.3	49	0.876	34.8	1.255	9.0	1.10	8.17
7.7	48	0.925	36.63	0.897	9.0	0.83	10.85
7.1	49	0.854	33.8	0.774	9.0	0.66	13.6
5.3	58	0.636	25.24	0.850	9.0	0.54	16.6
3.8	64	0.456	18.1	1.100	9.0	0.50	18.0
2.8	66	0.336	13.32	1.46	9.0	0.49	18.32

No Load Air Flow = 5.5 #/hr.

RESULTS AND DISCUSSION[1]

Ignition Requirements. The value of optimum spark-advance could only be found experimentally for any set of given engine conditions as there was no theoretical method of determining it. It will be affected by engine design and operational factors. The more important engine design factors include compression ratio, size and shape of combustion chamber, location of spark plug in the combustion chamber, thickness and continuity of carbon deposits on the combustion chamber walls, and the degree of cooling to which the combustion chamber is subjected.

The operational factors that greatly affect the optimum spark-advance are relative load, mixture ratio, engine speed, and water-jacket temperature. Spark-advance angle is defined as the angular travel in degrees of the engine crankshaft between its position when the spark occurs in the cylinder and its position at top-dead-center of the piston. Optimum spark-advance is the angle which produces the greatest power output for a given set of other engine conditions. In order to discuss the effect of each of these variables on optimum spark-advance, it is necessary to keep the other three at a constant value.

Figure 1 shows the effect of relative load on optimum spark-advance at different speeds while temperature and maximum power-mixture ratio were held constant. As the load was increased, higher pressures and

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[1]In this chapter, extensive use has been made of the manuscript written by Dr. Otto (7) since little equivalent published material was available in the literature on this subject.

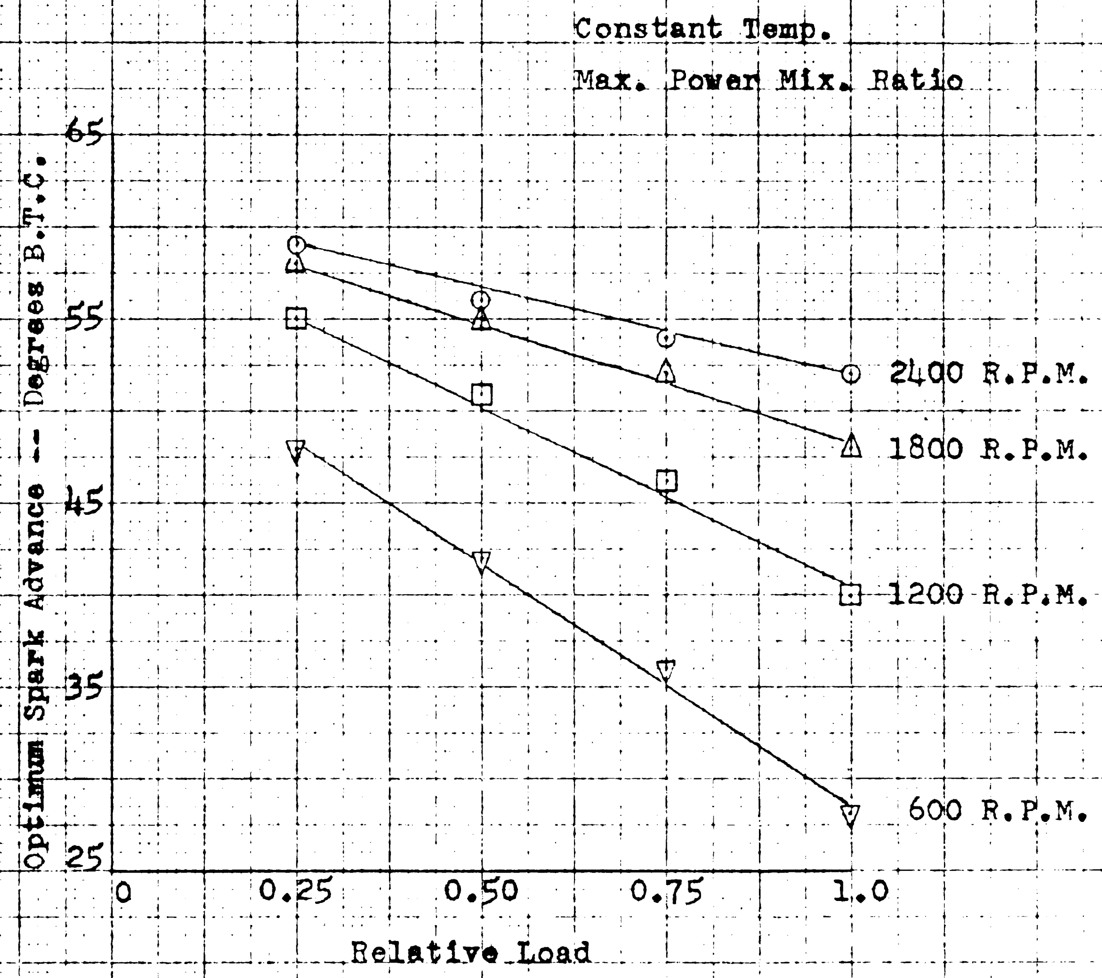
densities were attained at the end of the compression stroke, thus increasing the rate of combustion. Rate of combustion was also increased because of the reduced dilution effect in the combustion chamber. Optimum spark advance thus decreased with increase in relative load and less rapidly with increasing speed. This agrees well with the findings of Berry and Keggerreis (2).

The effect of air-fuel ratio on optimum spark-advance at different speeds and loads for each speed is shown in Figures 2 and 3, 4, 5, and 6, respectively, while the other operational factors were kept constant. The combustion rate was affected by the change in the purity of the mixture and to a small extent by the temperature change. At higher air-fuel ratios, excess air which did not enter the combustion process, separated the active portions of the mixture and absorbed heat from the combustion. Thus the two effects, one by acting as a mechanical barrier of inert distance; the other by absorbing a portion of the heat and keeping down the temperature of the reaction, held down the rate of combustion, thus requiring more spark-advance. The same was true with with rich mixtures but the effect was not so great. Thus, there was a particular mixture ratio between lean and rich which was subjected to the least amount of dilution effect that required the minimum spark-advance. As expected, optimum spark-advance decreased with increasing load (see Figures 3, 4, 5, and 6). As can be seen in Figure 2, reduction of engine speed decreased the optimum spark advance, the cause of which is explained below with reference to Figure 7.

If the combustion time remained constant, the optimum spark-advance angle should be a direct function of speed and should be a

FIGURE 1

OPTIMUM SPARK ADVANCE VS. RELATIVE LOAD



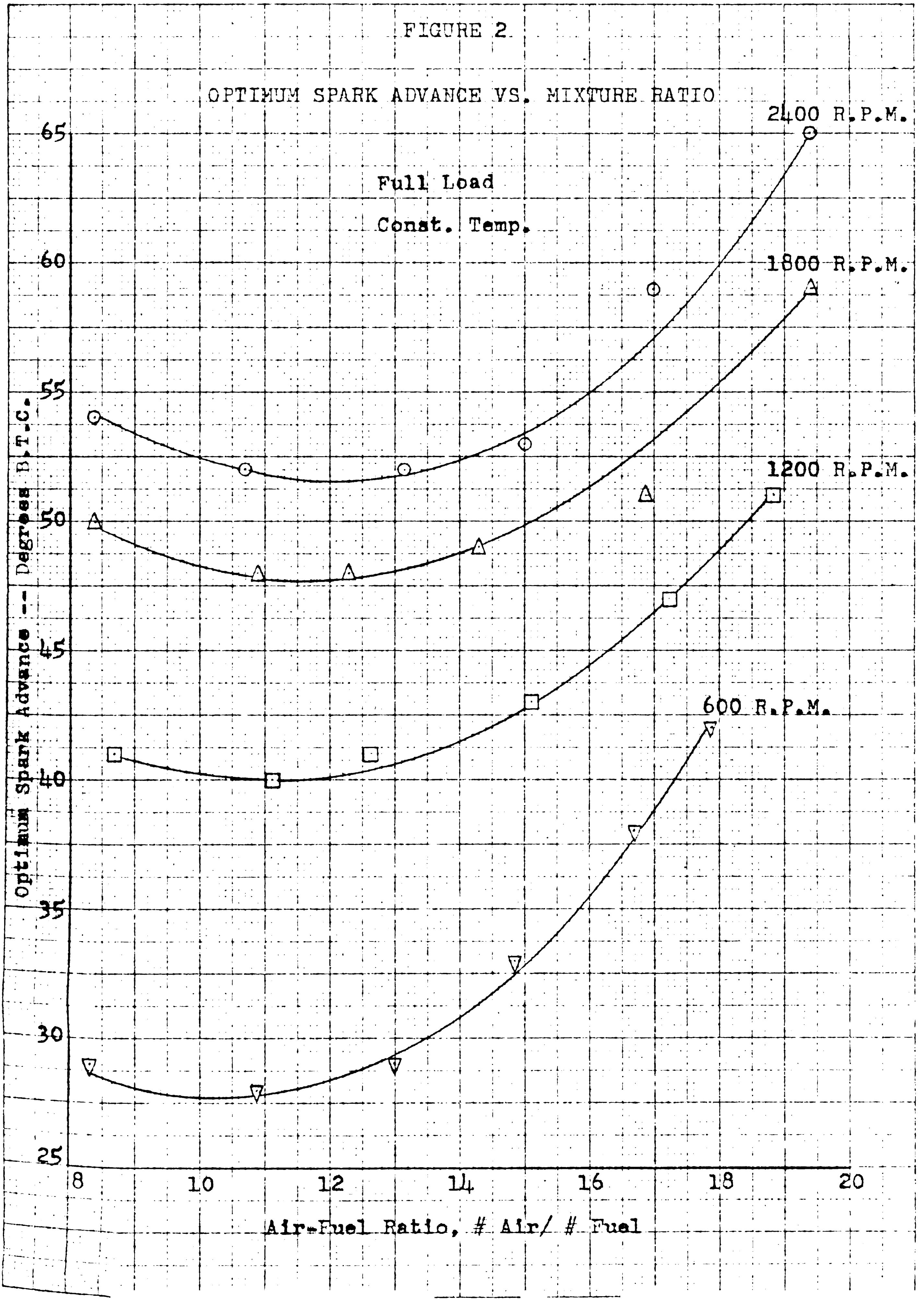
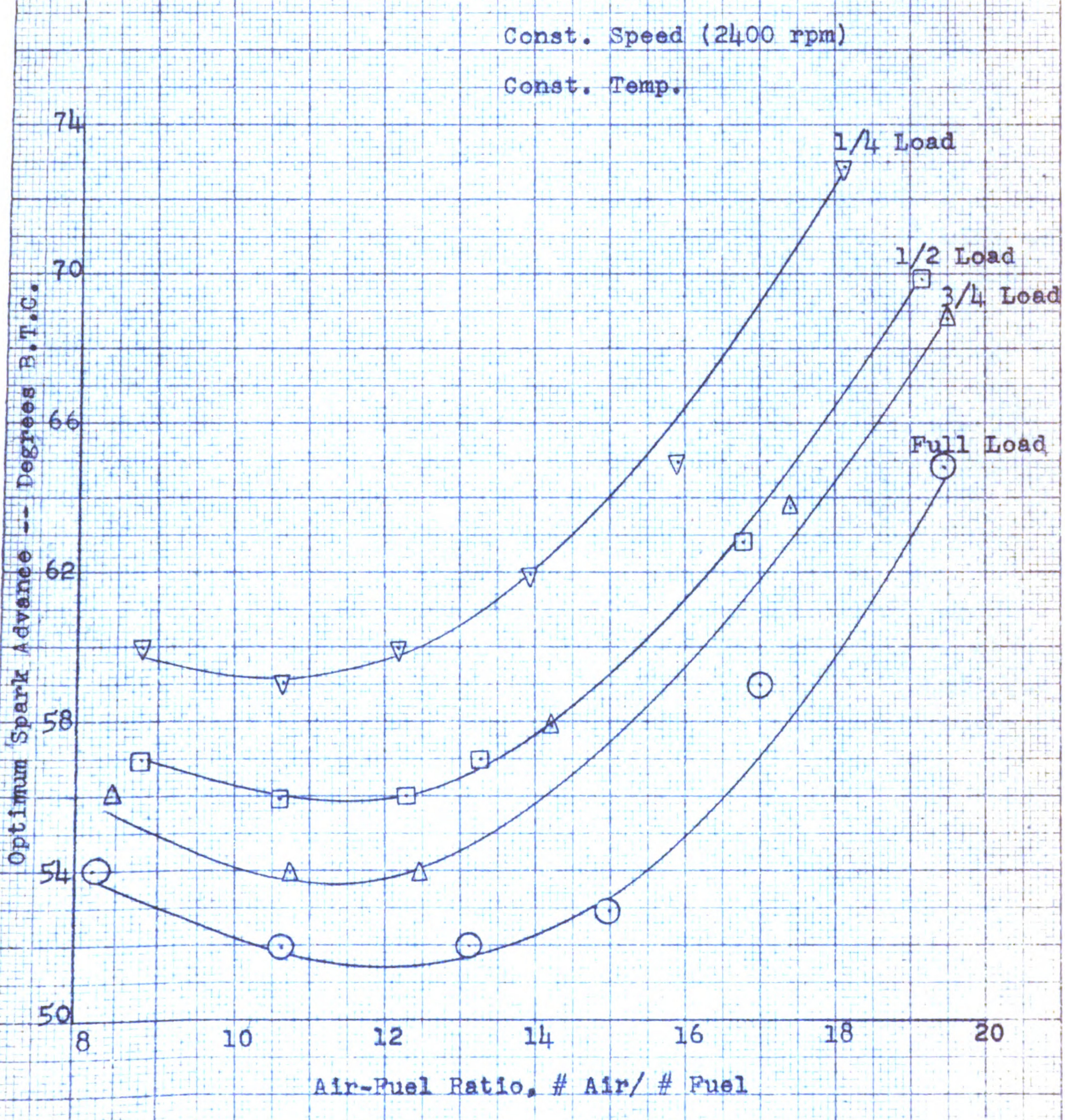


FIGURE 2
OPTIMUM SPARK ADVANCE VS. MIXTURE RATIO
Full Load
Const. Temp.
2400 R.P.M.
1800 R.P.M.
1200 R.P.M.
600 R.P.M.
Optimum Spark Advance -- Degrees B.T.C.
65
60
55
50
45
40
35
30
25
8
10
12
14
16
18
20
Air-Fuel Ratio, # Air/ # Fuel

FIGURE 3

OPTIMUM SPARK ADVANCE VS. MIXTURE RATIO



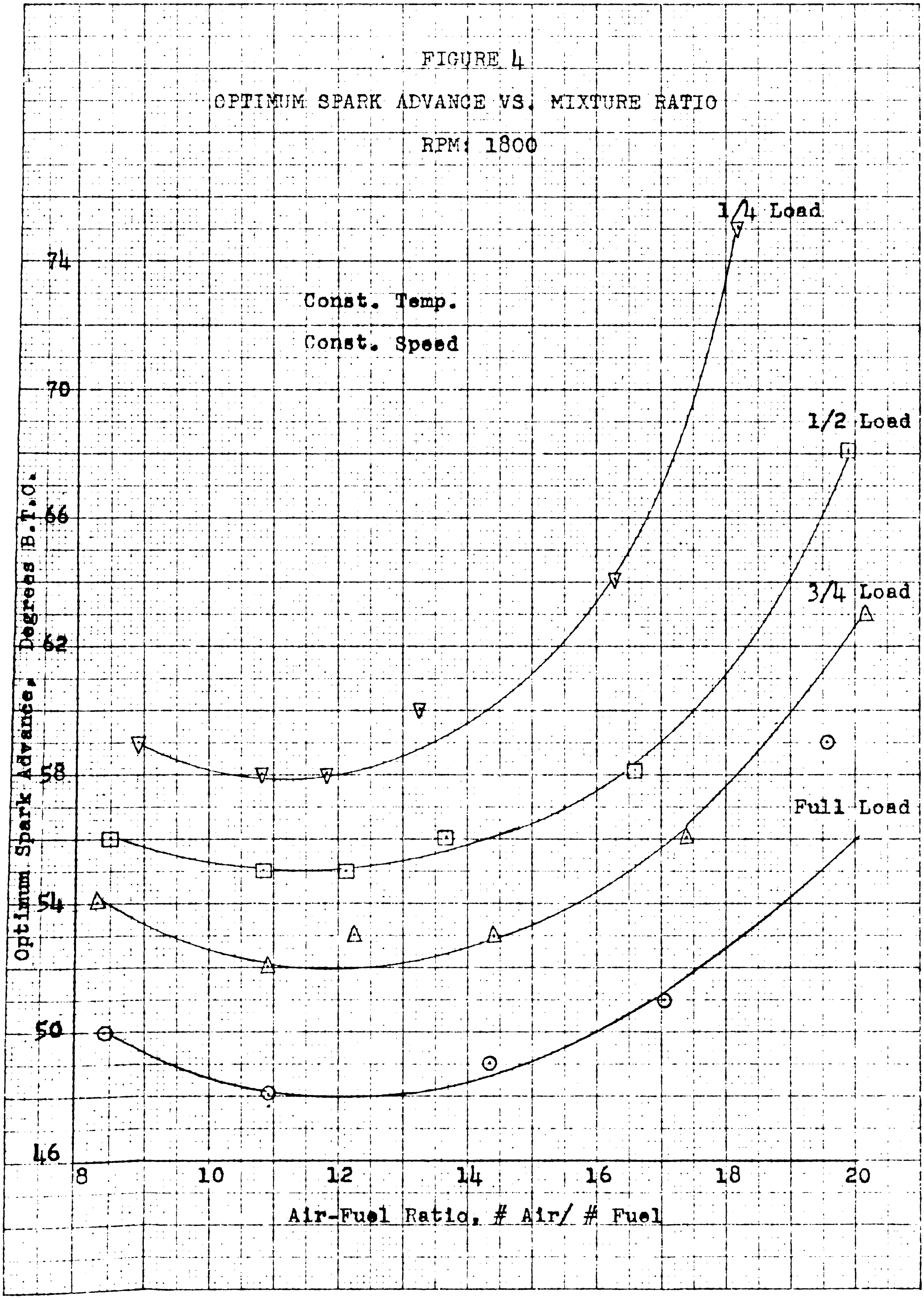
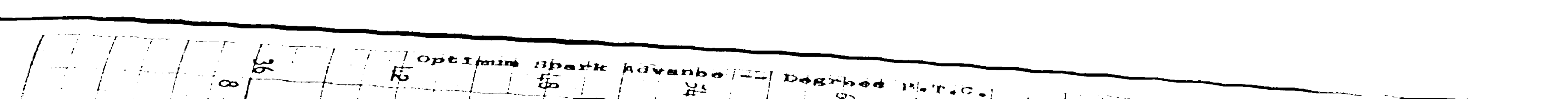


FIGURE 4
OPTIMUM SPARK ADVANCE VS. MIXTURE RATIO
RPM: 1800
Const. Temp.
Const. Speed
1/4 Load
1/2 Load
3/4 Load
Full Load
Optimum Spark Advance, Degrees B.T.C.
46
50
54
58
62
66
70
74
8
10
12
14
16
18
20
Air-Fuel Ratio, # Air/ # Fuel


Optimum Spark Advance -- Degrees B.T.C.
36

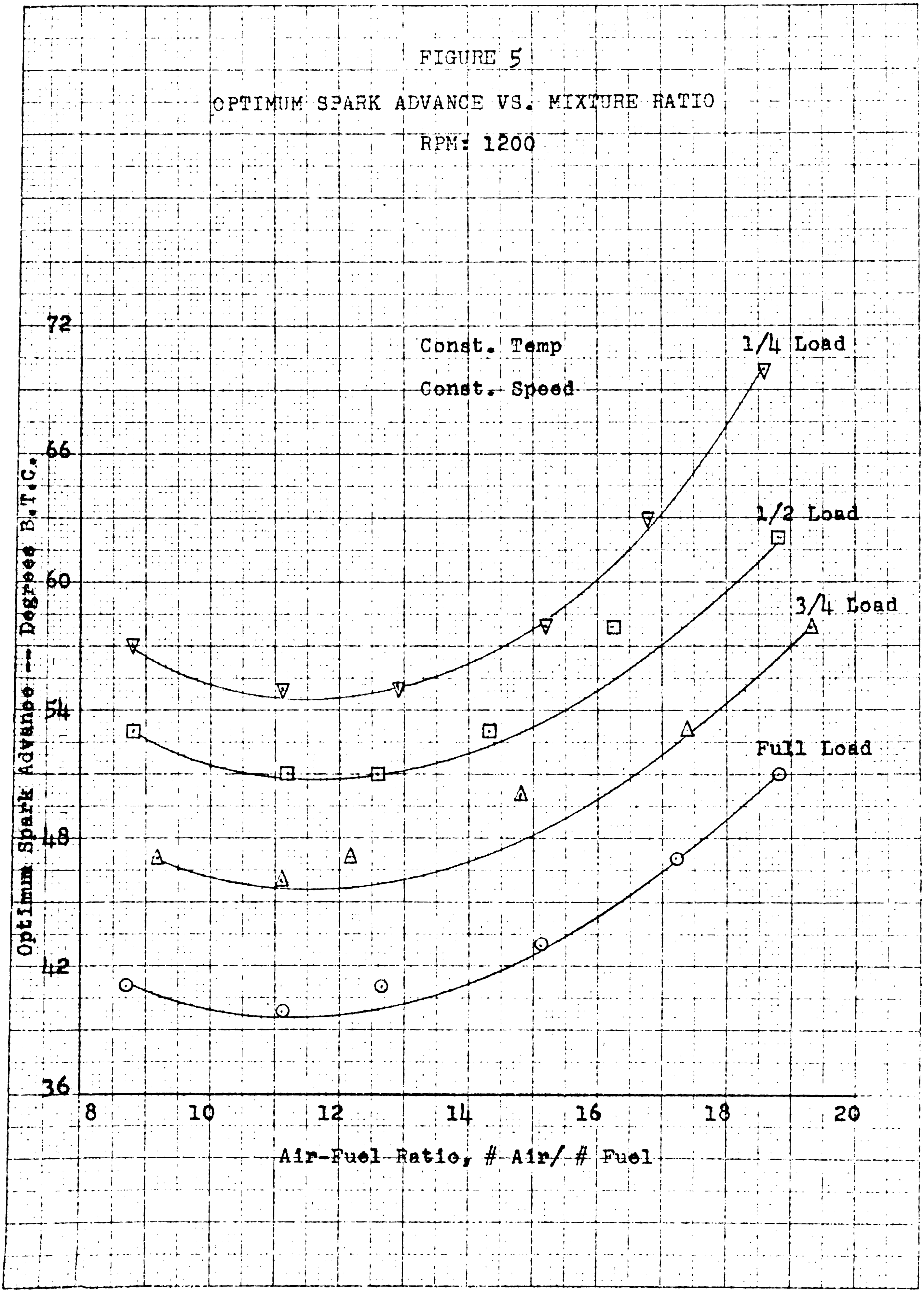
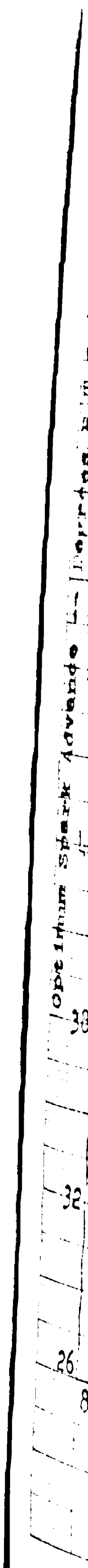


FIGURE 5
OPTIMUM SPARK ADVANCE VS. MIXTURE RATIO
RPM: 1200
Const. Temp
Const. Speed
1/4 Load
1/2 Load
3/4 Load
Full Load
Optimum Spark Advance -- Degrees B.T.C.
72
66
60
54
48
42
36
8
10
12
14
16
18
20
Air-Fuel Ratio, # Air/ # Fuel



Optimum Spark Advance
38
32
26

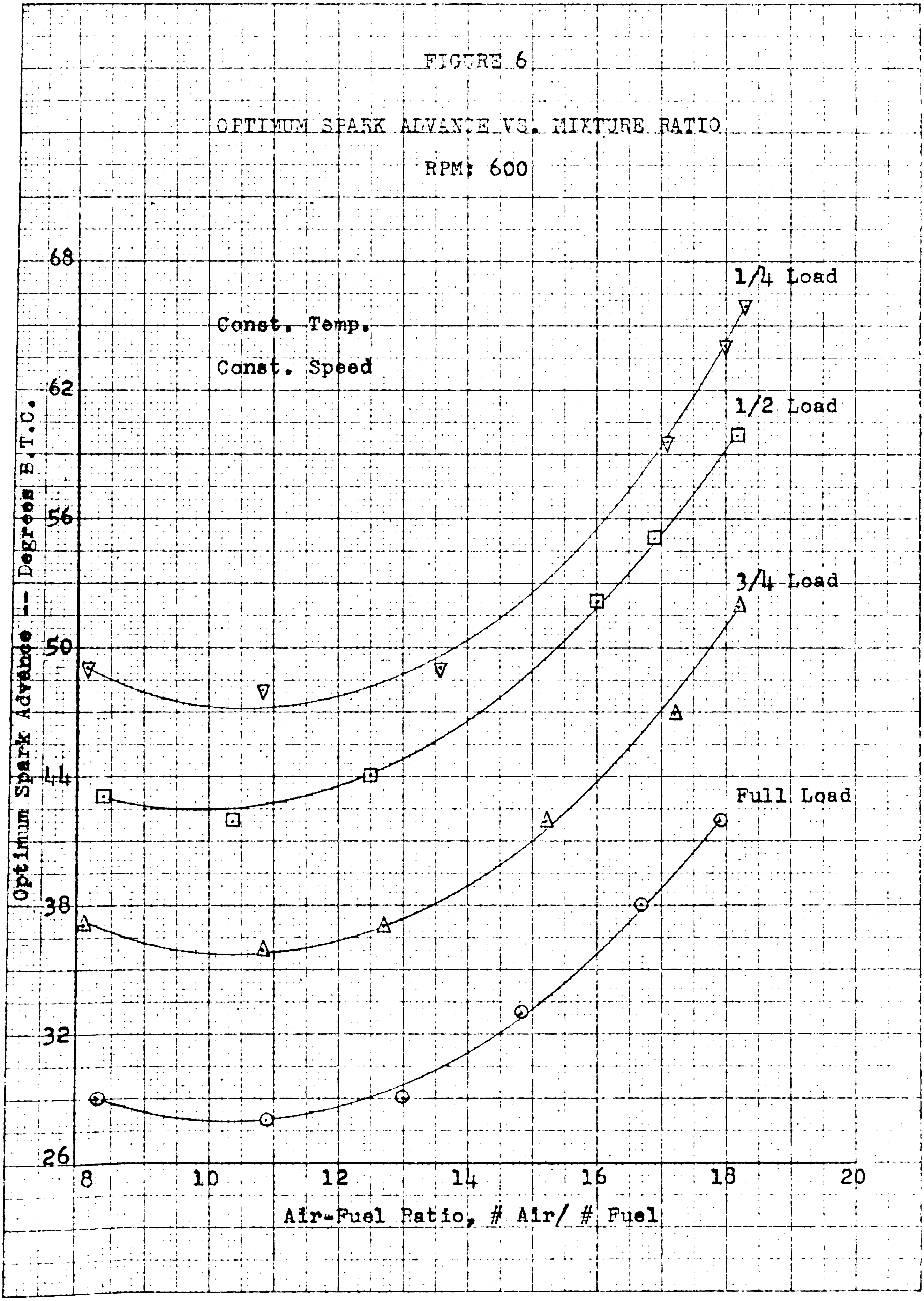


FIGURE 6
OPTIMUM SPARK ADVANCE VS. MIXTURE RATIO
RPM: 600
Const. Temp.
Const. Speed
1/4 Load
1/2 Load
3/4 Load
Full Load
Optimum Spark Advance -- Degrees B.T.C.
68
62
56
50
44
38
32
26
8
10
12
14
16
18
20
Air-Fuel Ratio, # Air/ # Fuel

straight line for optimum spark advance versus engine speed when the other factors were kept constant. In an actual case, constant combustion time does not exist. At higher engine speeds, the mixture enters the combustion chamber at a greater velocity producing turbulence in the combustion chamber which decreases the optimum spark-advance necessary. This effect will not be present at low speeds as the turbulence dies down before combustion occurs. This is well illustrated in Figure 7.

Carburetion Requirements. An engine can run over a wide range of mixture ratios. Somewhere within this range will be a mixture that produces maximum power and a mixture that produces maximum economy. For the operating conditions desired, investigation of the mixture-ratio requirements has to be made for the operating range of speed and relative load.

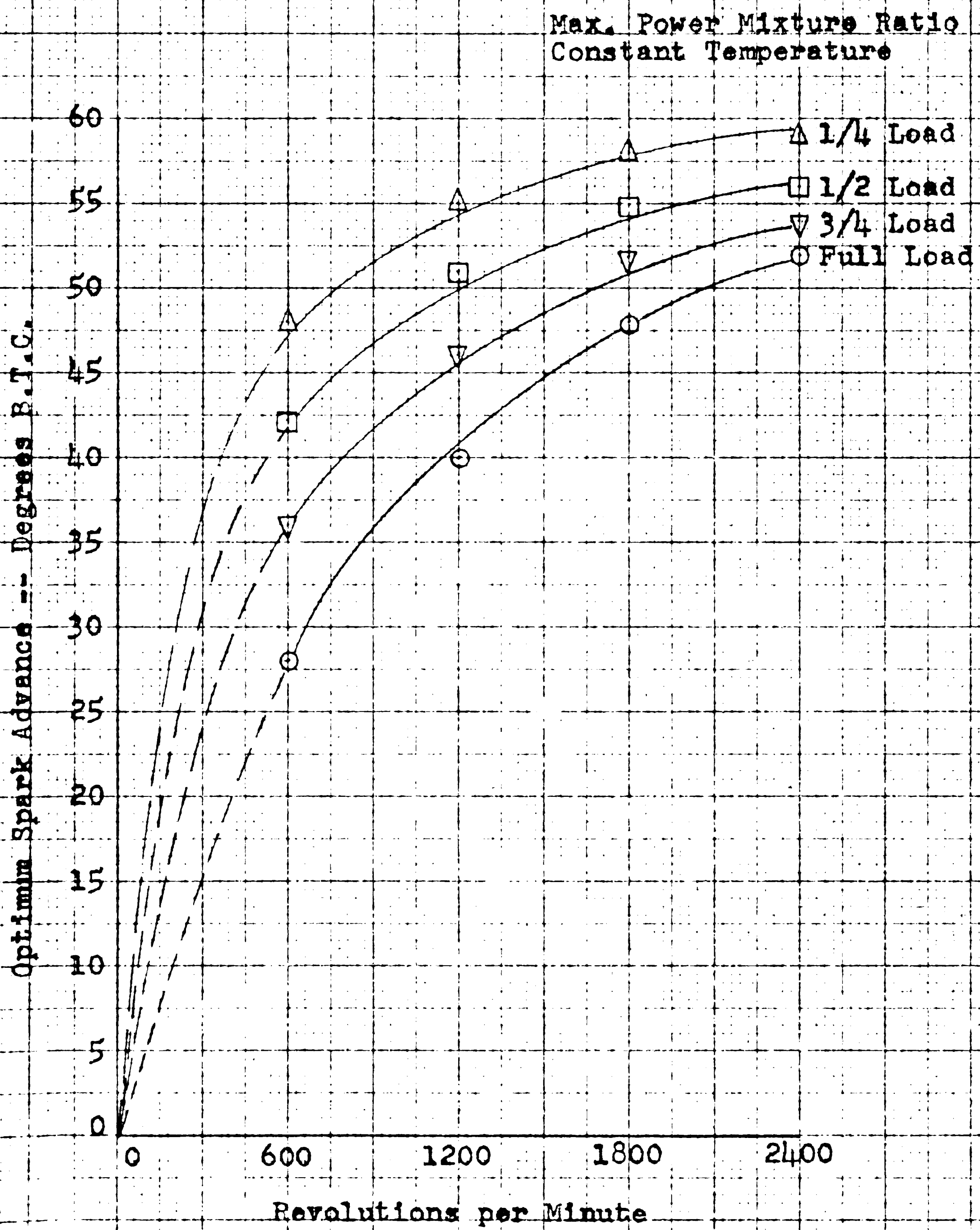
To find the effect of mixture ratio on power output (measured as brake mean-effective pressure) it is necessary to eliminate or hold constant the effect of other operating variables such as engine speed, intake vacuum and temperature, maintaining optimum spark-advance throughout.

Fuel, in rich mixtures containing excess fuel, displaces a small quantity of air, thus reducing the quantity of oxygen available for combustion. It also dilutes the mixture causing slower combustion and absorbs some of the heat of combustion decreasing the maximum temperature and pressure that can be attained. The final result of these effects is a slight reduction of power produced.

Lean mixtures similarly suffer from dilution effect and heat

Figure 7

OPTIMUM SPARK ADVANCE VS. R.P.M.



absorption because of the excess air contained. Heat absorption is more pronounced in this case. The heat that can be produced per pound of air present is considerably decreased due to the decrease of fuel available. The combined effect of all these factors caused a rapid drop of power as the mixture became leaner. A mixture, somewhere between the rich and lean limits, which is least affected by these losses and produces maximum power is the maximum power-mixture ratio. The resulting curves of brake mean-effective pressure versus air-fuel ratio are shown in Figures 8, 9, 10, 11 and 12.

The effect of mixture ratio on mean-effective power has been investigated by several authors (1, 2, 6, 7, 8, 10) over a wide range of operating conditions. It can be seen in Figures 9, 10, 11, and 12 that maximum mean-effective pressure occurred at the same air-fuel ratio regardless of throttle position. This mixture ratio gave maximum mean-effective pressure with variation in other operating conditions such as: revolutions per minute, inlet pressure, inlet temperature, etc. as long as cooling was adequate (8).

The criterion of engine operating economy is the amount of fuel used by the engine to produce a unit of output energy, or the brake specific fuel consumption. Rich mixture ratios have higher values of brake specific fuel consumption because of the unburned fuel that passes out of the exhaust. Lean mixtures contain excess air which dilutes the mixture and drops its heating value. The combination of these two effects decreases the power output. As the air-fuel ratio increases from rich mixture, the decrease in fuel flow is sufficiently greater than the drop in brake mean-effective pressure to cause a decrease in

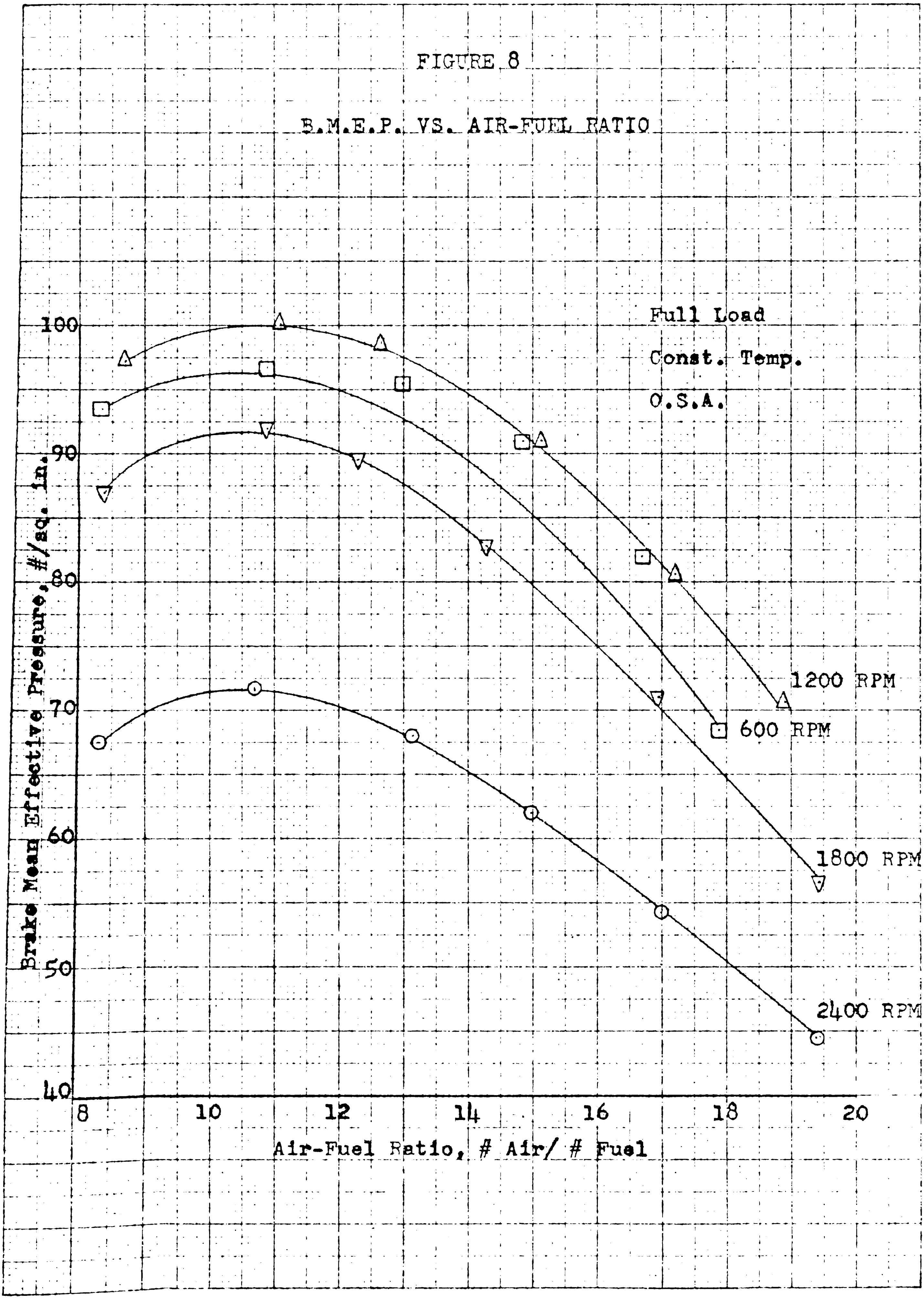
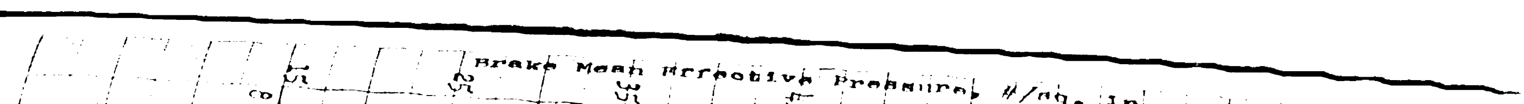


FIGURE 8
B.M.E.P. VS. AIR-FUEL RATIO
Full Load
Const. Temp.
O.S.A.
1200 RPM
600 RPM
1800 RPM
2400 RPM
Brake Mean Effective Pressure, #/sq. in.
100
90
80
70
60
50
40
8
10
12
14
16
18
20
Air-Fuel Ratio, # Air/ # Fuel


Brake Mean Effective Pressure, #/sq. in

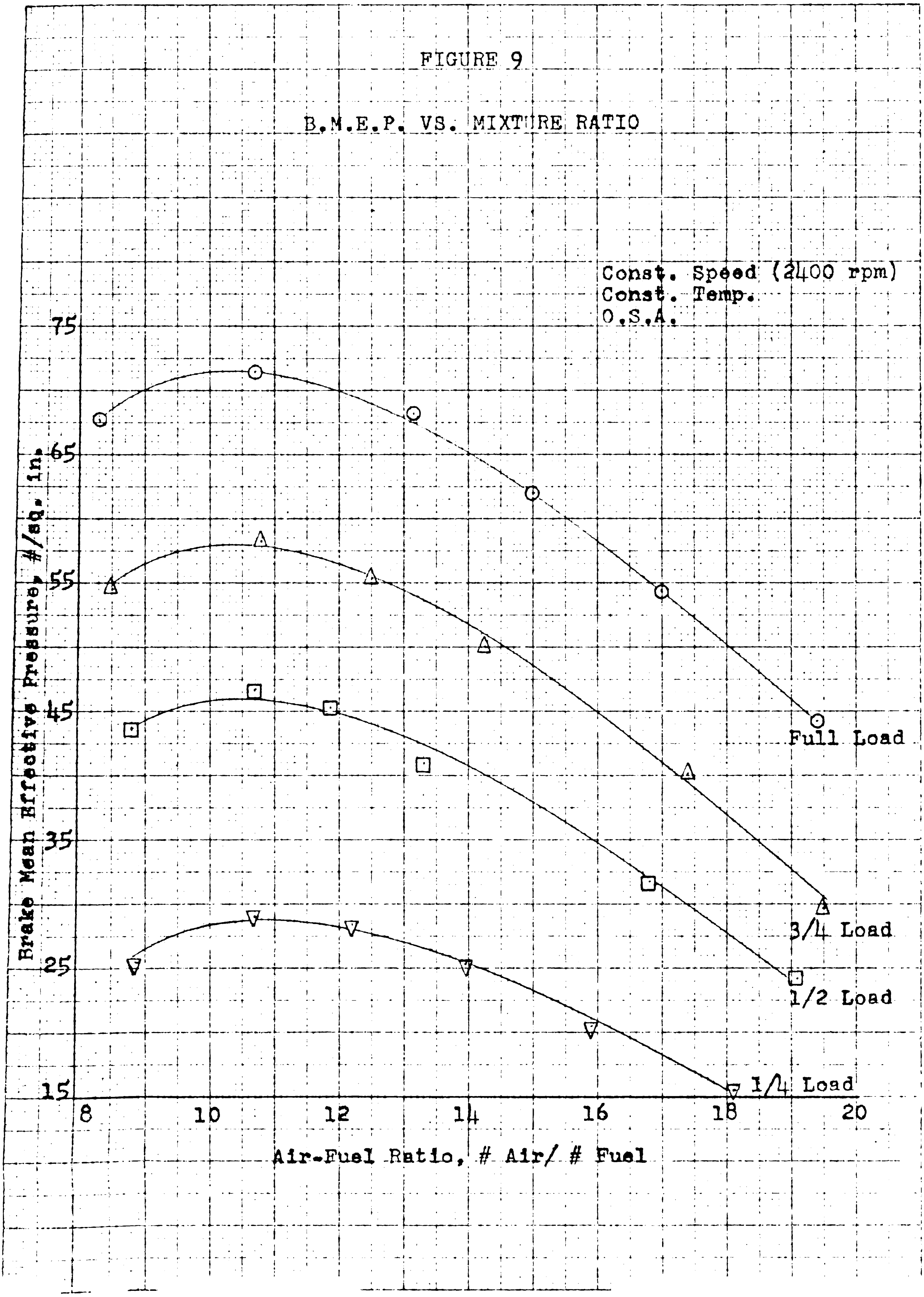


FIGURE 9
B.M.E.P. VS. MIXTURE RATIO
Const. Speed (2400 rpm)
Const. Temp.
O.S.A.
Brake Mean Effective Pressure, #/sq. in.
75
65
55
45
35
25
15
Full Load
3/4 Load
1/2 Load
1/4 Load
8
10
12
14
16
18
20
Air-Fuel Ratio, # Air/ # Fuel

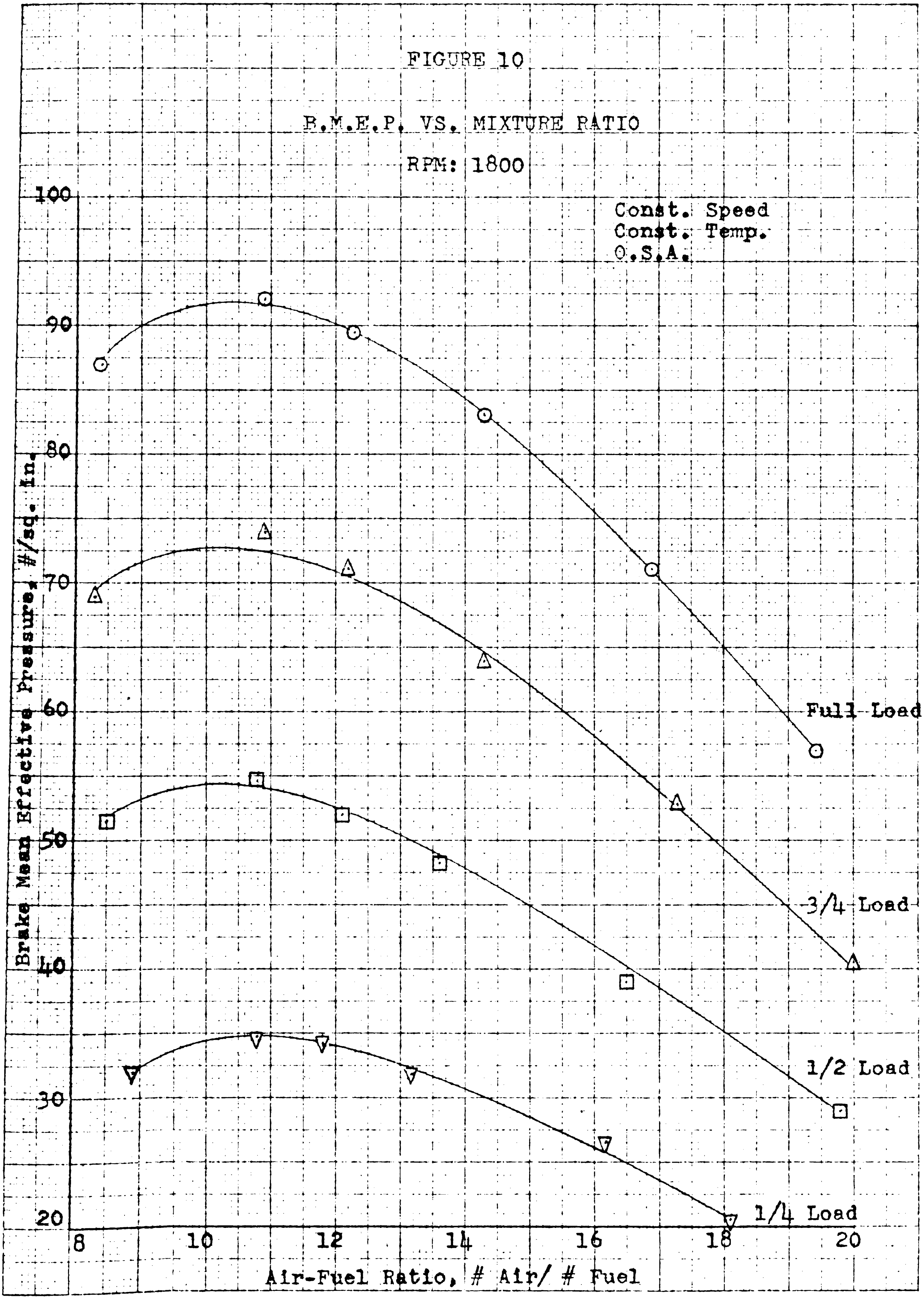
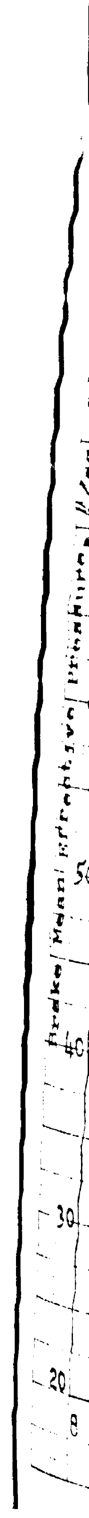


FIGURE 10
B.M.E.P. VS. MIXTURE RATIO
RPM: 1800
Const. Speed
Const. Temp.
O.S.A.
Brake Mean Effective Pressure, #/sq. in.
100
90
80
70
60
50
40
30
20
8
10
12
14
16
18
20
Full Load
3/4 Load
1/2 Load
1/4 Load
Air-Fuel Ratio, # Air/ # Fuel



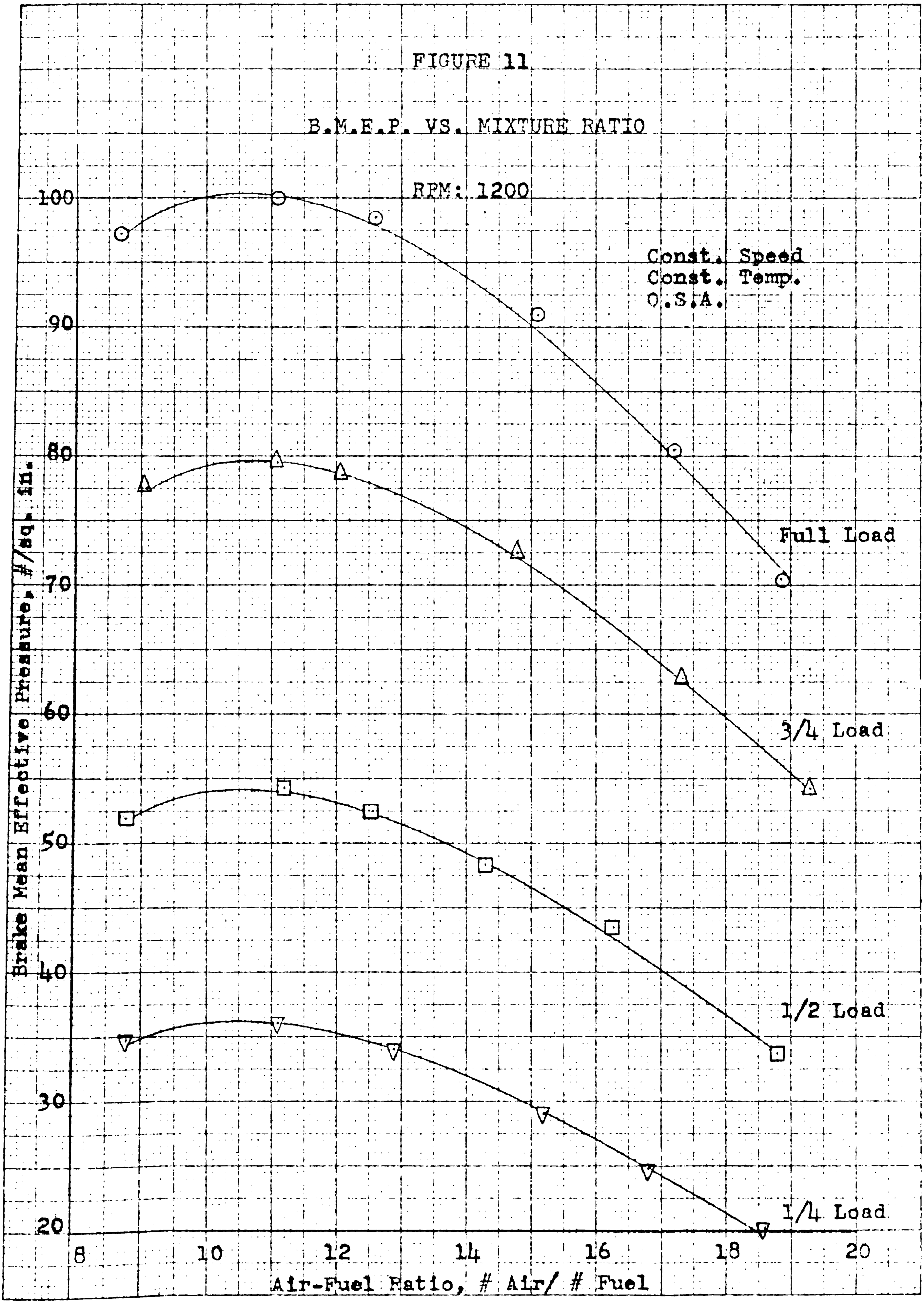


FIGURE 11
B.M.E.P. VS. MIXTURE RATIO
RPM: 1200
Const. Speed
Const. Temp.
O.S.A.
Full Load
3/4 Load
1/2 Load
1/4 Load
Brake Mean Effective Pressure, #/sq. in.
Air-Fuel Ratio, # Air/ # Fuel
100
90
80
70
60
50
40
30
20
8
10
12
14
16
18
20

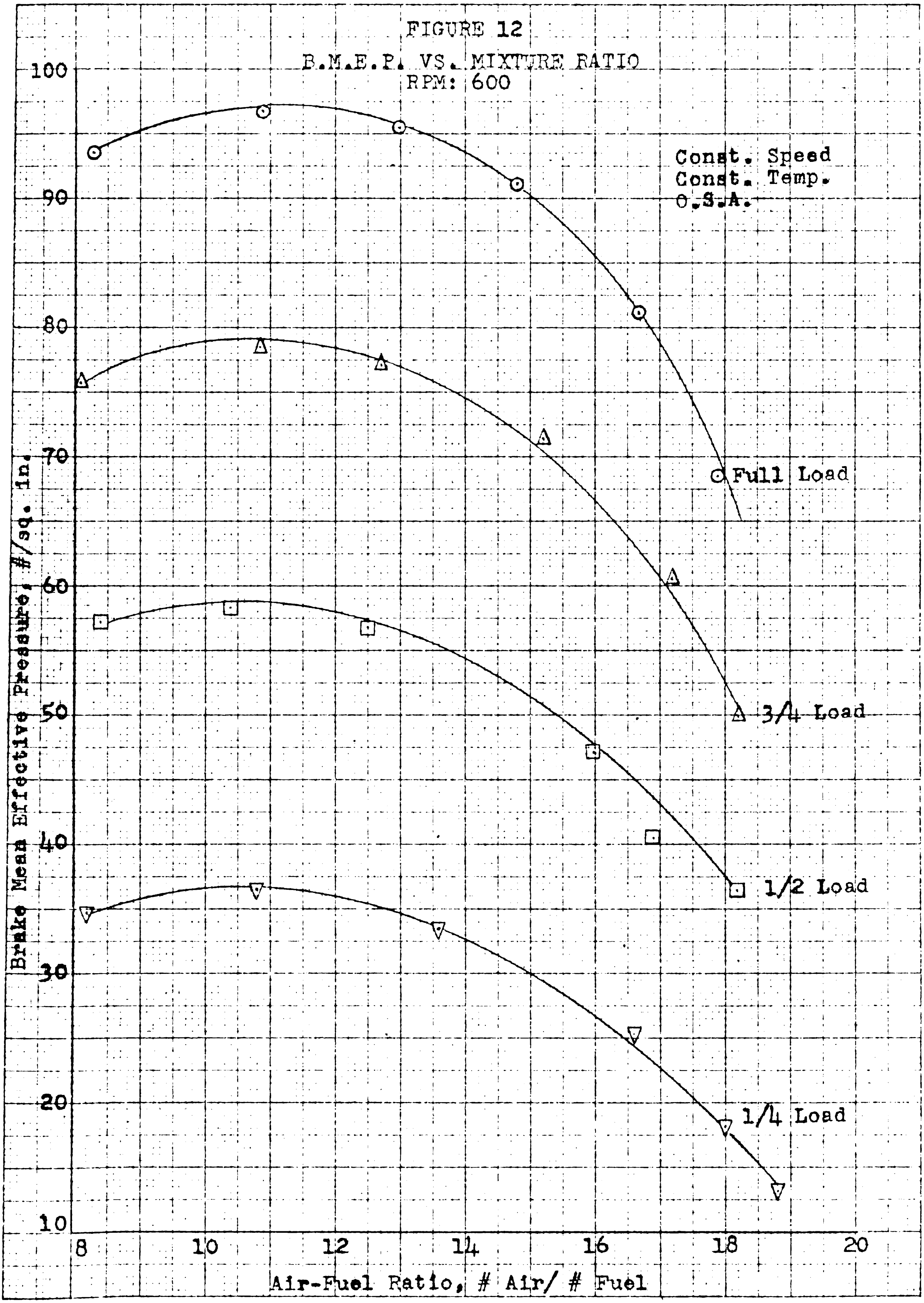


FIGURE 12
B.M.E.P. VS. MIXTURE RATIO
RPM: 600
Const. Speed
Const. Temp.
O.S.A.
Full Load
3/4 Load
1/2 Load
1/4 Load
Brake Mean Effective Pressure, #/sq. in.
Air-Fuel Ratio, # Air/ # Fuel
100
90
80
70
60
50
40
30
20
10
8
10
12
14
16
18
20

brake specific fuel consumption. As the power drops more and more rapidly in the lean mixture range, an increment of mixture-ratio is soon reached when there is not much change in brake specific fuel consumption. For leaner mixtures than this the power decreases more rapidly than the fuel flow as a result of which, brake specific fuel consumption increases. That particular mixture-ratio which produces the lowest value of brake specific fuel consumption is known as the maximum-economy mixture ratio for the set of conditions used. Figures 13, 14, 15, 16 and 17 illustrate the above effects of mixture-ratio on brake specific fuel consumption under the specified conditions as shown. The useful range of air-fuel ratios became narrower as the load was reduced. These effects were due to the fact that the frictional mean effective pressure changed much more slowly with the indicated mean effective pressure. This effect can be easily visualized by imagining the engine to be running at no load at the best power mixture. A small change in air-fuel ratio would reduce the indicated output below the frictional requirement and the engine would stop. Obviously at this point the best-power mixture ratio is also the best-economy mixture ratio and the useful mixture range is then zero (9). Since a change in the relative amount of clearance gas present in the charge has a decided effect on power production, a higher relative load or a lower clearance-gas dilution would permit a leaner mixture at which maximum economy conditions occur. As shown in Figures 14, 15, 16, and 17, relative load has another effect on brake specific fuel consumption. Pumping losses in an engine increase as the relative load decreases. When operating at higher relative loads, a certain amount of fuel is used to overcome friction (which is

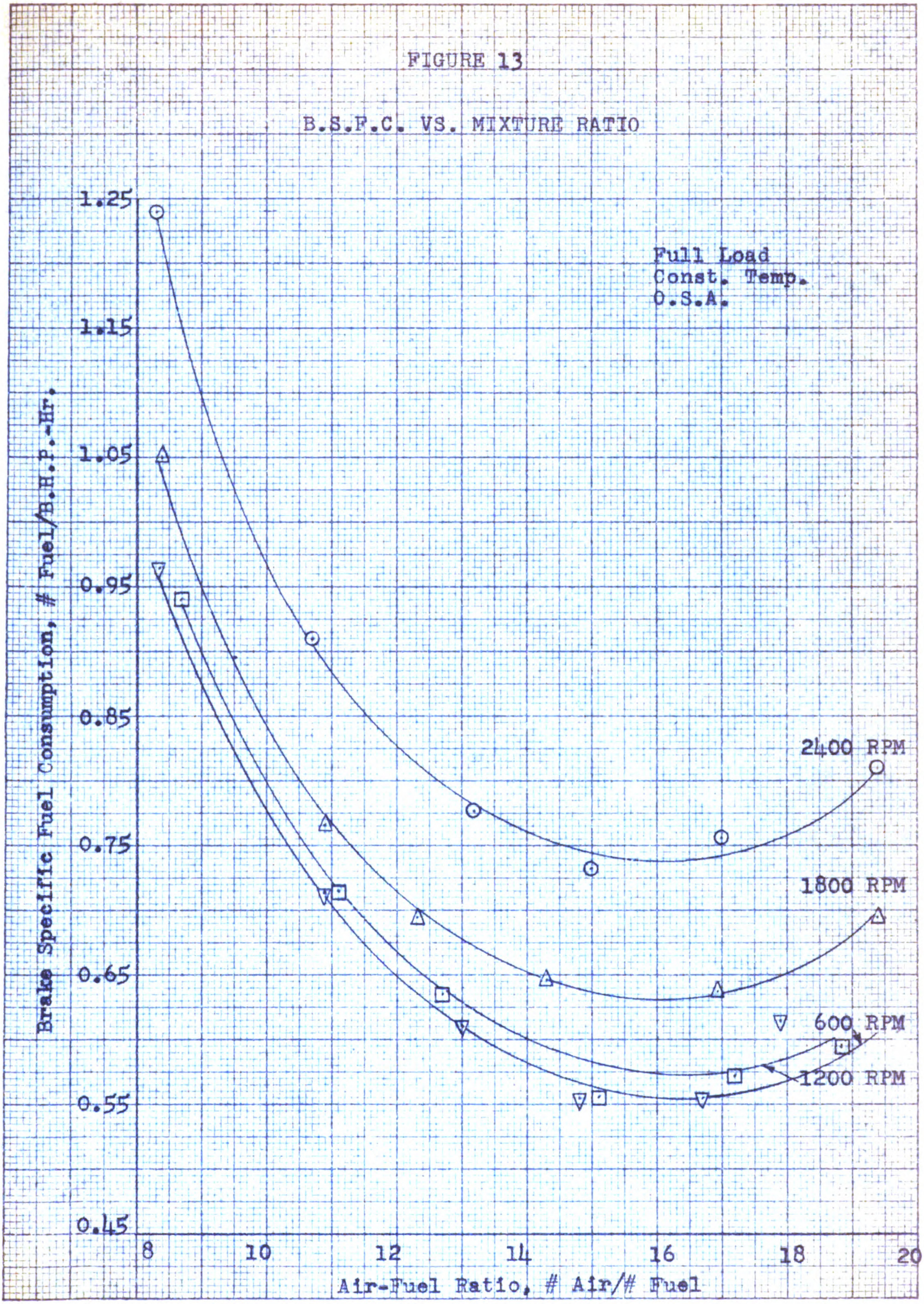


FIGURE 13
B.S.F.C. VS. MIXTURE RATIO
Full Load
Const. Temp.
O.S.A.
2400 RPM
1800 RPM
600 RPM
1200 RPM
Brake Specific Fuel Consumption, # Fuel/B.H.P.-Hr.
1.25
1.15
1.05
0.95
0.85
0.75
0.65
0.55
0.45
8
10
12
14
16
18
20
Air-Fuel Ratio, # Air/# Fuel

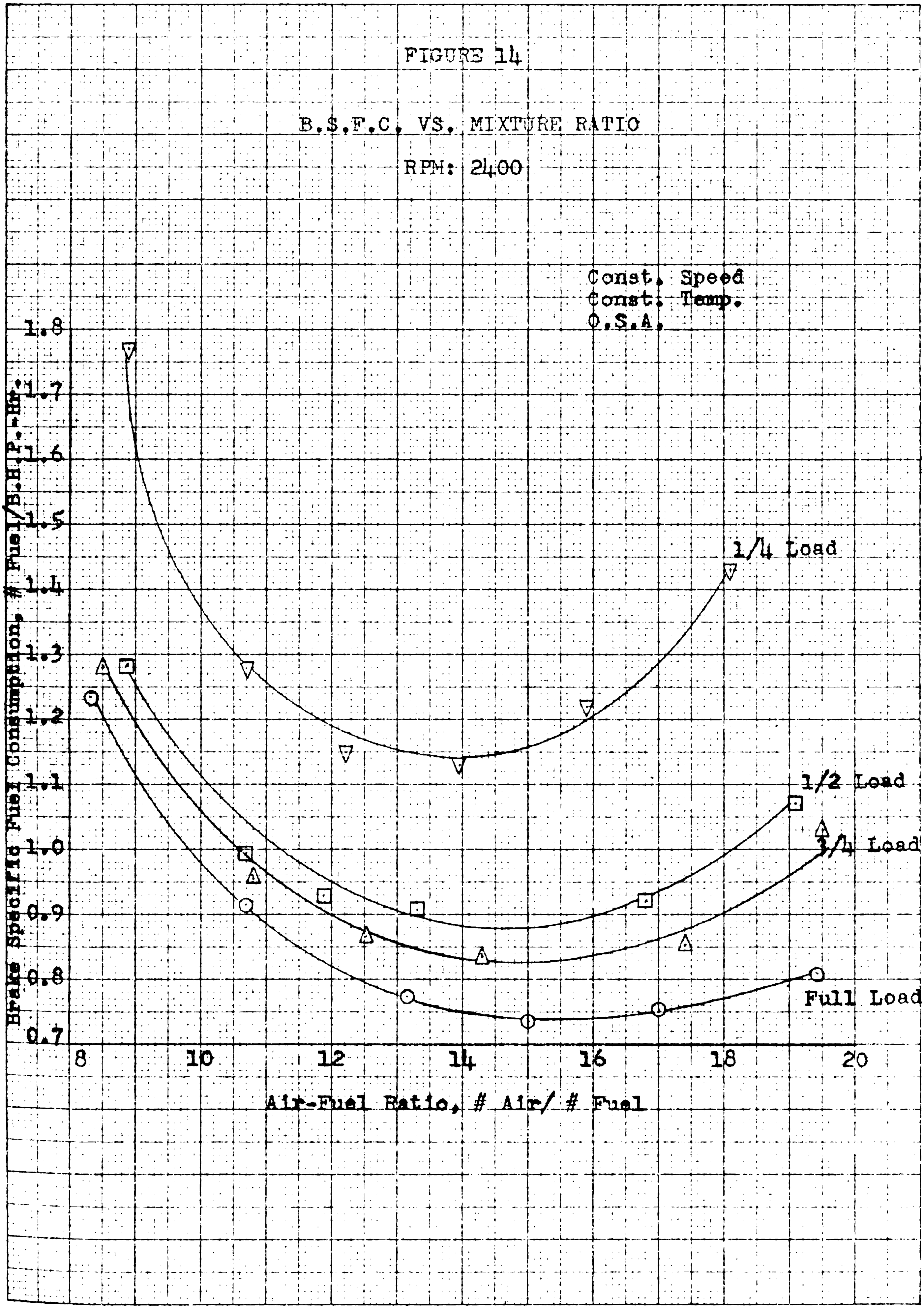


FIGURE 14
B.S.F.C. VS. MIXTURE RATIO
RPM: 2400
Const. Speed
Const. Temp.
O.S.A.
Brake Specific Fuel Consumption, # Fuel/B.H.P.-Hr.
1.8
1.7
1.6
1.5
1.4
1.3
1.2
1.1
1.0
0.9
0.8
0.7
1/4 Load
1/2 Load
3/4 Load
Full Load
8
10
12
14
16
18
20
Air-Fuel Ratio, # Air/ # Fuel

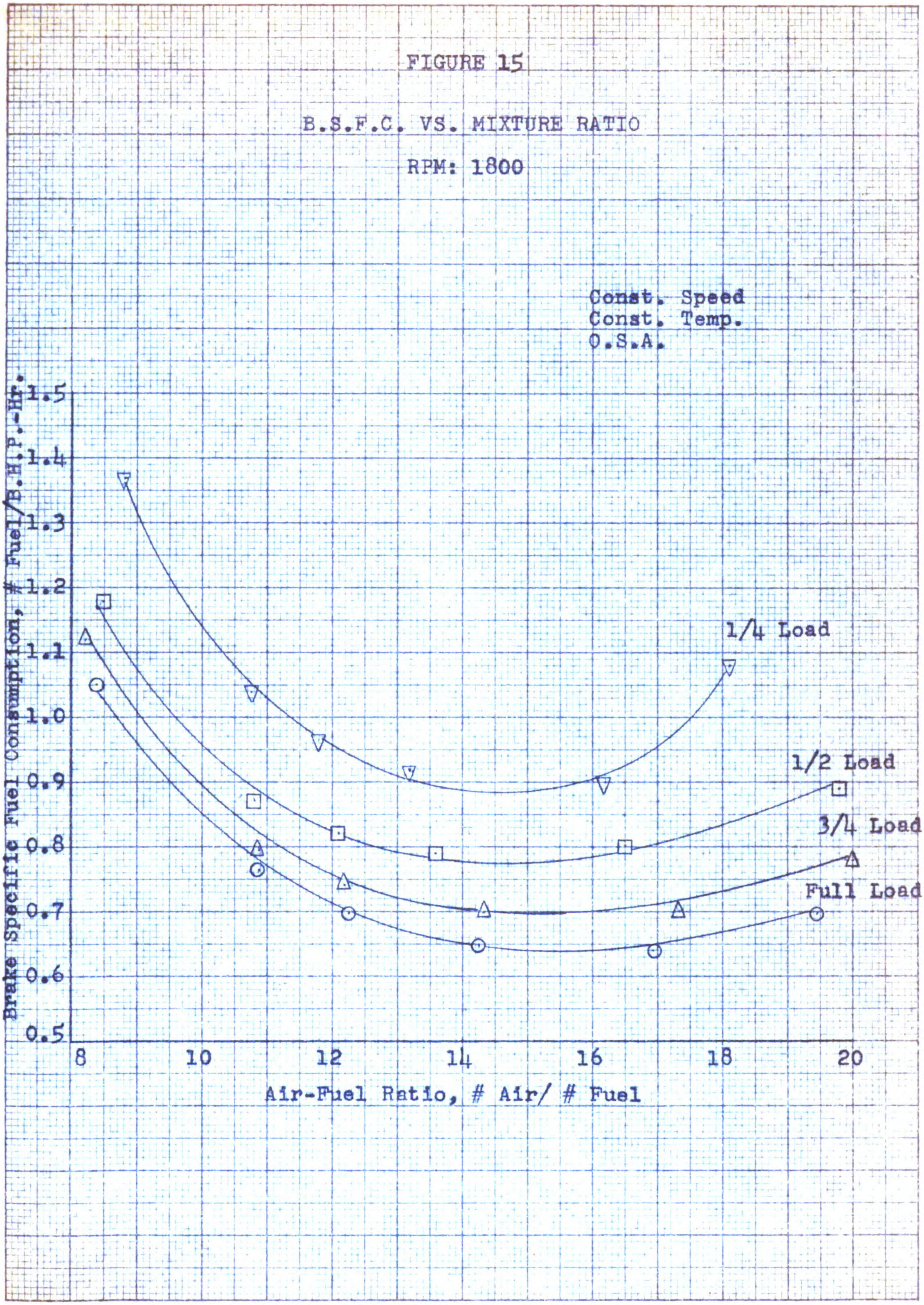
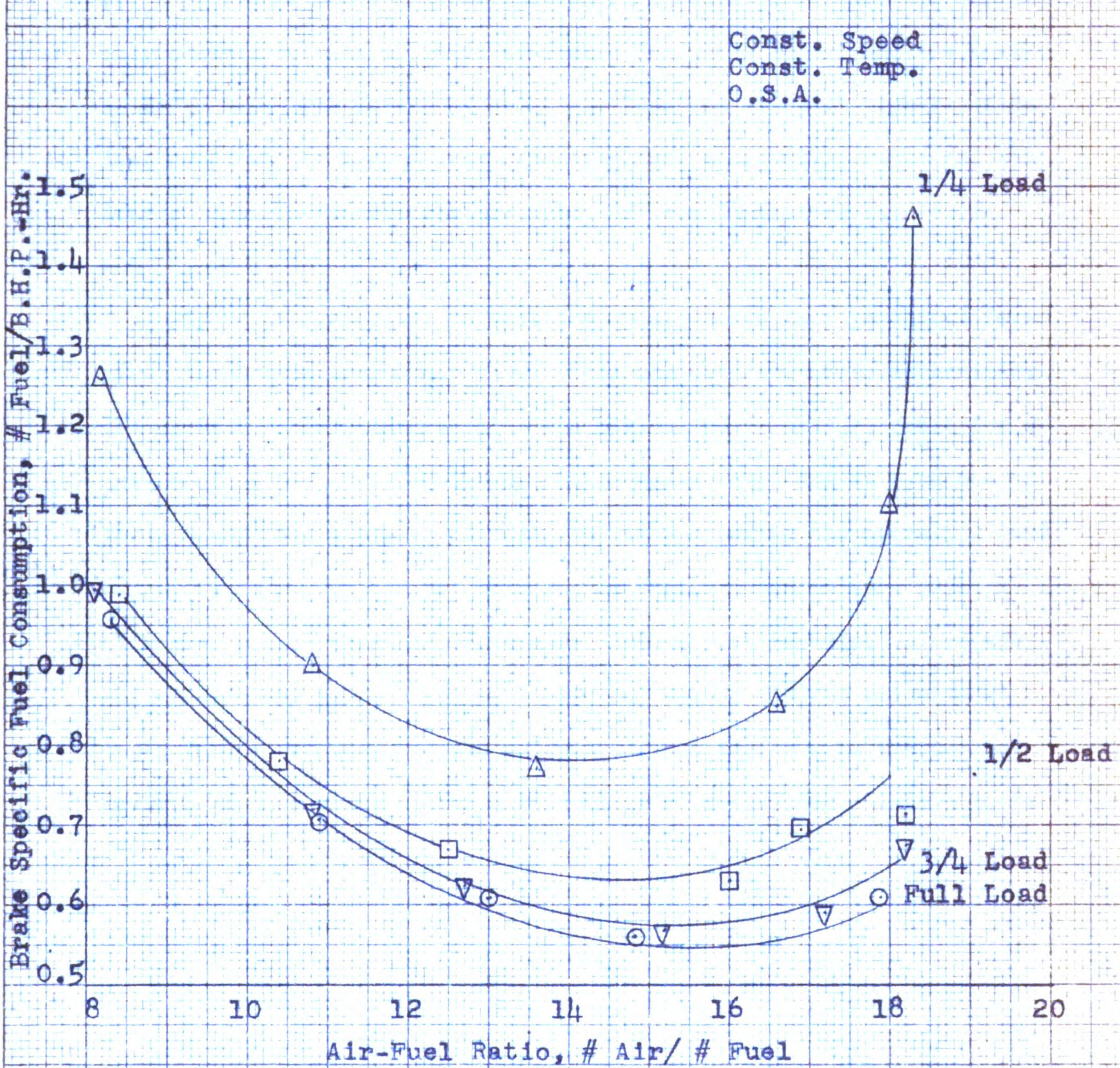


FIGURE 15
B.S.F.C. VS. MIXTURE RATIO
RPM: 1800
Const. Speed
Const. Temp.
O.S.A.
1/4 Load
1/2 Load
3/4 Load
Full Load
Brake Specific Fuel Consumption, # Fuel/B.H.P.-Hr.
1.5
1.4
1.3
1.2
1.1
1.0
0.9
0.8
0.7
0.6
0.5
8
10
12
14
16
18
20
Air-Fuel Ratio, # Air/ # Fuel

FIGURE 17

B.S.F.C. VS. MIXTURE RATIO

RPM: 600

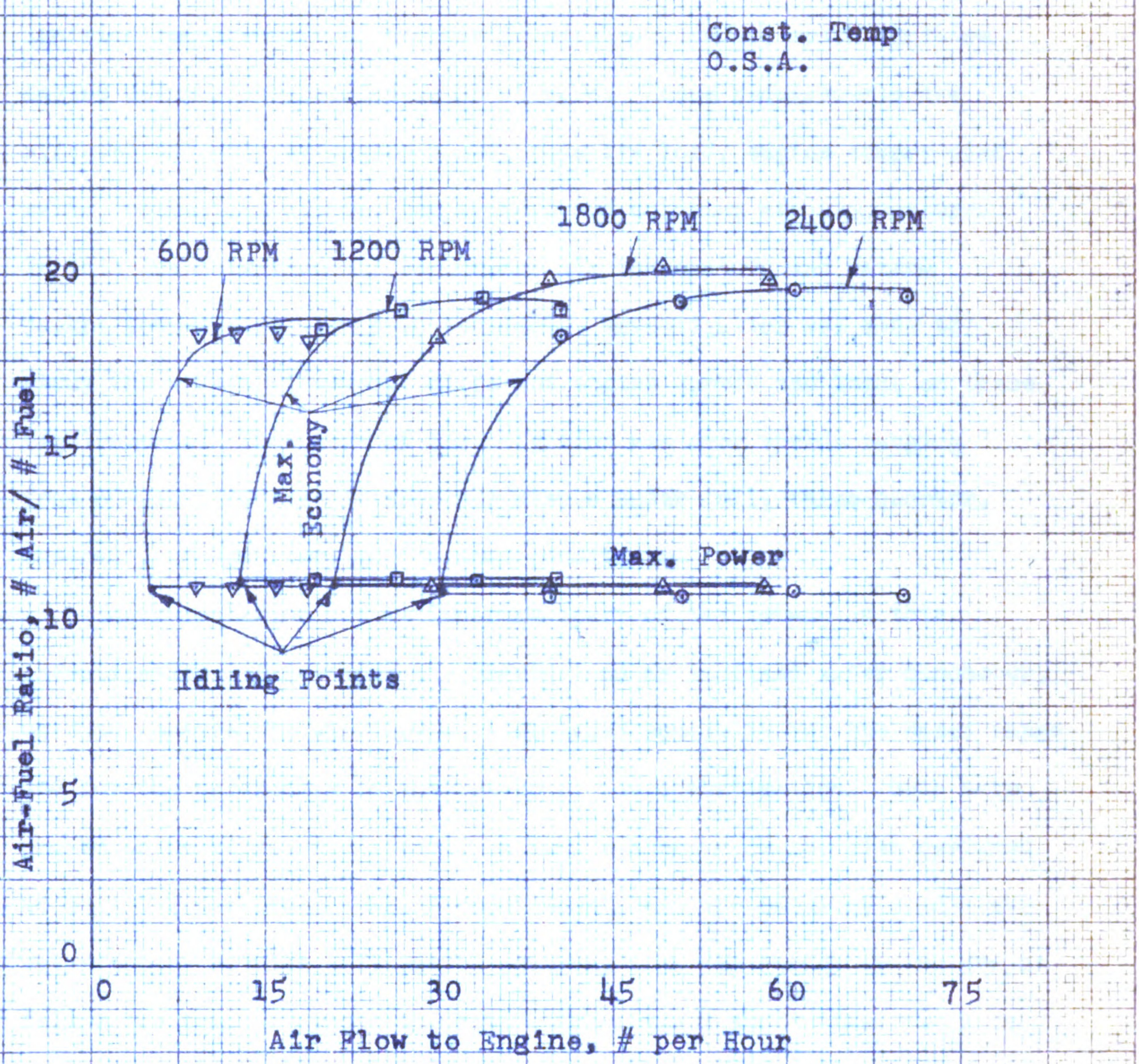


almost constant at constant speed and variable load) leaving most of the fuel to produce brake power output. When operating at the same speed at low relative loads, the amount of fuel used to overcome friction is the same and since less total fuel is taken into the engine, only a small portion of it is available for brake output and the brake specific fuel consumption is greater. Thus operation at high relative load produced the lowest brake-specific fuel consumption value and the leanest maximum-economy mixture ratio as seen in Figures 14, 15, 16, and 17.

Figure 18 summarizes carburetion requirements in a graphical form as mixture-ratio versus air flow. This was obtained after determining the maximum-eonomy mixture ratio and maximum-power mixture ratio for a series of speeds covering the operating speed range and at each one of these speeds for a series of relative loads covering the operating load range. Idle points did not coincide as engine's losses were greater at higher speeds which required more mixture just to idle. Since the maximum-economy mixture curves at several speeds did not coincide at one point, a carburetor could not provide the engine with the correct maximum economy mixture over a wide range of speeds and relative loads.

FIGURE 18

MIXTURE RATIO VS. AIR FLOW



SUMMARY

An experimental investigation was made of the carburetion and ignition requirements of a Christie single-cylinder, spark-ignition engine using a compression ratio of 6.08. These requirements are affected directly by engine speed, relative load, engine compression ratio, combustion chamber temperature and entering air temperature. In this investigation the carburetor air temperature and water-jacket temperature were held constant. Investigation was made to study the effect of several different engine speeds and loads on these requirements. The range of investigation was over speeds ranging from 600 to 2,400 revolutions per minute under full, three-quarter, one-half, one-quarter, and no load conditions.

Instrumentation was an important part of the set-up in this investigation. A U-shaped air chamber and an inclined water manometer, used to measure the air flow were designed specially for the speed range and for the particular engine set-up of the investigation. Both have proved to be very satisfactory. A fuel measuring device also designed to suit the particular engine requirements and the range of study gave satisfactory results. It can be observed that the curves in the graphs follow the trend of those theoretically expected quite closely. This was due largely to the accuracy of the air and fuel flow measuring equipment.

The experimental results obtained in the investigation can be summarized briefly as follows:

1. Ignition requirements:

   a) Reduction of engine speed decreased the optimum spark-advance (Figures 2 and 7).

   b) Optimum spark-advance decreased with increasing load as shown in Figures 3, 4, 5, and 6.

   c) For maximum power-mixture ratio, the material presented in Figures 2, 3, 4, 5, and 6 can be combined into a single ignition requirement curve, as shown in Figure 1. Optimum spark-advance thus decreased with an increase in relative load, and less rapidly with increasing speed.

2. Carburetion requirements:

   a) Maximum mean-effective pressure occurred at the same air-fuel ratio regardless of throttle position (Figures 9, 10, 11, and 12).

   b) The effect of mixture ratio on brake-specific fuel consumption under specified conditions are illustrated in Figures 13, 14, 15, 16, and 17. It can be noticed from Figures 14, 15, 16, and 17 that the useful range of air-fuel ratio became narrower as the load was reduced. Operation at high relative load produced the lowest brake-specific fuel consumption value and the leanest maximum-economy mixture ratio.

   c) Figure 18 vividly summarizes the carburetion requirements of the engine. The maximum-economy and maximum-power mixture ratios intersected at idling points, and thus represent the intersections of the two mixture conditions for each constant speed.

The experimental results, in general, are in reasonable agreement with the results obtained by previous investigators (1, 2, 6, 7, 8, and 10) over a wide range of speeds and loads.

APPENDIX A

Test Equipment. The engine tested was a Christie, single-cylinder, overhead valve, four-stroke cycle, water-cooled, spark-ignition engine. A 3.066-inch bore and 4.5-inch stroke gave a total displacement of 33.22 cubic inchaes. The engine compression ratio could be varied by changing the position of the cylinder by means of gearing and provided as part of the engine equipment. A dial was provided for calibrating the compression ratio of the engine. The calibration procedure is described in Appendix C. A compression ratio of 6.08 was used in this investigation. The engine was equipped with a 6-volt ignition system and a manually controlled spark-advance arrangement. Manually-controlled spark-advance mechanism also served as an indicator of the spark-advance in degrees by means of a spark when the engine was running. The spark plug was cleaned and correct gap was set.

Engine cooling was accomplished by a water-vapor condenser as shown in Figure 19. The circuit was so arranged that the expansion of the water by the heat absorbed in the jacket and its consequent tendency to rise, caused it to circulate through a water condenser. Water circulation in the condenser took place from water tap into the condenser and from there another tubing led to drainage. The simple down-draft, float-type carburetor was modified to control the fuel flow as required in the experimental investigation.

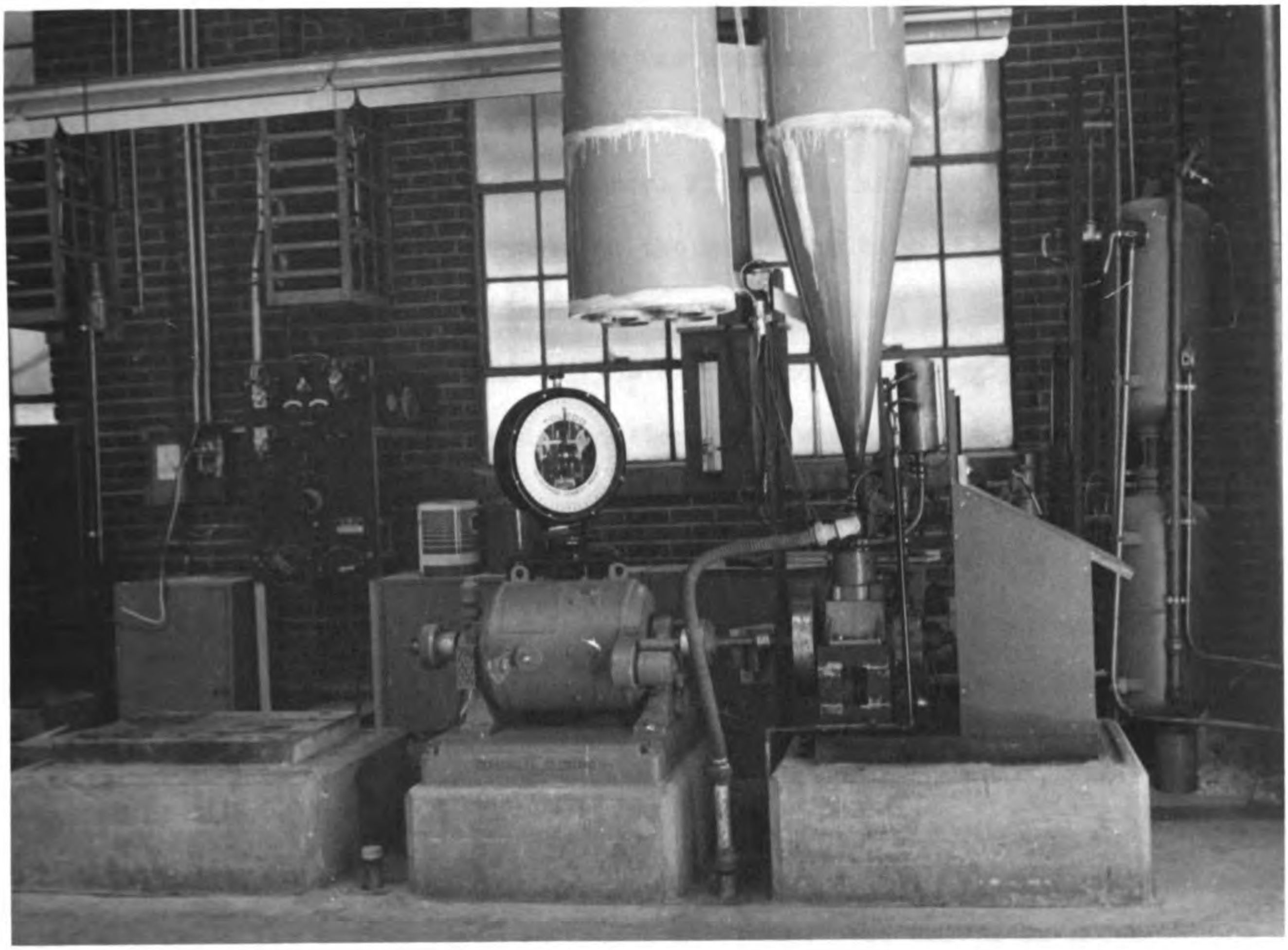


FIGURE 19

APPENDIX B

Instrumentation. Intake air flow was measured by means of a specially designed equipment as illustrated in Figures 20 and 21. This consisted of a U-shaped air chamber, Figure 20, air-flow inclined water manometer and scales attached to the manometer base, Figure 21. The air chamber admitted air through one of the orifices as required during the test. For each size orifice used, a correspondingly calibrated scale was attached to the manometer board which was calibrated directly to read the flow of air in pounds per hour. Appendix D gives the method adopted to determine the number and sizes of the orifices used in the air chamber. The air chamber hanging from above was balanced around a pulley by cast steel weights. This was intended to take up different heights with ease whenever the compression ratio was changed by moving the cylinder up or down.

Figure 22 shows the fuel-flow meter. On the scale attached to it, the fuel-flow was measured in pounds per hour. Two valves as shown in Figure 23 were joined to a common tubing that led to the carburetor flow chamber. Either one or both were kept in communication according to the fuel-flow requirements. The fuel was supplied from a fuel can by means of a fuel pump mounted on the back side of the flow meter, through the fuel-flow meter to the float chamber. The amount of fuel to be supplied by the pump to the cylinder was controlled as mentioned earlier by adjusting the relative position of a needle valve in the nozzle of the carburetor so as to change the flow area. Details of the

FIGURE 20

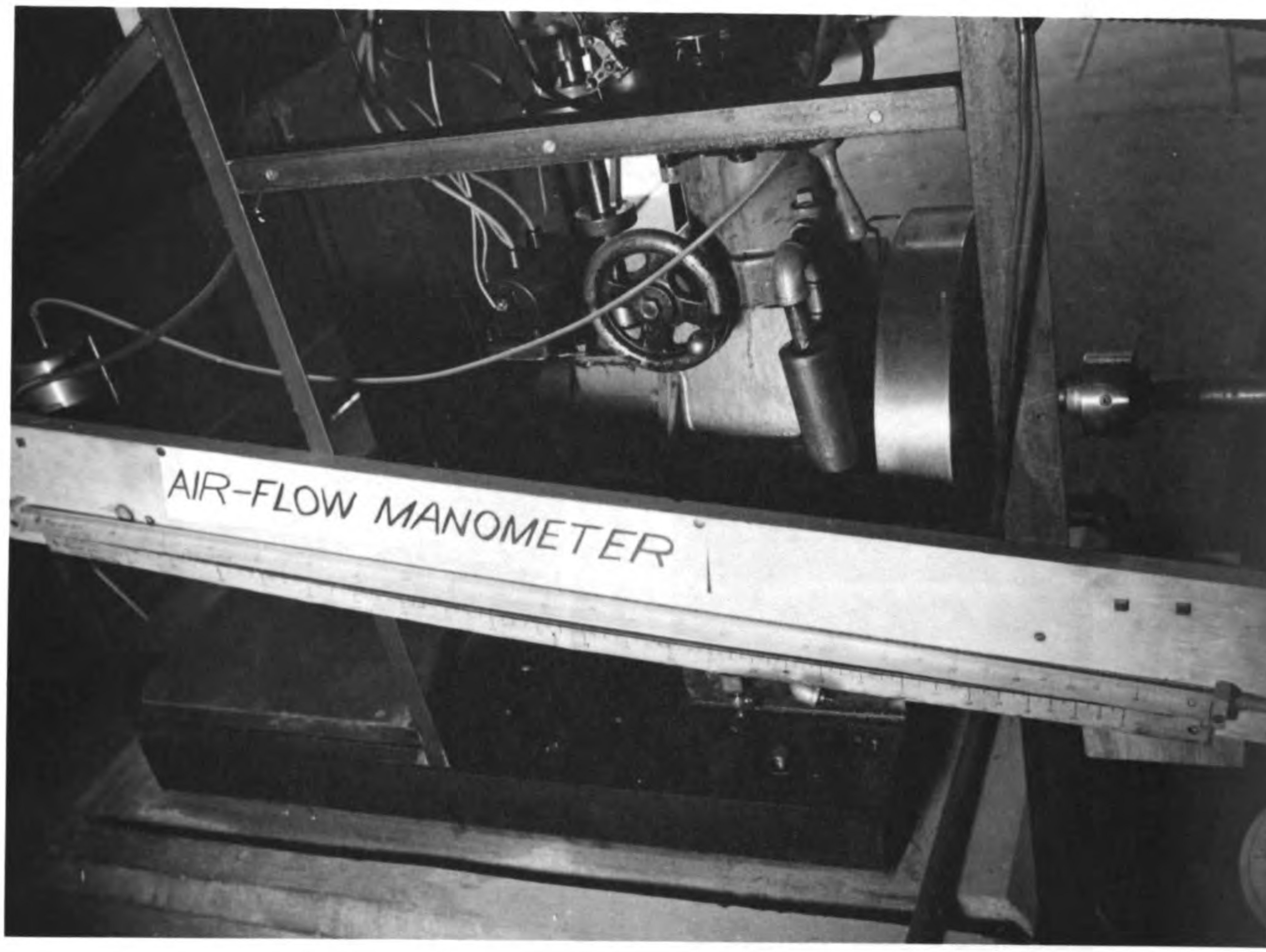


FIGURE 21

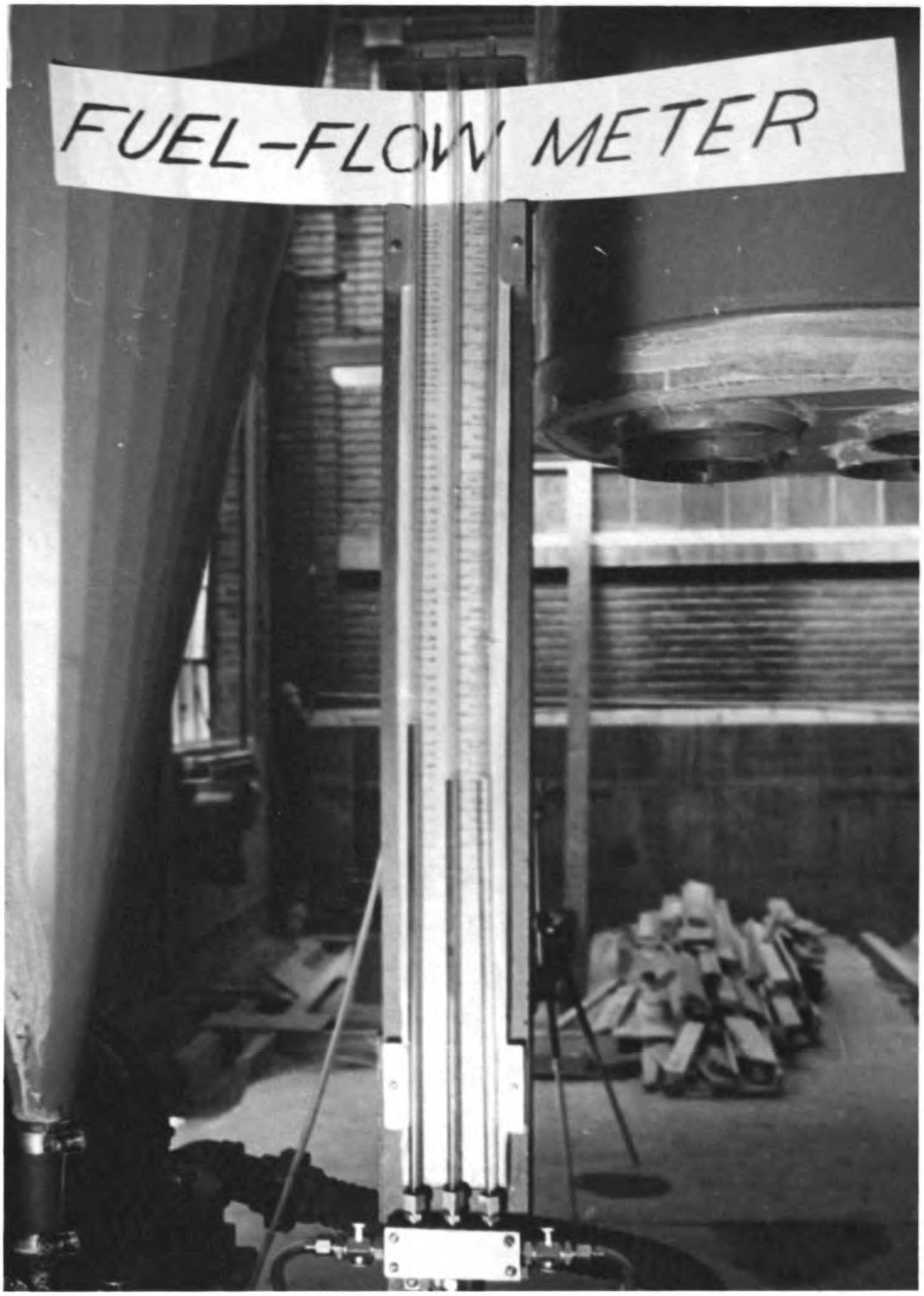


FIGURE 22

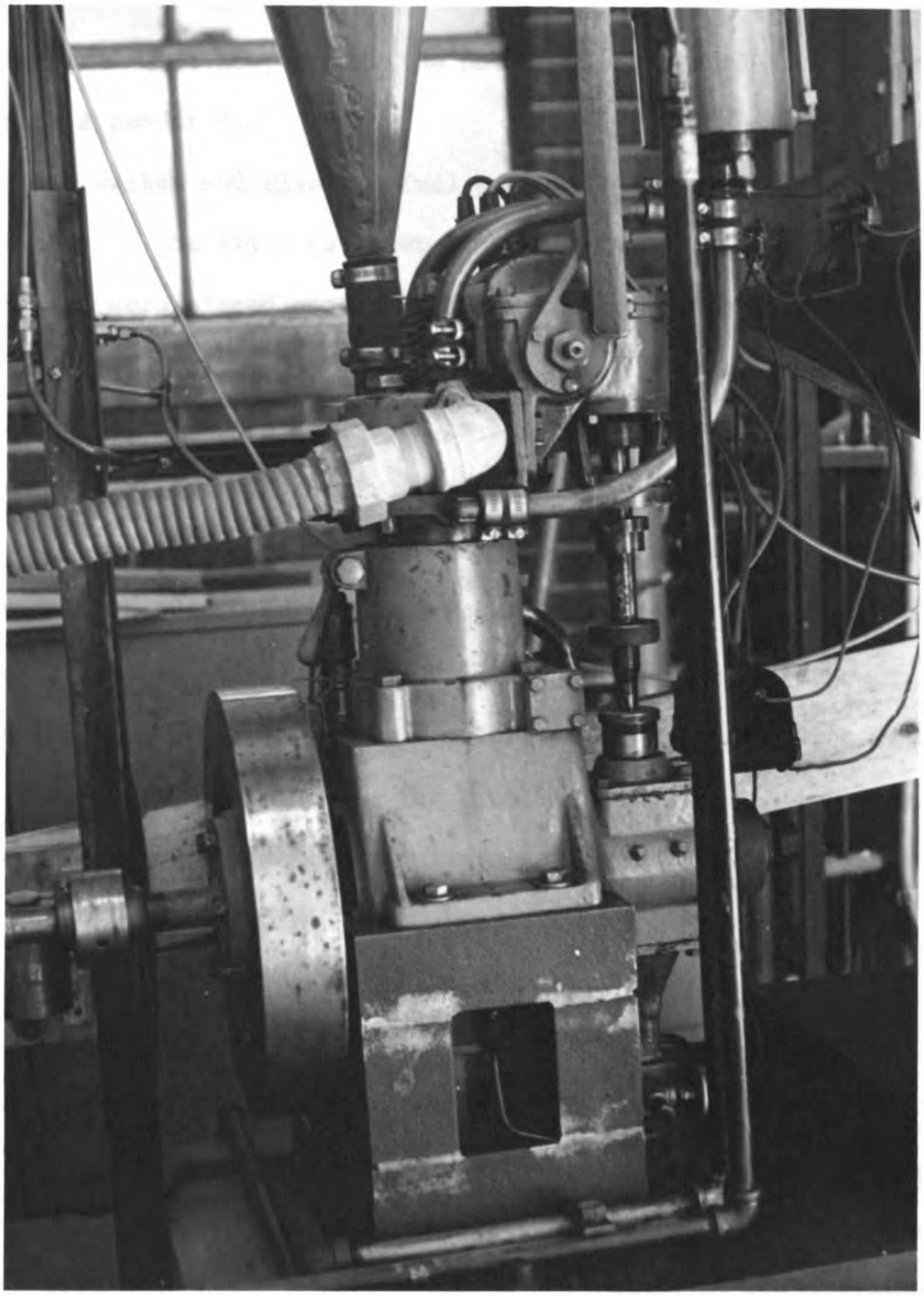


FIGURE 23

calibration procedure adopted for the nozzles in the fuel-flow meter are given in Appendix E.

Ignition switch and electric fuel pump switch were installed on the front side of the stand as shown in Figure 19. A 6-volt battery and fuel container were placed conveniently on the bottom shelf of the stand.

A 0-50 pound Toledo scale, seen in Figure 24, was connected to the dynamometer stator with a 12.065-inch long lever arm linkage giving a dynamometer constant of 5,000. The dynamometer on this installation was a General Electric type TLD 2242 Model 26G133 with electrical ratings of 250 volts and 27 amperes. It was rated at 10 horsepower as a generator and 7 horsepower as a motor, at any speed between 2,500 and 6,00 revolutions per minute. The coupling used between the dynamometer was designed for slight angular misalignment, seen in Figure 23.

Engine speed was indicated by the Standard Electric Time Unit. It consisted of a tachometer, a revolution counter, and an electric timer. The counter was connected electrically to the timer so that they could be started and stopped simultaneously. The timer was supplied with a control which would stop it automatically at 0.1 minute intervals. The revolutions per minute could be determined by multiplying the revolutions in 0.1 minute by ten, thus providing an accurate calibration correction for the tachometer. Figure 25 shows the control panel. An additional control resistance was fixed, seen in Figure 24, under the coupling in order to facilitate the control of the load on the engine without reaching the panel board.

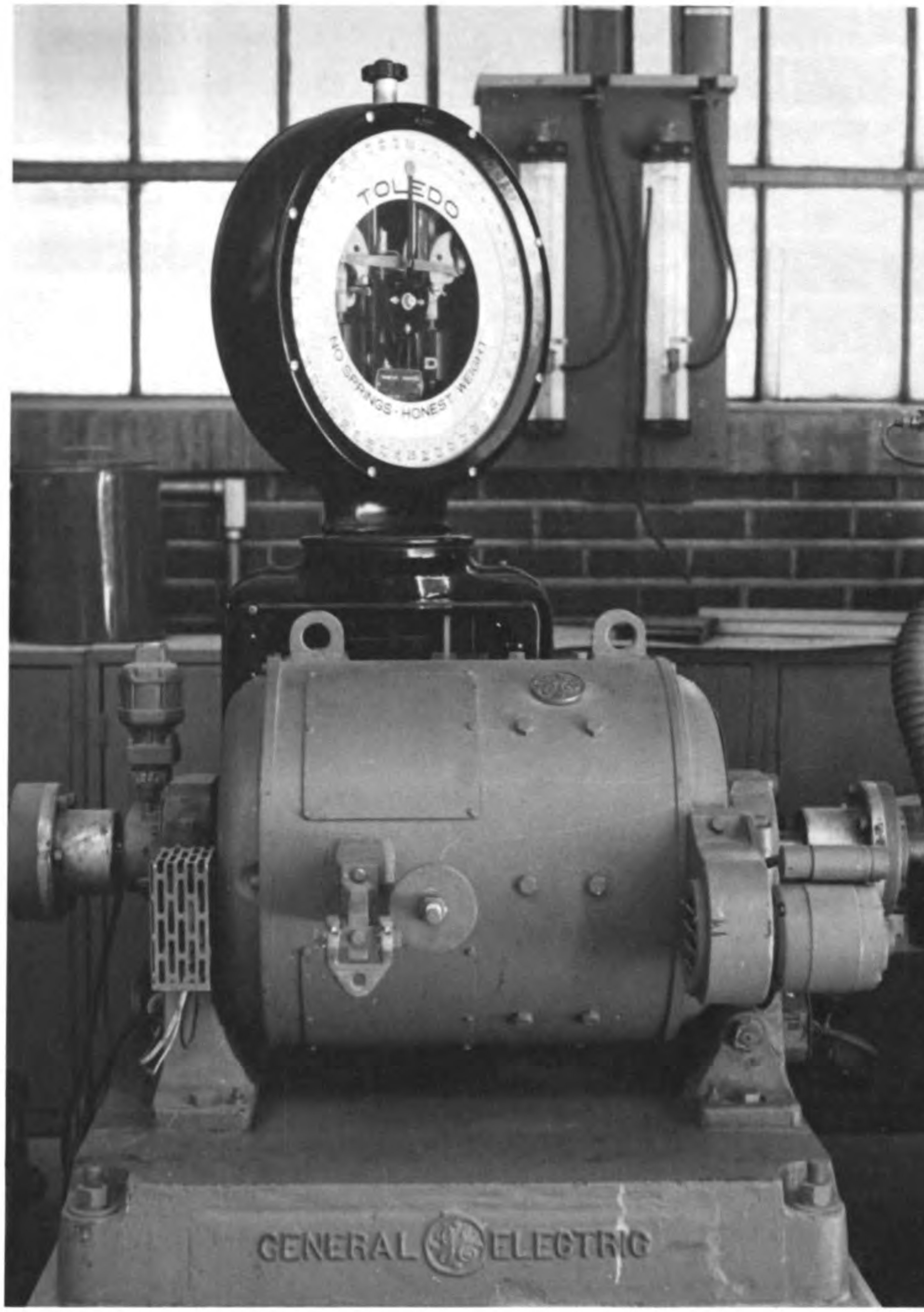


FIGURE 24

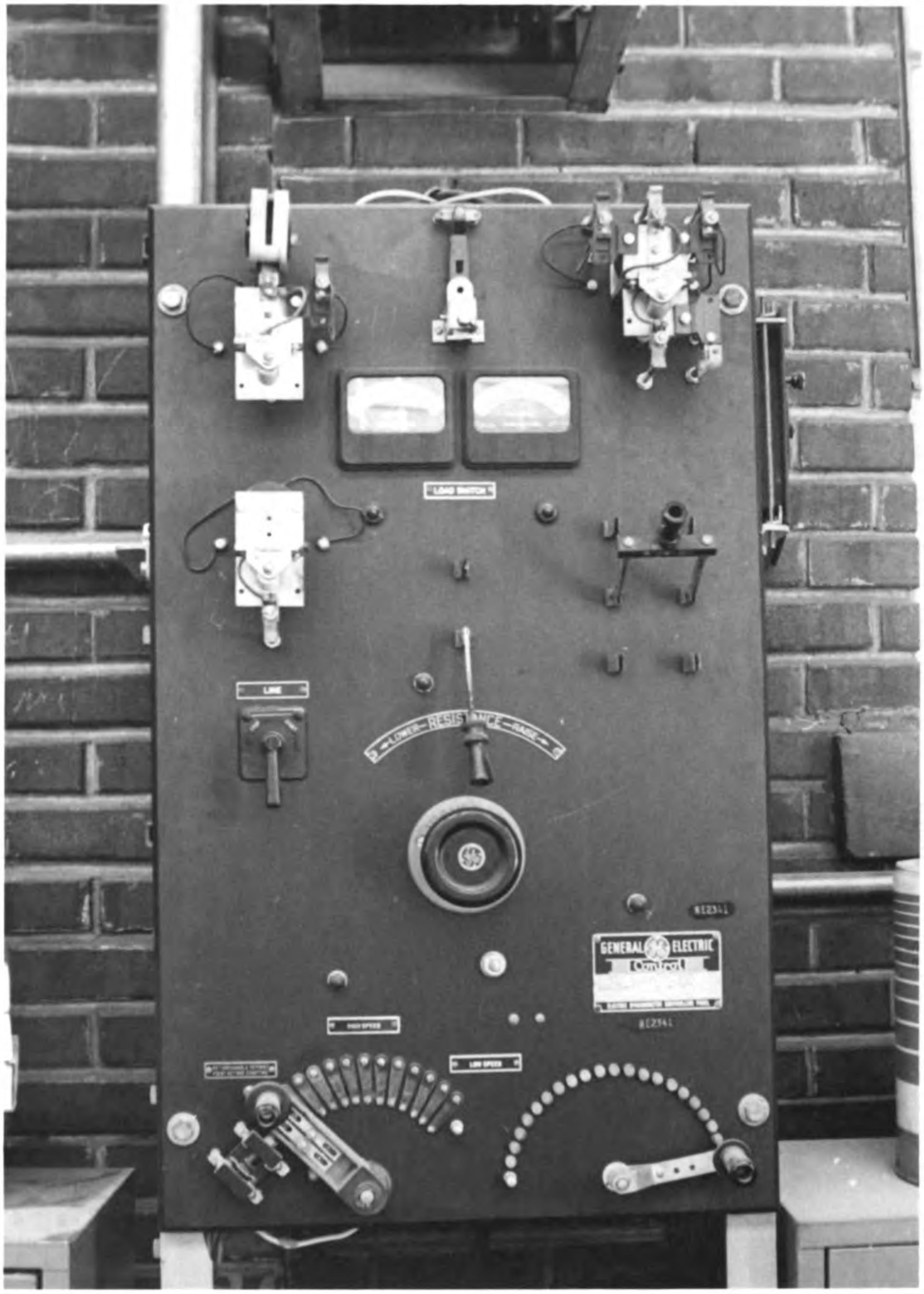


FIGURE 25

APPENDIX C

Dial zero reading and dial maximum reading were noted for the maximum and minimum compression ratio positions, respectively. Displacement volume and gasket volume were then measured, and clearance volumes were calculated at several dial readings. The detailed calculations are as follows:

Dial zero reading: 00000

Dial Maximum reading: 01570

Total volume: = displacement volume + clearance volume + gasket volume

Displacement volume = $\frac{3.14\ (3.066)^2}{4} \times \frac{9}{2} = 33.4$ cu. in.

Gasket volume = $0.090 \times 3.14\ \frac{(3.066)^2}{4} = 0.0665$ cu. in.

Clearance volume = $\frac{83 \times 1}{16.4} = 5.05$ cu. in.

1. At zero reading:

   clearance volume = $5.05 - (0.0665 + 0.155^*) = 4.9615$ cu. in.

   Total volume = $33.4 + 5.05 = 38.45$ cu. in.

   $\therefore$ compression ratio = $\frac{38.45}{4.96} = 7.7$

2. At 500 reading:

   Additional displacement = $0.3 \times \frac{3.14\ (3.066)^2}{4} = 2.22$ cu. in.

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*additional displacement volume = $0.210 \times 3.14\ \frac{(3.066)^2}{4} = 0.155$ cu. in.

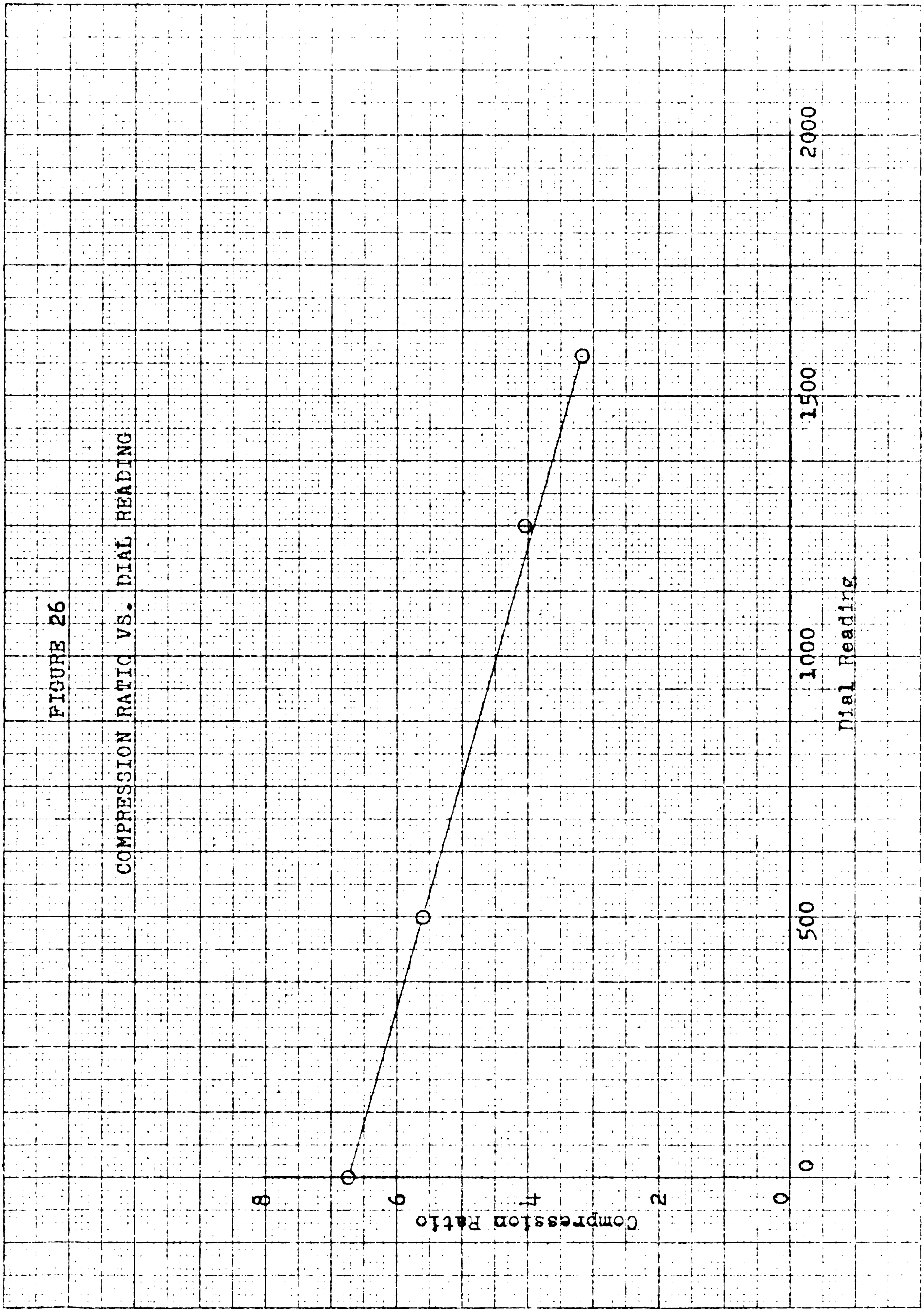

FIGURE 26

COMPRESSION RATIO VS. DIAL READING

$\therefore$ clearance volume = 2.2 + 0.0665 + 5.05 = 7.3165 cu. in.

$\therefore$ compression ratio = $\frac{40.7165}{7.3165} = 5.49$

3. At 1000 reading:

additional displacement volume = $\frac{0.8 \text{ X } 3.14\ (3.066)^2}{4} = 5.93$ cu.in.

clearance volume = 5.93 + 0.0665 + 5.05 = 11.0465 cu. in.

$\therefore$ compression ratio = $\frac{44.4465}{11.0465} = 4$ (approximately)

4. At 01570 reading:

additional displacement volume = $1.367 \text{ X } 3.14 \frac{(3.066)^2}{4} = 10.1$ cu.in.

clearance volume = 5.05 + 0.0665 + 10.1 = 15.2165 cu. in.

total volume = 10.1 + 0.0665 + 5.05 + 33.4 = 48.6165 cu. in.

$\therefore$ compression ratio = $\frac{48.6165}{15.6165} = 3.22$

Compression ratio versus dial reading was plotted in Figure 26 from which the required compression could be chosen and the dial reading set accordingly.

APPENDIX D

The flow rate of a gas can be determined by the following equation when value of discharge coefficient, diameter of orifice, density of the gas and the pressure differential are known (3):

$$w = c_1{}'A_2 \sqrt{2\ g\ \varsigma_1\ (P_1 - P_2)} \quad \text{pounds per second}$$

where w = flow rate in pounds per second

c_1'= discharge coefficient

A_2= area of cross=section of orifice in sq. ft.

g = acceleration due to gravity in ft./sec.^{2}

ς_1= density in cu. ft./ sec.

P_1-P_2= pressure differential in pounds/sq. ft.

It is felt that the detailed procedure of the determination of orifice sizes and number is unnecessary and therefore only the method of approach is given here.

A maximum pressure drop of five inches was assumed. The values of c_1' were taken for the pressure drop assumed in each case from table on page 632, reference 3. Orifice diameter was then calculated knowing flow rate determined by the dimensions and speed of the engine.

APPENDIX E

Flow through right hand side nozzle:

READING	DISTANCE (inches)	TIME (minutes)	FUEL FLOW (#/hr.)
1	2 7/30	15 46/100	0.97
2	6 27/32	8 4/100	1.868
3	13	6	2.5
4	21 9/16	4 33/100	3.72
5	25 5/8	3 67/100	4.10

Weight of fuel used in each case was 4 ounces. Calculations:

$$\text{Pounds of fuel/hour} = \frac{\text{ounces}}{\text{min.}} \text{ X } \frac{60}{16}$$

1. $\frac{240}{1546} \text{ X } \frac{100}{16} = \frac{15.5}{16} = 0.97$ pounds/hour

2. $\frac{240 \text{ X } 100}{804 \text{ X } 16} \text{ X } \frac{29.9}{16} = 1.868$ pounds/hour

3. $\frac{240}{6\text{X}16} = 2.5$ pounds/hour

4. $\frac{240 \text{ X } 100}{433 \text{ X } 16} = 3.72$ pounds/hour

5. $\frac{4 \text{ X } 60}{3.67 \text{ X } 16} = 4.1$ pounds/hour

Flow through left hand side nozzle:

READING	DISTANCE (inches)	TIME (minutes)	WEIGHT (ounces)	FUEL FLOW (#/hr.)
1	2 1/8	10	½	0.1875
2	9	7 40/100	1	0.506
3	17 3/32	5	1	0.750
4	21 1/16	4 35/100	1	0.867
5	28	3 75/100	1	1.00

Calculations:

$$\text{pounds of fuel/hour} = \frac{\text{ounces}}{\text{min.}} \times \frac{60}{16} \quad \text{since 1 pound = 16 ounces}$$

1. $\frac{60}{16} \times \frac{1}{2} \times \frac{1}{10} = 0.1875$ pounds/hour

2. $\frac{60}{16} \times \frac{100}{140} = 0.506$ pounds/hour

3. $3.75 \times \frac{1}{5} = 0.75$ pounds/hour

4. $3.75 \times \frac{100}{435} = 0.863$ pounds/hour

5. $3.75 \times \frac{100}{375} = 1.00$ pounds/hour

Graphs were plotted pounds per hour versus distance for both the nozzles on a log log scale (Figures 27 and 28).

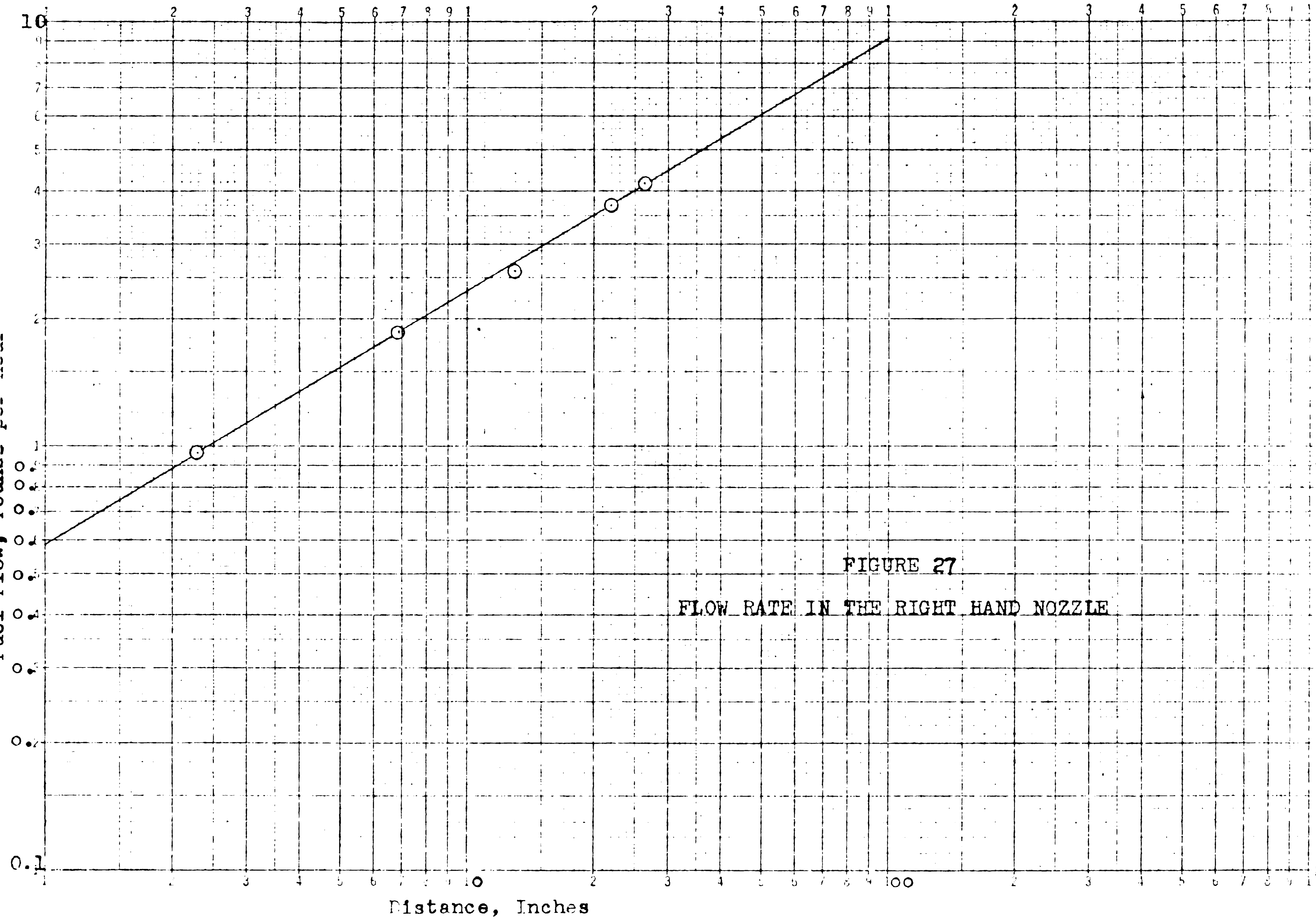

FIGURE 27

FLOW RATE IN THE RIGHT HAND NOZZLE

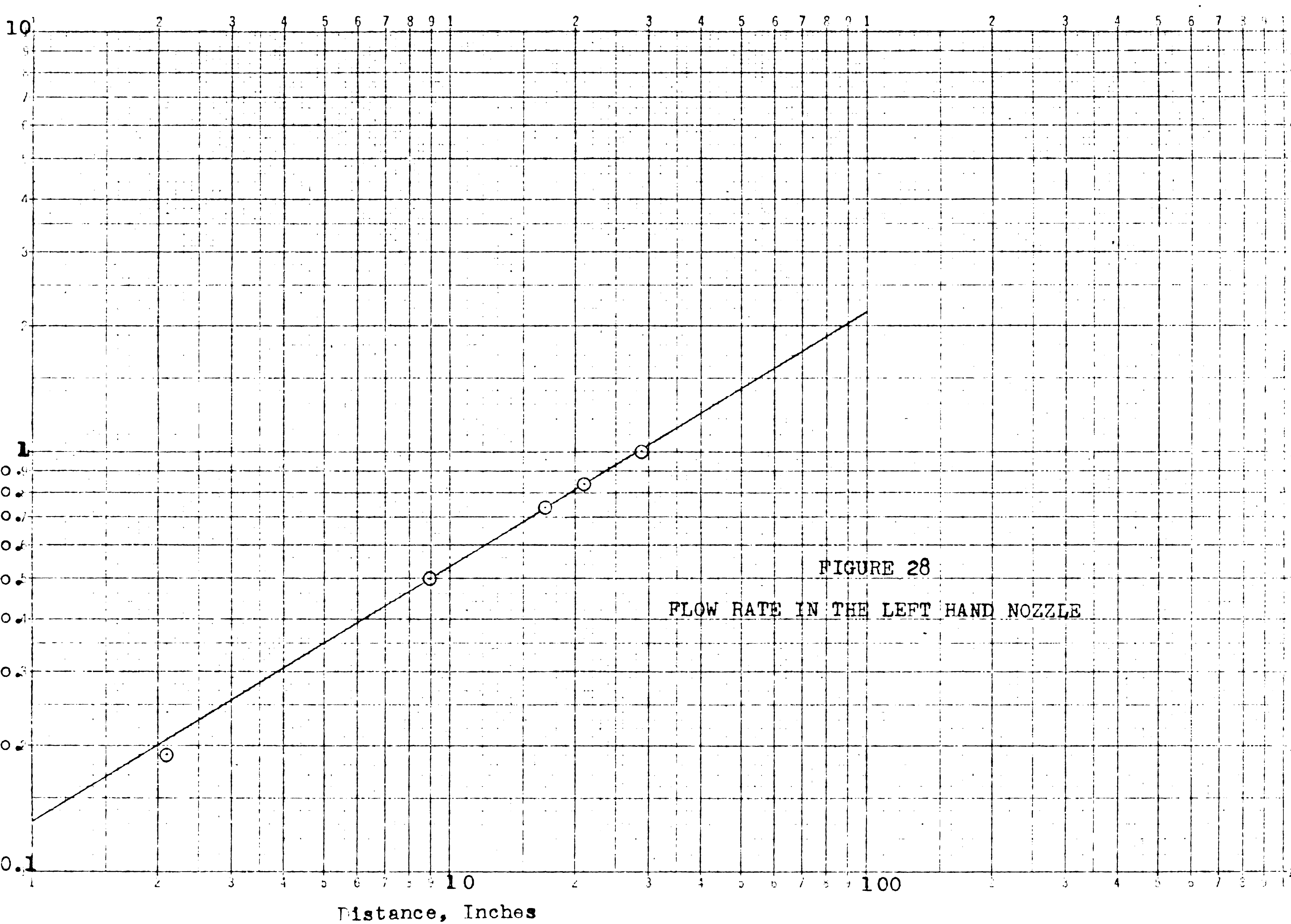

FIGURE 28

FLOW RATE IN THE LEFT HAND NOZZLE

APPENDIX F

Sample calculations:

$$\text{Horse power} = \frac{WRN}{5252.1}$$

where N = revolutions per minute

R = lever arm in feet

W = force acting at radius (R) in pounds

For the dynamometer, the constant in the horse power equation was 5000.

$$\therefore R = \frac{5252.1}{5000} = 1.05 \text{ feet}$$

Torque = arm X scale load = 1.05 X scale load

$$\text{Brake - mean effective pressure} = \frac{150.8\ T}{D}$$

where T = torque, foot pounds

D = displacement, cubic inches

$$\therefore \text{B. M. E. P.} = \frac{150.8 \text{ X } 1.05 \text{ X scale load}}{D} =$$

$$\frac{158 \text{ X scale load}}{D}$$

Bore = 3.066 inches

Stroke = 4.5 inches

$$\text{Displacement, } D = \frac{X\ d^2}{4} \text{ X } 1 = \frac{(3.066)^2}{4} \text{ X } 4.5 = 33.22 \text{ cu. in.}$$

$$\text{B. M. E. P.} = \frac{158}{33.22} \text{ X scale load} = \frac{158}{33.2} \text{ X } 14.2 = 67.6 \text{ p. s. i.}$$

$$\text{B. H. P.} = \frac{WN}{5000} = \frac{14.2 \text{ X } 2400}{5000} = 6.815$$

Brake specific fuel consumption = $\frac{\text{pounds fuel per hour}}{\text{B. H. P.}}$

$\therefore$ B. S. F. C. $= \frac{8.43}{6.815} = 1.24$ pounds fuel/B. H. P.-hr.

Air-fuel ratio = $\frac{\text{pounds air/hour}}{\text{pounds fuel/hour}} = \frac{70}{8.43} = 8.31$ pounds air/ # fuel

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