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FUEL CONSUMPTION OF A CARBURETOR ENGINE AT VARIOUS SPEEDS AND TORQUES

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SUMMARY

An investigation was conducted to obtain fuel-consumption curves for a single-cylinder engine with a Wright 1820-G and a Pratt & Whitney 1340-H cylinder at varying speeds, manifold pressures, and air-fuel ratios. The 1340-H cylinder was tested at speeds from 1,200 to 2,400 r.p.m. and at manifold pressures from 21 to 38 inches of mercury absolute. Less extensive tests were made of the 1820-G cylinder.

The results of the tests showed that the minimum brake fuel consumption was obtained when the engines were operating at high torques and at speeds from 60 to 70 percent of the rated speed. The fuel consumption increased at an increasing rate as the torque was reduced; and, at 45 percent of maximum torque, the fuel consumption was 20 percent higher than at maximum torque when the engines were operating at 70 percent of rated speed. Minimum specific fuel consumption was obtained at the same air-fuel ratio regardless of compression ratio. No improvement in fuel consumption was obtained when mixtures leaner than an air-fuel ratio of 15.5 were used. The leanest mixture ratio on which the engine with the 1340-H cylinder would operate smoothly was 18.5 and the spark advance for maximum power with this mixture ratio was 50° B.T.C. A method is discussed for reducing the amount of testing necessary to obtain curves for minimum brake fuel consumption.

INTRODUCTION

The fuel consumption of an aircraft engine is an important consideration, especially if the engine is used in long-range aircraft. Even the most efficient engines will consume their weight in fuel in 3 to 5 hours of flying depending on what percentage of the total power is being used.

During the past few years, a large improvement has been made in the fuel consumption of aircraft engines. This improvement has resulted principally from the fact that better cooling of the cylinders and valves and the availability of fuels of high antiknock value make it possible to operate at higher compression ratios and with leaner mixtures. Increased cylinder turbulence and more uniform mixture distribution have also resulted in reduced fuel consumption. Automatic mixture controls and mixture-strength indicators are being used to assure that economical operation is maintained in flight. These devices are very essential when constant-speed propellers are used because the pilot cannot lean his mixture to the point where the engine speed starts to fall off and then enrich the mixture slightly, as is the practice when fixed-pitch propellers are used. The mixture-control and indicating devices serve to establish the most economical mixture setting for a particular speed and manifold pressure but not the most economical engine operating condition. The engine speed and the manifold pressure for the most economical operation must be determined by testing the engine.

An investigation has been completed at the N.A.C.A. laboratory at Langley Field, Va., to determine the effect of engine torque and speed on the fuel consumption.

Two single-cylinder air-cooled engines were used in these tests: one cylinder is from a Pratt & Whitney 1340-H Wasp engine, which is rated at 550 horsepower at 2,200 r.p.m. at 8,000 feet altitude; and the other is from a Wright 1820-G Cyclone engine, which is rated at 800 horsepower at 2,100 r.p.m. at sea level.

EQUIPMENT

<u>Test engine</u>.- A photograph of the single-cylinder engine with some of the test equipment is shown in figure 1. A diagrammatic sketch showing the arrangement of the equipment is given in figure 2. The engine consisted essentially of a universal test-engine base on which the two service air-cooled cylinders were mounted by means of an adapter (reference 1). The adapter also served as a housing for

cams, valve gear, and cam followers. The engine was coupled to an electric dynamometer.

Cylinders from a Wright 1820-G Cyclone and a Pratt & Whitney 1340-H Wasp engine were used in these tests (fig. 3). The cylinders were equipped with standard pistons. The compression ratio, the length of stroke, and the valve timing of each of the two test engines compared with the standard engines are given in the following table.

ENGINE CHARACTERISTICS

Cylinder	Compression ratio	Stroke (in.)	Valve timing (0.010-in. cold clearance)			
			Intake		Exhaust	
			Open (deg. B.T.C.)	Close (deg. A.B.C.)	Open (deg. B.B.C.)	Close (deg. A.T.C.)
Pratt & Whitney 1340-H:						
Standard	6.0	5-3/4	60	125	90	56
Test	5.6	6	23	69	80	10
Wright 1820-G:						
Standard	6.4	6-7/8	47	70	104	56
Test	7.4	7	13	54	70	39

In order to obtain sufficient piston and cylinder-wall lubrication for the test engine, oil was supplied under a pressure of 2 pounds per square inch to six equally spaced holes 1/16 inch in diameter about 2 inches below the mounting flange.

The engine was equipped with a Stromberg NAL-5 carburetor, modified by installing needle valves in the main jets to regulate the fuel flow.

The fuel consumed was measured by a small weighing tank suspended from a sensitive beam balance, which electrically controlled the operation of a revolution counter and a stop watch. The length of a fuel run was the time required to consume one-half pound of fuel.

The carburetor-air system consisted of an independently driven supercharger, an air cooler, regulating valves, and a surge tank. A 4-inch N.A.C.A. Roots type supercharger supplied air at pressures higher than atmospheric. A large water-cooled aftercooler was used to maintain the carburetor-air temperatures at approximately 85° F. The valves between the supercharger and the surge tank served as throttling valves when the engine was operating with less than atmospheric pressure in the manifold and as an auxiliary control when the desired boost pressure could not be obtained by the supercharger speed control. The surge tank served to damp out the pressure pulsations from the supercharger and the engine and also as a depression tank when the manifold pressure was reduced to less than atmospheric.

The cooling-air system, shown in figures 1 and 2, consisted of an orifice tank for measuring the quantity of cooling air supplied, a centrifugal blower for forcing the air past the cylinder, two 30-kilowatt air heaters, the ducts for conveying the air, and the aluminum jacket enclosing the cylinder (reference 1). During tests with the 1340-H cylinder at speeds of 2,100 and 2,400 r.p.m., when the manifold pressure was higher than atmospheric, the cooling was augmented by spraying water into the air stream.

Instruments.- Liquid thermometers were used to measure the room, the carburetor-air, and the oil-out temperatures.

Water manometers were used to measure the pressures in the orifice tank and at the jacket inlet, and a mercury manometer was used to measure the manifold pressures.

Measurements of mixture strength were obtained with a commercial air-fuel-ratio indicator, which gave a continuous reading of air-fuel ratio based on measurements obtained from the variation in the composition of the exhaust gas for air-fuel ratios varying from 10 to 15; above an air-fuel ratio of 15, the indicator did not given consistent results. For mixtures leaner than 15, the air-fuel ratio was determined in a few cases by analyzing samples of the exhaust gas.

METHODS AND TESTS

The initial test condition for each fuel-consumption run is shown in the following table.

INITIAL TEST CONDITIONS

Engine speed (r.p.m.)	Gross b.m.e.p. (lb./sq. in.)	Carburetor pressure (in. Hg absolute)
Pratt & Whitney 1340-H cylinder		
1,200	160	37.1
	141	33.7
	116	29.2
	92	25.0
	71	21.3
1,500	160	36.7
	140	33.2
	116	29.0
	93	25.0
	68	20.6
1,800	159	36.7
	136	32.6
	115	28.8
	92	24.7
	70	20.8
2,100	157	37.5
	138	34.2
	116	30.3
	91	26.0
	70	22.3
2,400	(a)	(a)
	137	36.9
	116	32.9
	92	28.3
	70	24.1
Wright 1820-G cylinder		
1,300	112	24.0
1,500	135	28.4
	112	24.6
	91	21.4
	71	18.1
1,700	111	26.1
1,900	112	28.1

[a]Not obtained because of insufficient supercharger capacity.

The engine speed and manifold pressure were held constant but the torque changed as the fuel consumption was varied. For each test condition, at least five mixture runs were made over a range of mixtures from an air-fuel ratio of 10 to the limit of stable operation. The limit of stable engine operation varied from an air-fuel ratio of 15 to 18.5, depending upon the engine speed and the manifold pressure. The spark timing was adjusted for maximum power for each mixture setting. For the leanest mixtures, the spark advance in all cases varied from 50° to 60° B.T.C., depending upon the engine speed and the manifold pressure.

The friction of the engine was determined by motoring it at the manifold pressures and the speeds used in the power runs. The lubricating oil and the cooling air were heated so as to maintain the oil-out temperature and the cylinder temperature at 160° F. and 250° F., respectively. With a cylinder temperature of 250° F., the viscosity of the oil on the cylinder walls closely simulated that for actual operating conditions. The friction curves obtained with each cylinder are shown in figures 4(a) and (b).

The gross brake power readings were corrected for the power required to compress the carburetor air; an over-all adiabatic efficiency of 70 percent was assumed. The indicated power readings were obtained by adding the gross brake to the friction readings.

Gasoline conforming to the Army specification Y-3557 and having an octane number of 87 was used for most of the tests. For the most severe operating conditions, a sufficient amount of ethyl fluid was added to the fuel to suppress audible knock.

RESULTS AND DISCUSSION

The curves in figure 5 show the relation between the brake specific fuel consumption and the brake mean effective pressure for a large range of manifold pressures and engine speeds for the 1340-H cylinder. The curves in figure 6 show the same data on an indicated basis. The fuel consumption and the brake mean effective pressures for the runs in which the supercharger was used have been corrected for the power required to supply air at pressures higher than atmospheric. Curves for air-fuel ratios of 11, 12, 13, and 15 are also shown in figures 5 and 6.

The decrease in minimum specific brake fuel consumption at constant air-fuel ratio with increase in torque is to be expected because the mechanical efficiency of the engine increased. At 70 percent of the rated speed, the fuel consumption decreased from 0.557 to 0.460 pound per brake horsepower per hour as the torque was increased from a brake mean effective pressure of 65 to 127 pounds per square inch. At a speed of 1,200 r.p.m., the mechanical efficiency increased from 81.7 percent at a brake mean effective pressure of 65 pounds per square inch to 91 percent at a brake mean effective pressure of 150 pounds per square inch and, at 2,100 r.p.m. for the same torque values, it increased from 70.7 to 84.8 percent. When the engine speed was increased and the torque kept constant, the mechanical efficiency of the engine decreased because the pumping and the mechanical losses increased. At a brake mean effective pressure of 150 pounds per square inch, the mechanical efficiency is 91 percent at 1,200 r.p.m. as compared with 84.8 percent at 2,100 r.p.m. The difference in brake fuel consumption, except for the low torques at low speeds, is directly proportional to the change in the mechanical efficiency. The exception is shown by the data for indicated fuel consumption in figure 6. This increased fuel consumption at 1,200 r.p.m. is probably caused by poor carburetion and reduced turbulence, as difficulty was experienced in obtaining stable operation for these conditions when the air-fuel ratio was more than 15.

The minimum indicated fuel consumption in all tests on the 1340-H cylinder is about 0.4 pound per indicated horsepower per hour, except for speeds of 1,200 r.p.m. These tests therefore show that the minimum fuel consumption on a brake basis may be obtained for a wide range of operating conditions from the mechanical efficiency and the minimum indicated fuel consumption for one condition as follows: Establish for a constant engine speed and a manifold pressure a curve of indicated fuel consumption against mixture ratio. Determine the friction and the brake mean effective pressures over the desired range of engine speeds and manifold pressures for use in determining the mechanical efficiency. Divide the minimum indicated fuel consumption by the mechanical efficiency for a particular condition to obtain the minimum brake fuel consumption for the specified condition. This method of calculating minimum brake fuel consumption would dispense with the establishment of a mixture curve for each condition.

For a constant engine speed and manifold pressure, the fuel consumption at the maximum power mixture is from 10 to 15 percent higher than that obtained at the maximum economy mixture. Likewise, the mechanical efficiency, based on conditions with the maximum-power mixture, is from 1/2 to 1-1/2 percent higher than the mechanical efficiency based on conditions with the maximum-economy mixture. The larger difference in mechanical efficiency is obtained at low torque values.

The data for the 1340-H and the 1820-G cylinders are plotted for comparative purposes in figure 7. The difference between the fuel consumption for these two cylinders is caused largely by the difference in compression ratio. Tests conducted on another single-cylinder engine with varying compression ratio showed that, when the compression ratio was increased from 5.6 to 7.4, the fuel consumption was reduced 0.035 pound per indicated horsepower per hour (reference 2). The increased turbulence obtained with the 1820-G cylinder by locating the intake so as to obtain a tangential flow may have accounted for the remaining difference between the 1820-G and the 1340-H cylinders. Apparently the increased turbulence for the 1820-G cylinder was sufficient to cause the mixture to burn earlier in the cycle, even though the spark setting was about 5° later with maximum-power mixtures and about 10° later for maximum-economy mixtures than for the 1340-H cylinder.

The air-fuel-ratio curves in figures 7 and 8 show that little, if any, improvement in specific fuel consumption can be obtained by operating with mixtures leaner than 15 or 16. On the basis of these tests it is believed that, if a perfectly homogeneous mixture could be obtained, there would be no gain in economy by using mixtures leaner than the chemically correct one for an engine of high output. When good mixing and distribution are obtained and higher output is essential, as in military engines, there is no reason for operating with a deficiency in either fuel or air. A deficiency in air results in wasted fuel and a deficiency in fuel results in wasted air, which is also objectionable, especially on a highly supercharged engine. In practice, however, particularly on multicylinder engines, perfect mixing and distribution are not obtained and, as a result, the minimum specific fuel consumption is obtained with mixtures slightly leaner than the chemically correct mixture. A comparison of tests from a large number of full-size engines with the present single-cylinder engine tests has shown that there is little difference, if any, in the minimum indicated fuel consumption. The minimum fuel consumption for the multicylinder engine is obtained, however, with a slightly leaner mixture than for the single-cylinder engine.

The curves in figures 7 and 8 also show that an engine with a compression ratio of 5.6 will burn as lean a mixture as an engine with a compression ratio of 7.4.

The curves in figure 9 show the minimum brake fuel consumption of the 1340-H cylinder as affected by engine speed and torque. As the engine speed is increased or the torque reduced, the fuel consumption increases because the mechanical efficiency decreases. The curves show that maximum economy is obtained while the engine is operating at high torque and at speeds of about 60 to 70 percent of the maximum rated speed. As the mechanical-efficiency curves for the single-cylinder engine with varying speed and torque are of the same shape as for a multicylinder engine, the fuel-consumption curves for the single-cylinder engine should closely approach those for a multicylinder engine. A comparison of the mechanical-efficiency curves for the single-cylinder engine with those for a moderately supercharged two-row radial engine shows that, as the speed of each is increased from 1,600 to 2,400 r.p.m. at high torque, the mechanical efficiency of each decreases 5 percent. At an engine speed of 2,400 r.p.m., the difference in mechanical efficiency at maximum torque and at two-thirds of maximum torque is 6 percent for each engine. The fuel-consumption values given by the curves of figure 9 should therefore be applicable to a multicylinder moderately supercharged engine. It must be appreciated, however, that in the application of these fuel-consumption values to an engine in flight, other aerodynamic features must be considered so as to obtain optimum fuel consumption per mile.

CONCLUSIONS

The results from these tests of a single-cylinder engine show that:

1. In order to obtain minimum specific brake fuel consumption, an engine should be operated at high torque and at speeds from 60 to 70 percent of the rated speed. Operating at 45 percent of maximum torque increased the

fuel consumption 20 percent over the fuel consumed at maximum torque when the engine was operating at 70 percent of rated speed.

2. The indicated mean effective pressure and the engine speed had only a small effect on the minimum indicated fuel consumption within the practical range of operation.

3. An engine having a compression rato of 5.6 can burn as lean a mixture as an engine having a compression ratio of 7.4.

4. Practically no improvement in fuel consumption was obtained by operating with mixtures leaner than an air-fuel ratio of 15.5.

5. A method for reducing the amount of testing necessary to obtain minimum brake fuel-consumption curves is proposed.

Langley Memorial Aeronautical Laboratory,
National Advisory Committee for Aeronautics,
Langley Field, Va., May 10, 1938.

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2. Schey, Oscar W., and Young, Alfred W.: The Use of Large Valve Overlap in Scavenging a Supercharged Spark-Ignition Engine Using Fuel Injection. T.N. No. 406, N.A.C.A., 1932.

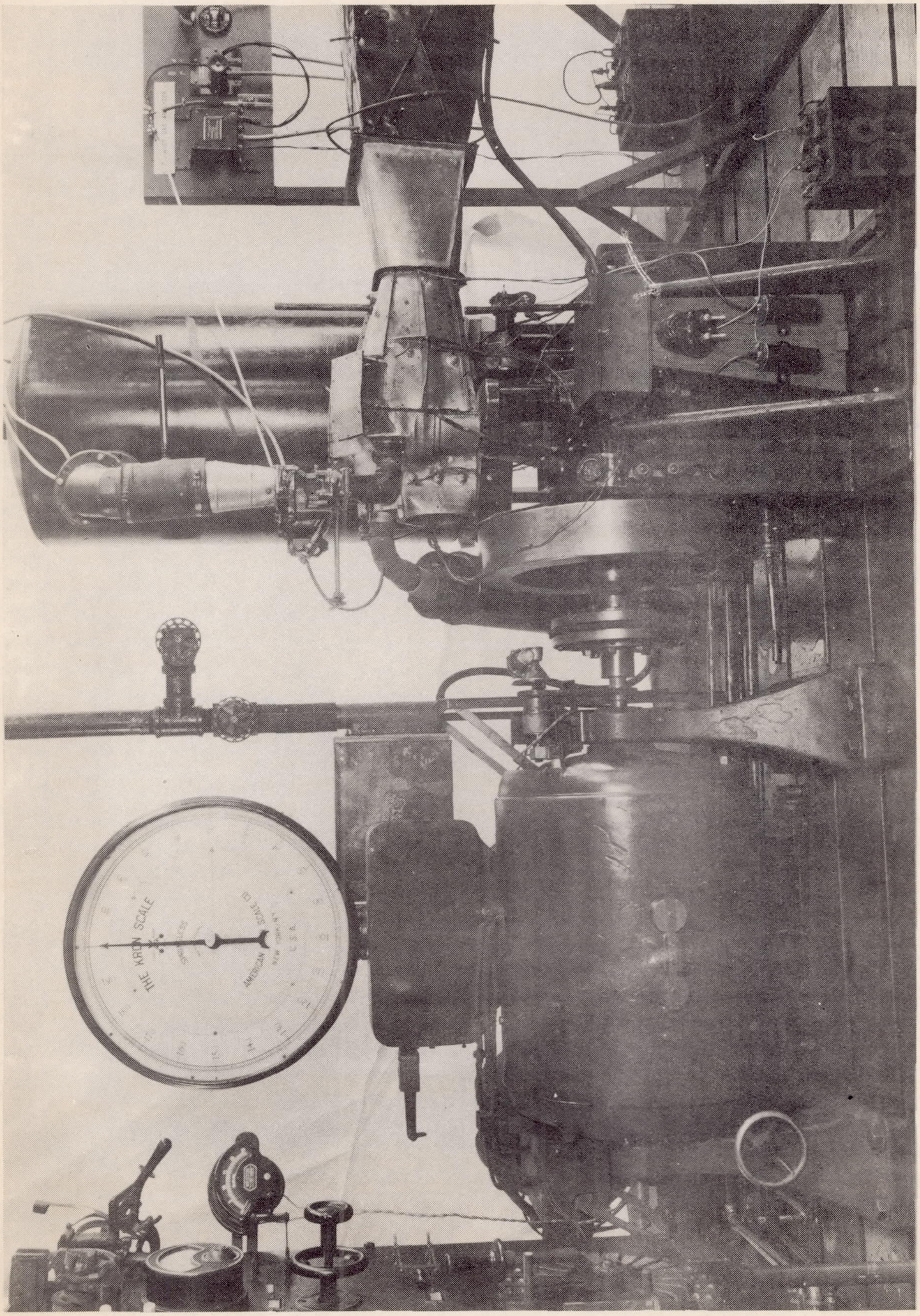
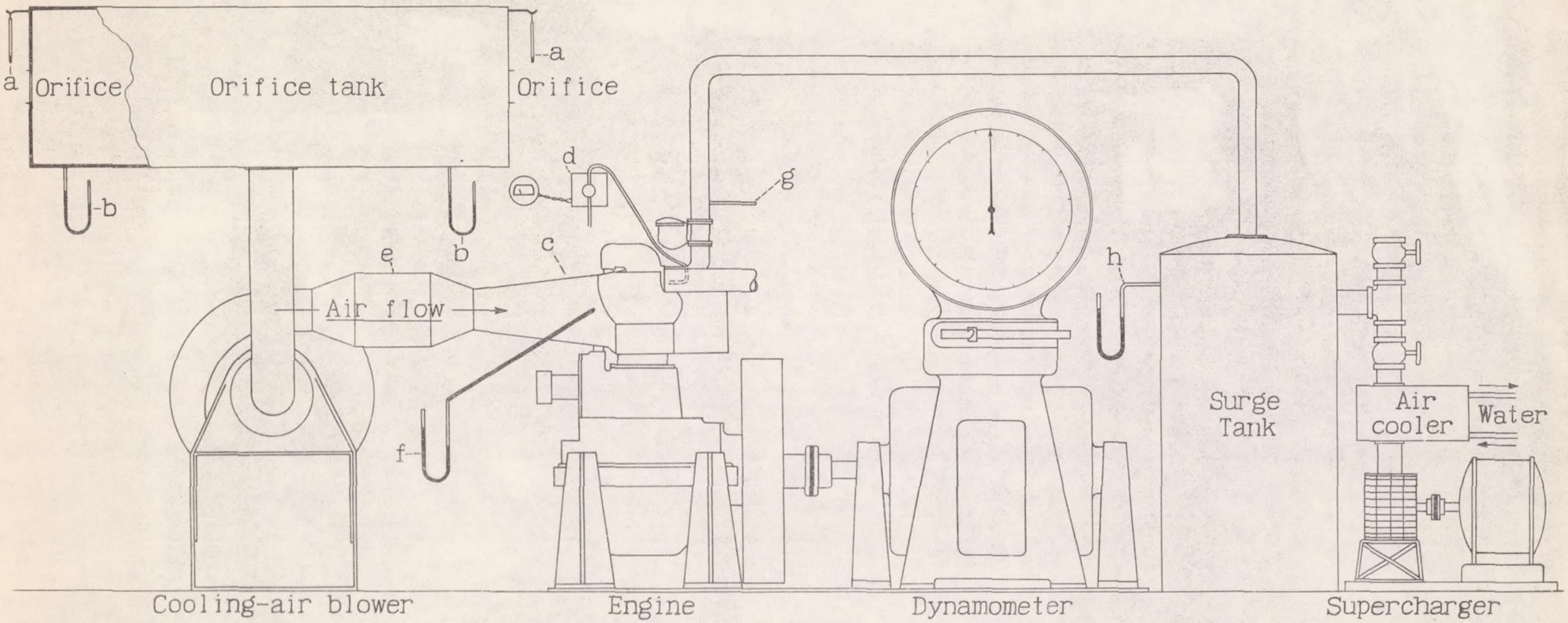
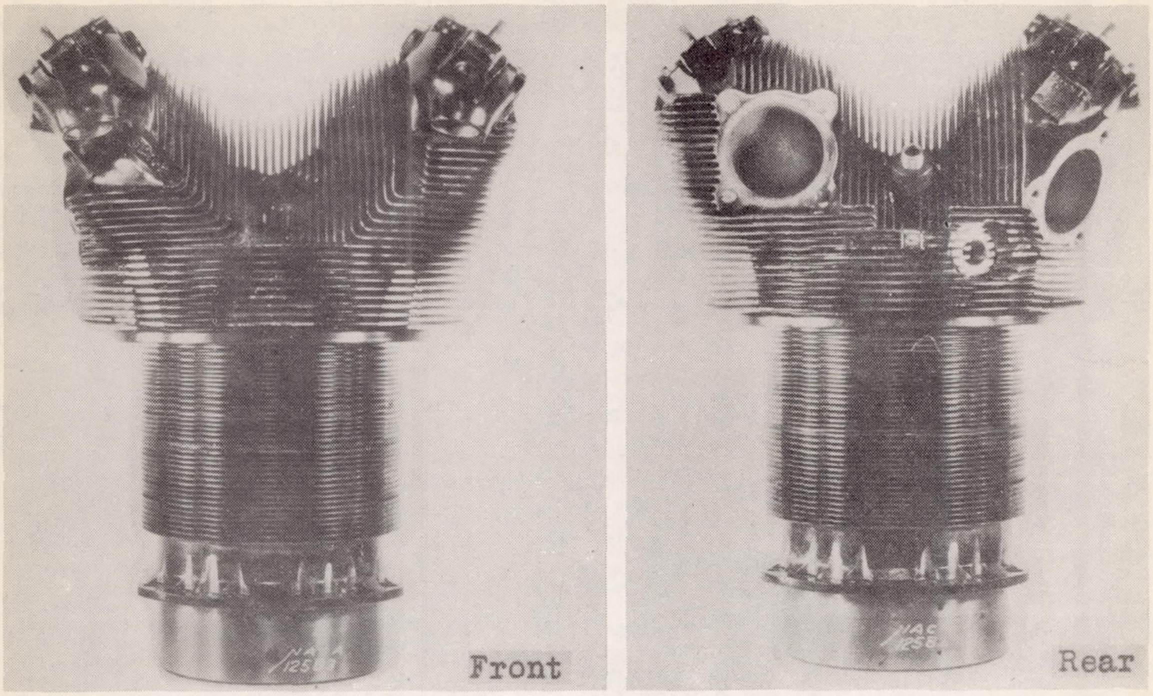


Figure 1.- Set-up of single cylinder air-cooled engine.

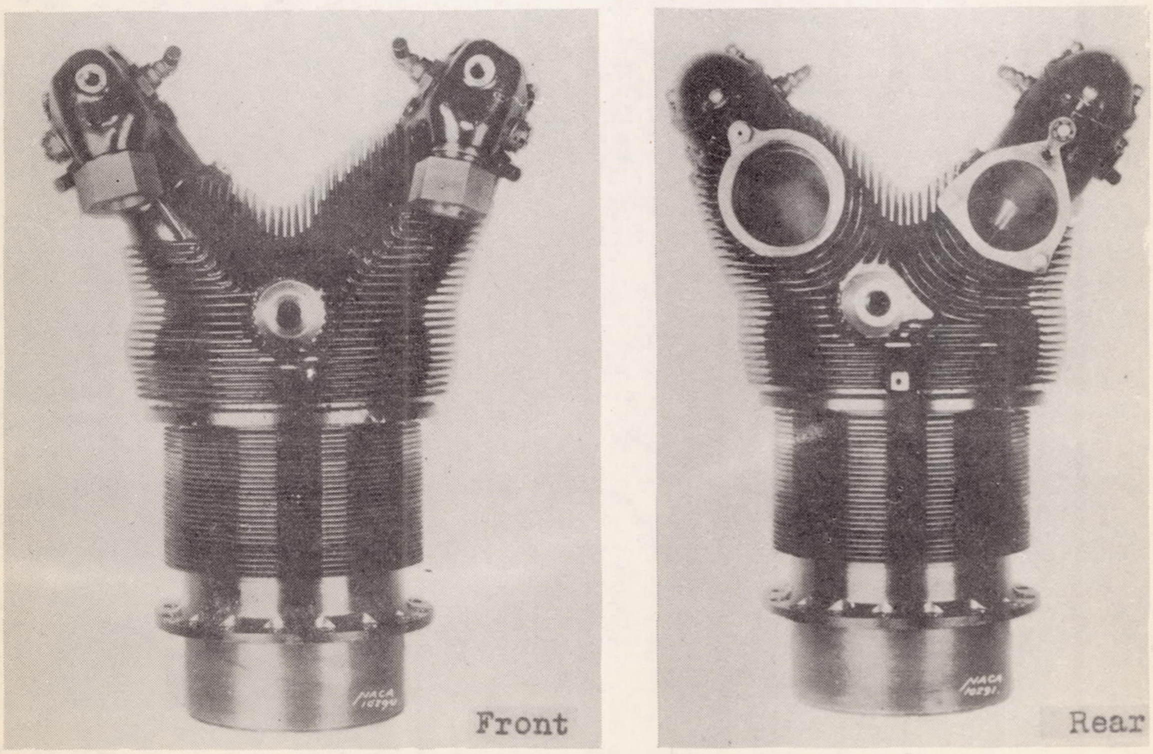


a. Thermometer
b. Water manometer
c. Cylinder-air jacket
d. Fuel-air-ratio indicator
e. Two 30 kw electric air-heaters
f. Static-pressure manometer
g. Carburetor-air thermometer
h. Manometer

Figure 2.-Diagrammatic layout of equipment.

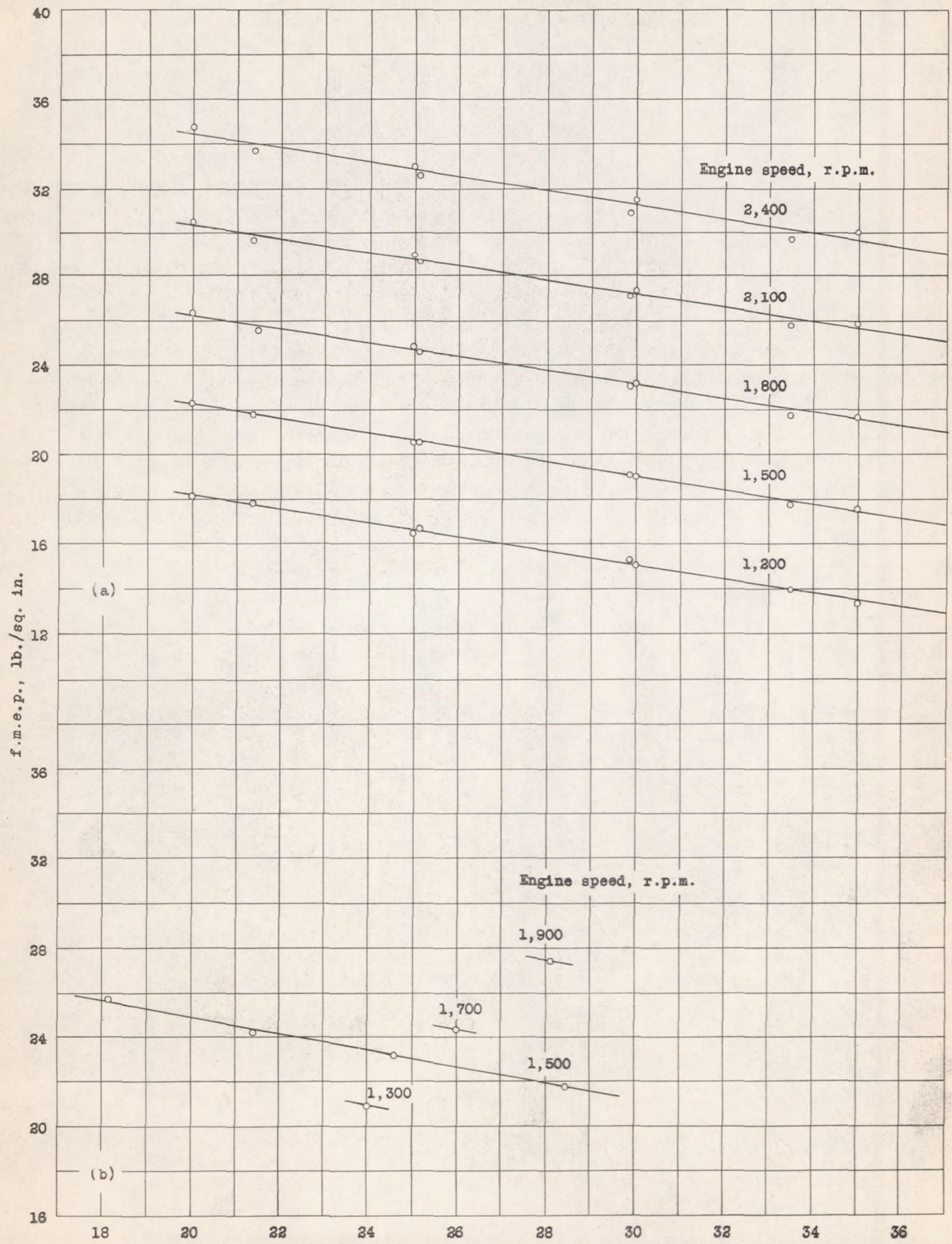


(A) Wright 1820-G cylinder.



(B) Pratt & Whitney 1340-H cylinder

Figure 3.- Front and rear views of cylinders.



(a) Pratt and Whitney 1340-H (b) Wright 1820-G

Figure 4.- Friction curves for the engines tested.

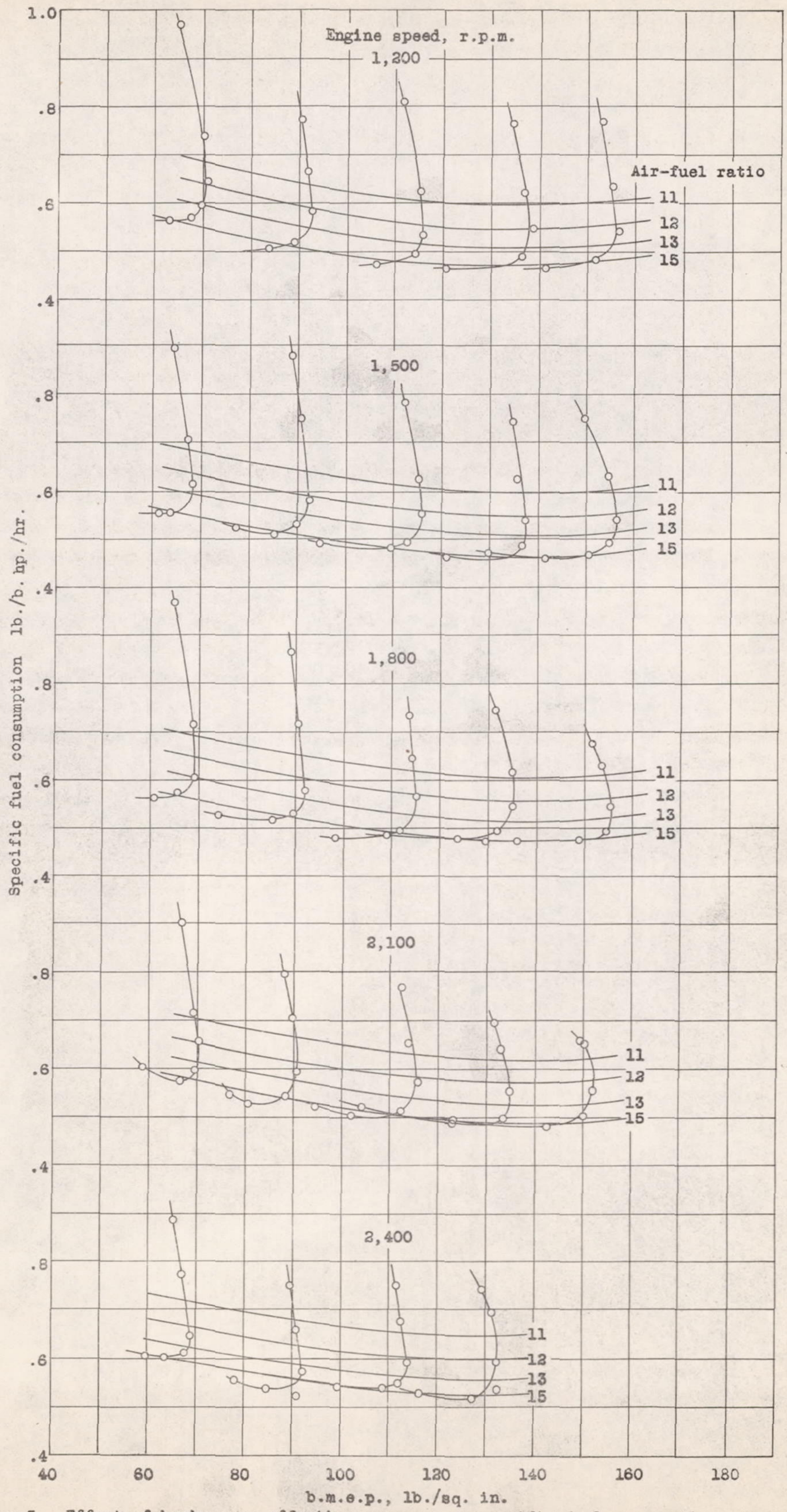


Figure 5.- Effect of brake mean effective pressure on specific fuel consumption at several throttle openings and engine speeds. Pratt and Whitney 1340-H cylinder.

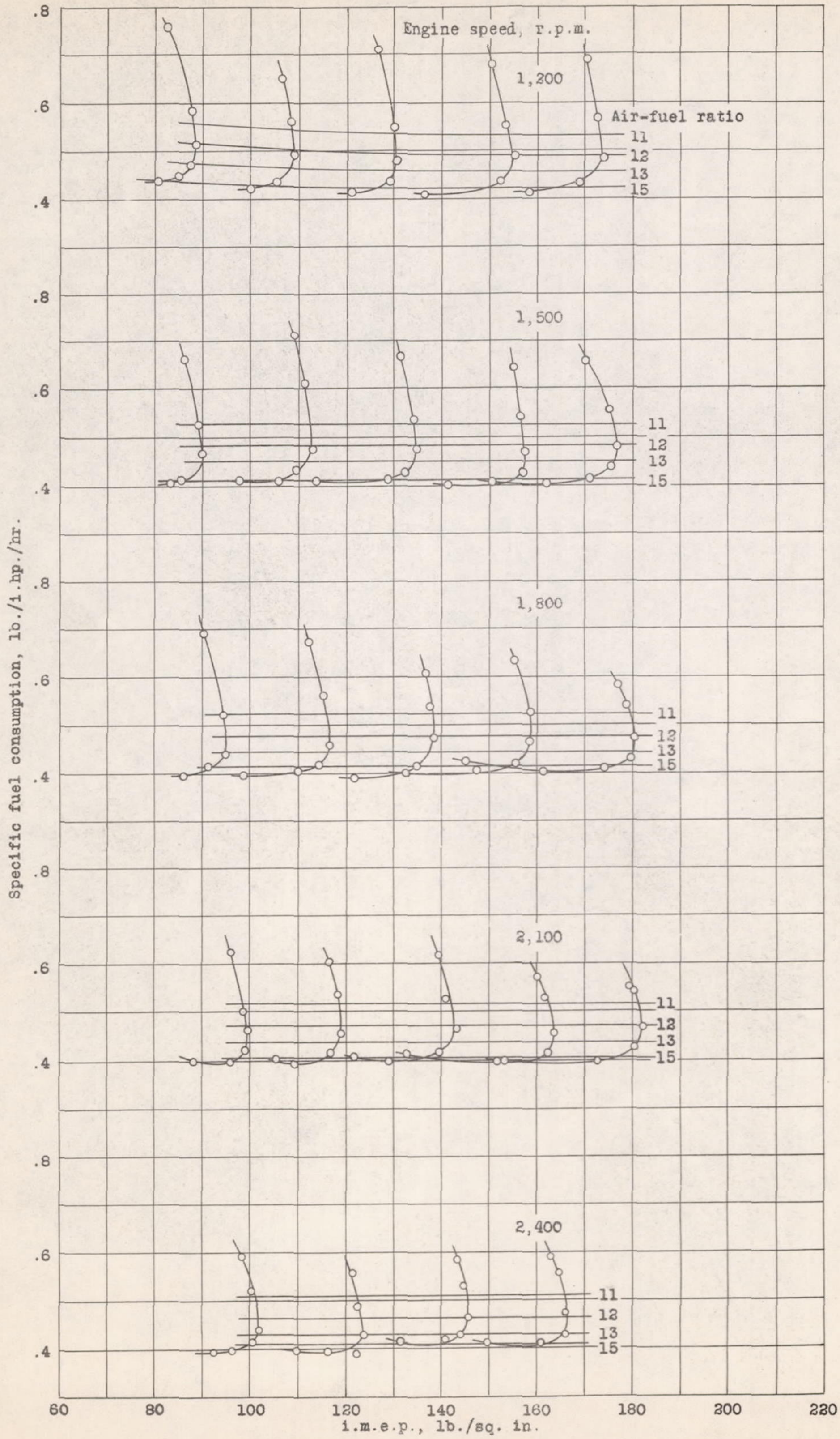


Figure 6.- Effect of indicated mean effective pressure on specific fuel consumption at several throttle openings and engine speeds. Pratt and Whitney 1340-H cylinder.

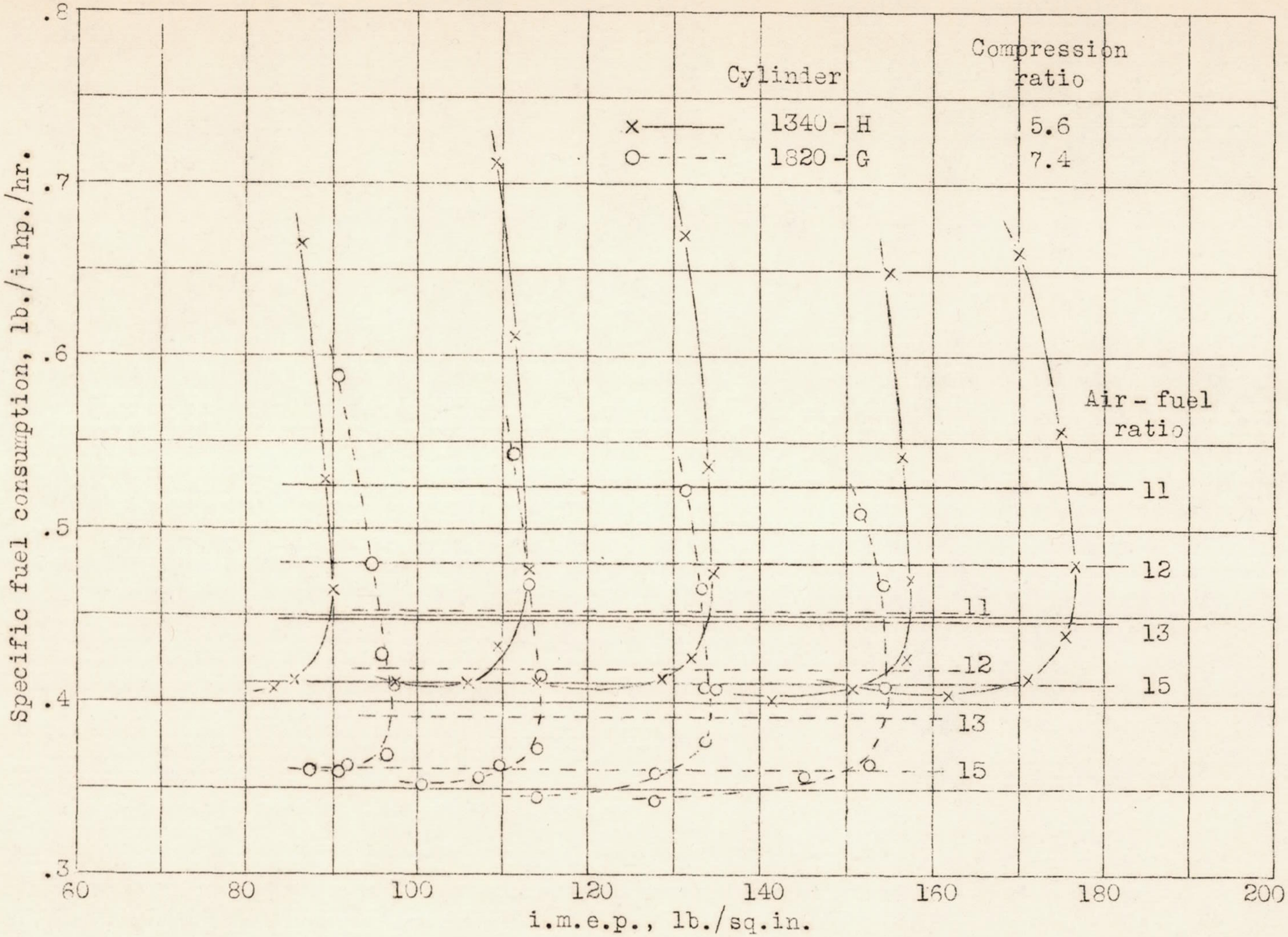


Figure 7.- Comparison of the effect of indicated mean effective pressure on specific fuel consumption at several throttle openings. Engine speed, 1,500 r.p.m.

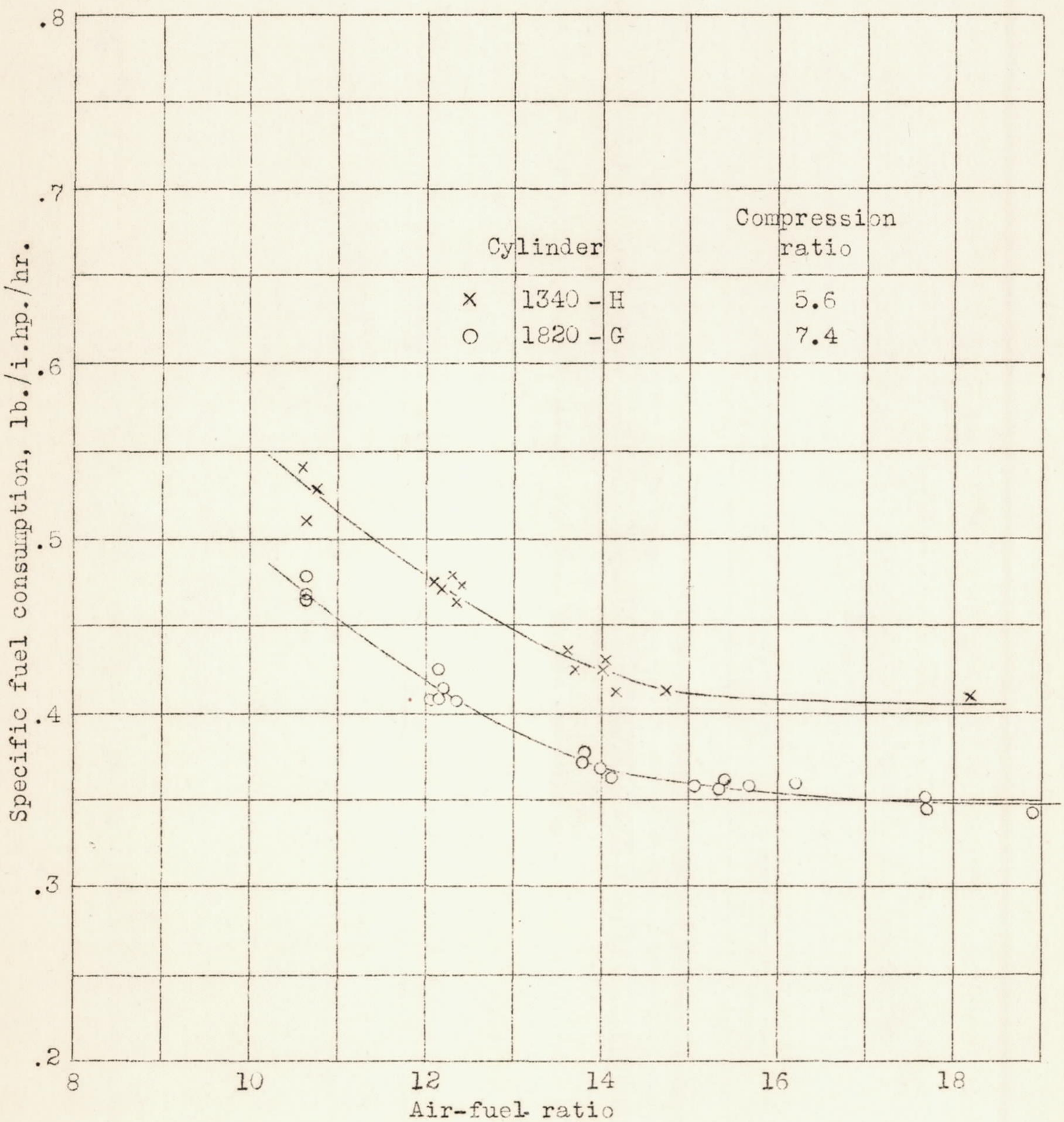


Figure 8.- Variation of indicated fuel consumption with air-fuel ratio at several throttle openings. Engine speed, 1,500 r.p.m.

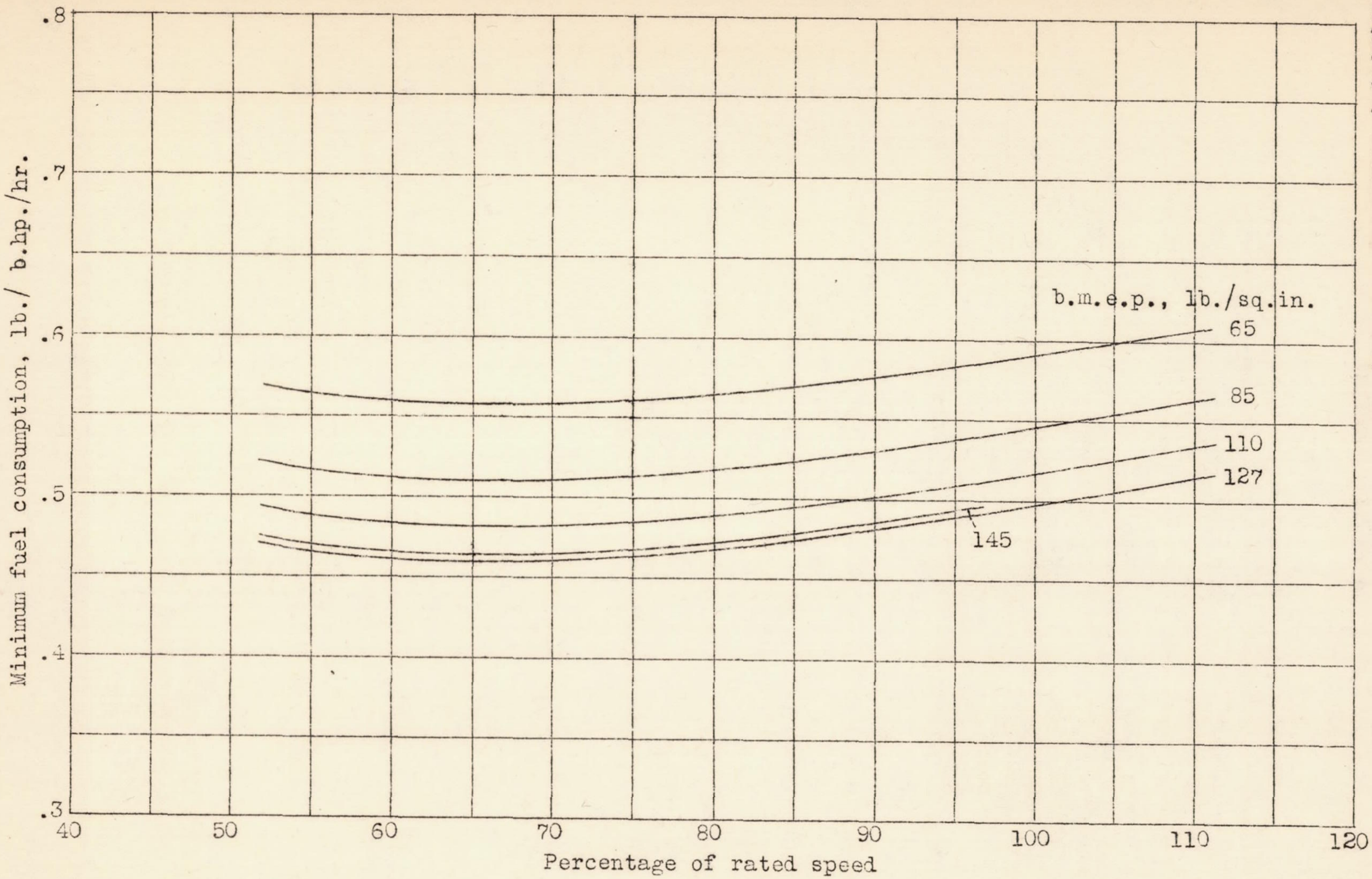


Figure 9.- Effect of engine speed on minimum fuel consumption of several torque values. Pratt and Whitney 1340-H cylinder.