REPORT No. 471

PERFORMANCE OF A FUEL-INJECTION SPARK-IGNITION ENGINE USING A HYDROGENATED SAFETY FUEL

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SUMMARY

This report presents the performance of a single-cylinder test engine using a hydrogenated safety fuel. The safety fuel has a flash point of 125° F. (Cleveland open-cup method), which is high enough to remove most of the fire hazard, and an octane number of 95, which permits higher compression ratios to be used than are permissible with most undoped gasolines. The fuel was injected into the engine cylinder, except for a few comparative runs with gasoline, when a carburetor was used. The tests were made with compression ratios of 5.85 and 7.0, valve timings giving 30° and 130° overlap, inlet pressures from atmospheric to 6 inches of mercury boost, and engine speeds from 1,250 to 2,200 r.p.m. Under similar conditions the power obtained with the safety fuel was the same as that obtained with gasoline, whereas the fuel consumption was from 5 to 10 percent higher. With a compression ratio of 7.0, a valve overlap of 130 crankshaft degrees, and a boost pressure of 2 inches of mercury, the safety fuel gave a brake mean effective pressure of 175 pounds per square inch with a fuel consumption of 0.50 pound per brake horsepower hour.

INTRODUCTION

The importance of replacing gasoline with a fuel that would reduce or eliminate the fire hazard in aircraft has long been recognized. The use of gasoline is a fire hazard because inflammable vapors are given off in nearly all climates and seasons. Aviation gasoline has a flash point of about −30° F. Those acquainted with the problem of fire prevention in airplanes agree that the highly inflammable gasoline should be replaced by a fuel having a higher flash point, preferably over 105° F. as determined by the closed-cup method.

One of the advantages of the compression-ignition engine is that it uses a fuel of such a high flash point (approximately 175° F.) that no inflammable vapors are given off even in the warmest climate. Aircraft-engine operators, however, have considered the advantage of reducing the fire hazard by using compression-ignition engines to be insufficient to offset the disadvantage of the decreased power per unit of weight and displacement obtained with this type engine.

In France, Sabatier has reported an investigation on the use of fuels having flash points of 100° F. and 77° F., obtained from coal-tar and petroleum derivatives, respectively (reference 1). The commercial use of these fuels was restricted, if not entirely prevented, by their poor performance as compared with gasoline: the power was reduced, the fuel consumption was increased, starting was difficult, and increased heating of the carburetor was necessary.

The National Advisory Committee for Aeronautics has conducted tests with safety fuels manufactured by the hydrogenation process (reference 2). Because of the low volatility of the fuel it has been injected into the engine cylinder instead of being introduced through a carburetor. The first fuel investigated had a flash point of 137° F. as determined by the Cleveland open-cup method. The full-throttle power obtained with this fuel was lower than with gasoline, and the fuel consumption was considerably higher. The second fuel tested had a flash point of 115° F. With this fuel the power was as high as that with gasoline, but the fuel consumption was from 25 to 30 percent higher (reference 3).

The results obtained from an investigation conducted with a third fuel, which had a flash point of 125° F., are presented in this report. The object of this investigation was to determine the performance obtained with a spark-ignition engine when operating with a hydrogenated safety fuel injected into the engine cylinder. As a basis for comparison the performance was obtained for several comparable conditions with gasoline. The tests were conducted at Langley Field, Va., in December 1932 and January 1933.

APPARATUS AND METHOD

Figure 1 shows the set-up of the test equipment. A single-cylinder 4-stroke-cycle water-cooled test engine of 5¼-inch bore and 6-inch stroke was used. The engine could be operated with either a fuel-injection system or a carburetor. A commercial fuel-injection pump was driven from the engine crankshaft through a reduction gear which permitted the phase of the injection to be changed at will. A spring-loaded automatic injection valve and a multi-orifice nozzle of

N.A.C.A. design were used (fig. 2). When the injection system was used the carburetor was left in place and the throttle valves were used to control the air supply for starting.

The engine was directly connected to an electric dynamometer. A small weighing tank suspended from a sensitive beam balance was used to measure the fuel during a run, the length of the run being the time required to consume one-half pound of fuel. The engine coolant was piped to a radiator, which was cooled by a water spray when necessary. The small volume of liquid necessary to fill this cooling system made it feasible to use Prestone and operate at high coolant temperatures when desired. Temperatures up to 280° F. at the engine outlet could be obtained.

This engine has a pent-roof form of combustion chamber, with two inlet and two exhaust valves (fig. 3). The inlet-valve ports are $1\frac{13}{16}$ inches and the exhaust-valve ports $1\frac{5}{8}$ inches in diameter. There are five tapped holes in the head, permitting some choice in locating the spark plugs and injection valve. The injection valve was located between the exhaust valves, and directed the spray horizontally across the combustion chamber toward the inlet valves. This injection-valve location gave the best performance; however, the performance was only slightly better than with the valve located in the center hole. Two spark plugs were located in opposite sides of the combustion chamber.

Two different pistons and two different sets of valve cams were used. These pistons gave compression ratios of 5.85 and 7.0. The set of cams that gave normal valve timing caused the inlet valves to open 15° before top center and close 55° after bottom center, while the exhaust valves opened 55° before bottom center and closed 15° after top center. The other set of cams did not change the events at the bottom of the stroke, except to advance the inlet closing 10°, but caused the inlet valves to open 70° before top center and the exhaust valves to close 60° after top center. This valve timing results in an overlap of the open periods of the exhaust and inlet valves of 130°, giving improved scavenging of the clearance volume, particularly when some boosting is used (reference 4). Figure 4 shows the amount of valve opening during the period of overlap. A separately driven Roots blower was connected to the inlet system through a large surge tank placed near the carburetor.

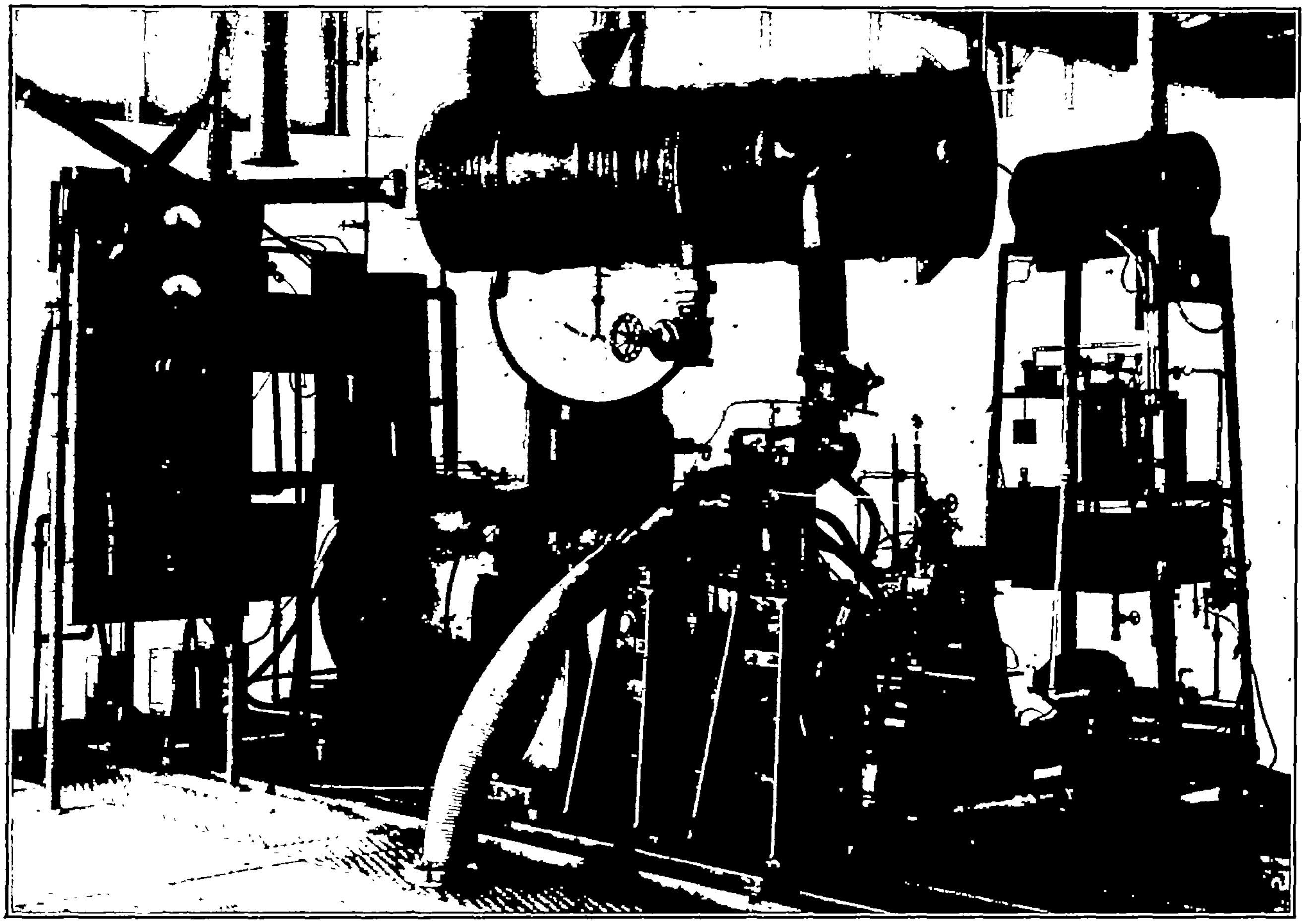

FIGURE 1.—Set-up of test equipment.

The results obtained from runs in which the length of the exhaust pipe was varied caused the adoption of a length of 2 feet for these tests. Shorter pipes caused lower torque at all speeds and more variation of torque over the useful speed range, unless boosting was used. The length of inlet pipe to the point of attachment at the surge tank was 2½ feet, but this length was not critical. With a large valve overlap the effect of pressure waves in the exhaust and inlet pipes becomes negligible when a supercharging pressure of several inches of mercury is used.

The engine performance with the hydrogenated safety fuel using fuel injection was obtained for speeds

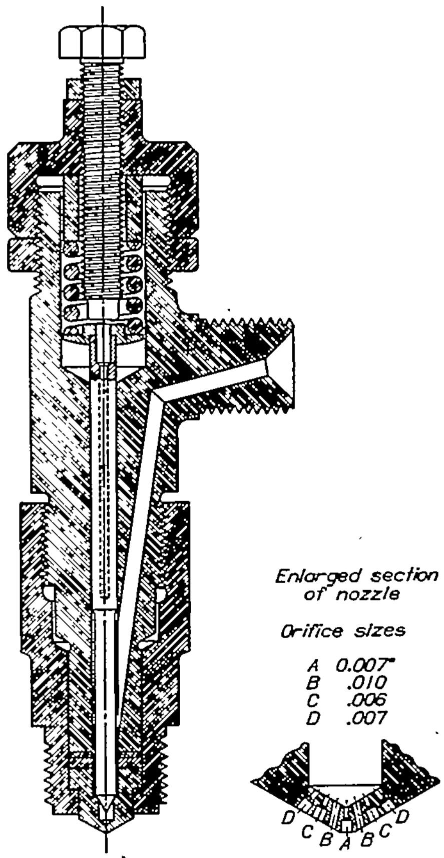

FIGURE 2.—Fuel-injection valve and nozzle.

from 1,250 to 2,200 r.p.m., compression ratios of 5.85 and 7.0, valve timings giving 30° overlap and 130° overlap, and boost pressures up to 6 inches of mercury. A sufficient number of these tests using fuel injection were repeated with aviation gasoline as fuel to furnish a reliable comparison of the safety fuel and the gasoline. A few runs were made with gasoline using the carburetor.

The procedure for each test condition was to make three or four full-throttle runs using fuel quantities that gave mixtures ranging from one richer than necessary for maximum power to a very lean one. The engine torque and fuel consumption were measured for each run. The brake power was corrected to an atmospheric pressure of 29.92 inches of mercury and a temperature of 59° F. on the assumption that it varied directly as the pressure and inversely as the square root of the absolute temperature. No correction was made for humidity or for the power required to drive

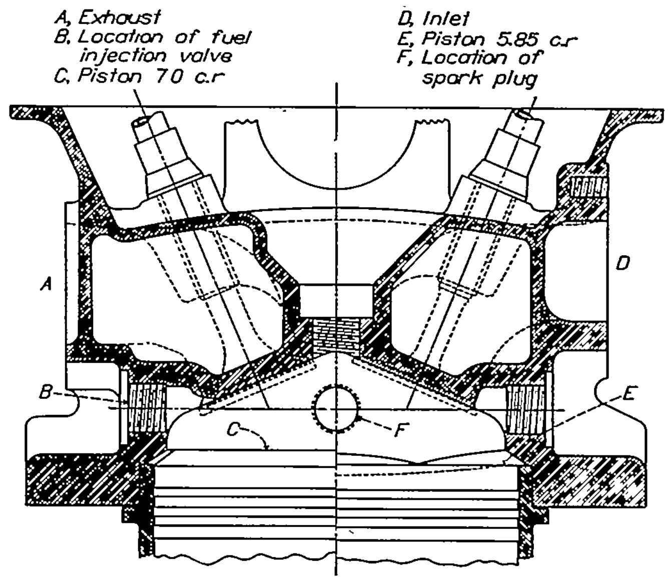

FIGURE 3.—Combustion chamber form.

the supercharger. The correction for power required to drive the supercharger, when used, would not be over 3 percent of the engine power at 6 inches of mercury boost pressure.

Some additional data were obtained with special equipment. Where maximum cylinder pressures were

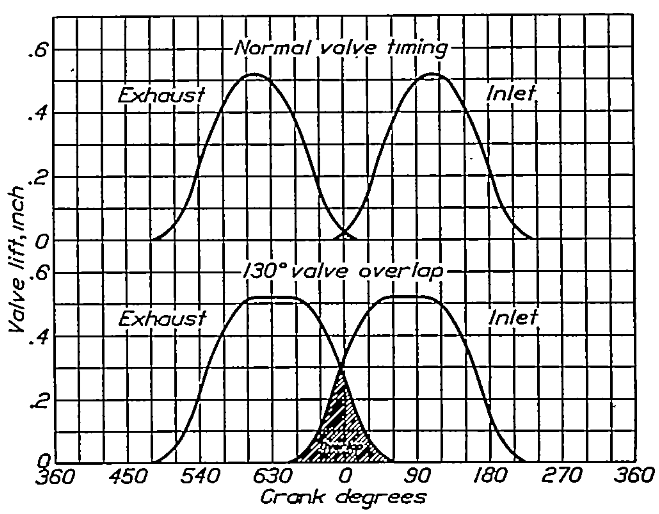

FIGURE 4.—Valve motion with normal valve timing and with 130° overlap.

taken a trapped-pressure valve was used. A reduced back pressure on the exhaust of the engine was obtained for a few tests by discharging the exhaust into a large tank, the outlet of which was connected to the section side of a supercharger. With the same equipment the outlet of the tank was throttled to produce increased exhaust back pressure. Data on the characteristics of the fuel-injection system were obtained with the N.A.C.A. rate-of-discharge apparatus, which is described in reference 5.

FUELS

Distillation curves for the gasoline and the hydrogenated safety fuel are given in figure 5. The flash point of gasoline is below ordinary atmospheric temperatures even in winter, while that of the safety fuel (125° F. by the Cleveland open-cup method or 106° F. by the Abel closed-cup method) is well above the highest operating temperatures usually encountered. Besides reduced fire hazard, the hydrogenated safety fuel has excellent antidetonating qualities, so that the fuel may be used at high compression ratios without the use of fuel dopes such as tetraethyl lead. The hydrogenated safety fuel has an octane number of 95 as determined by the manufacturer using a series 30 Ethyl Gasoline Corporation test engine operated at a speed of 600 r.p.m. and with a coolant temperature of 300° F. Sufficient ethyl fluid was added to the aviation gasoline to prevent detonation under any of the test conditions, thereby placing the engine performance with the two fuels on a comparative basis that is independent of antiknock characteristics. A study of the behavior of the hydrogenated fuel at low temperatures showed satisfactory characteristics. At a temperature of −25° F. a few solid particles appeared in the fuel, but at temperatures as low as −100° F. there was no tendency for all of the fuel to solidify.

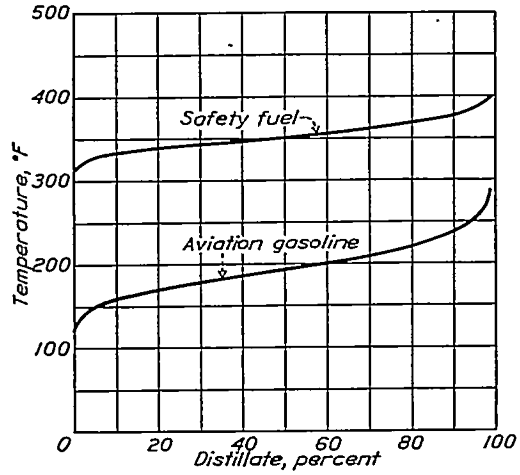

FIGURE 5.—Distillation curves for the gasoline and safety fuel.

RESULTS AND DISCUSSION

EFFECT ON BRAKE MEAN EFFECTIVE PRESSURE AND FUEL CONSUMPTION

Compression ratio, scavenging, fuel, and fuel system.—Figure 6 presents the comparative performance obtained at a compression ratio of 5.85 with gasoline and safety fuel and with the fuel-injection system and the carburetor. The performance curves with gasoline show that the maximum brake mean effective pressure obtained with the fuel-injection system is greater than that obtained with the carburetor. The difference in brake mean effective pressure decreases as the quantity of fuel per cycle is decreased, indicating that the volumetric efficiency was probably slightly higher with the use of injection into the cylinder than with the use of the carburetor.

In most of these tests no air measurements were made because the use of the air-measuring system caused a small reduction in power. A few runs were made, however, in which measurements of air consumption were obtained. The first set of these air measurements was made to determine the difference in volumetric efficiency obtained when operating with the fuel-injection system and when operating with the carburetor. The results showed that the volumetric efficiency was from 1 to 3 percent higher with the injection system than with the carburetor, depending on the engine speed. As the carburetor was left in place when operating with the injection system, any gain in volumetric efficiency must be attributed to the difference between external and internal carburetion.

A comparison of the brake mean effective pressure and the economy obtained with safety fuel and gasoline when operating with the fuel-injection system shows that the maximum power is the same for the two fuels, and that the fuel consumption is 5 to 10 percent lower with gasoline.

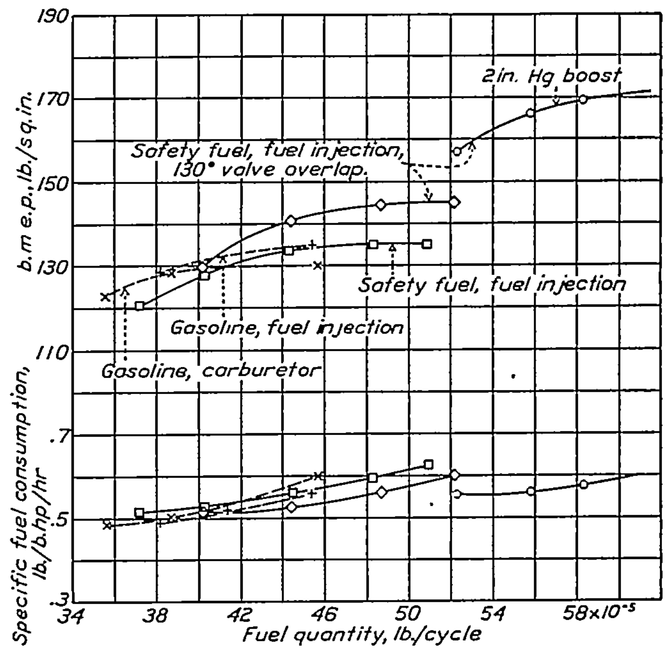

FIGURE 6.—B.m.e.p. and fuel consumption obtained when operating at a compression ratio of 5.85 and an engine speed of 1,750 r.p.m. with gasoline and with safety fuel.

There were more exhaust odors and fumes present when operating with the safety fuel than when operating with the gasoline. However, the exhaust fumes were not so noticeable that the operating conditions could be considered disagreeable or unsatisfactory.

The results that have been discussed so far are for standard valve-timing conditions. In figure 6 there are also shown performance curves for valve timing giving 130 crankshaft degrees overlap. When operating with atmospheric pressure at the intake with this valve overlap a maximum brake mean effective pressure of 145 pounds per square inch is obtained and when

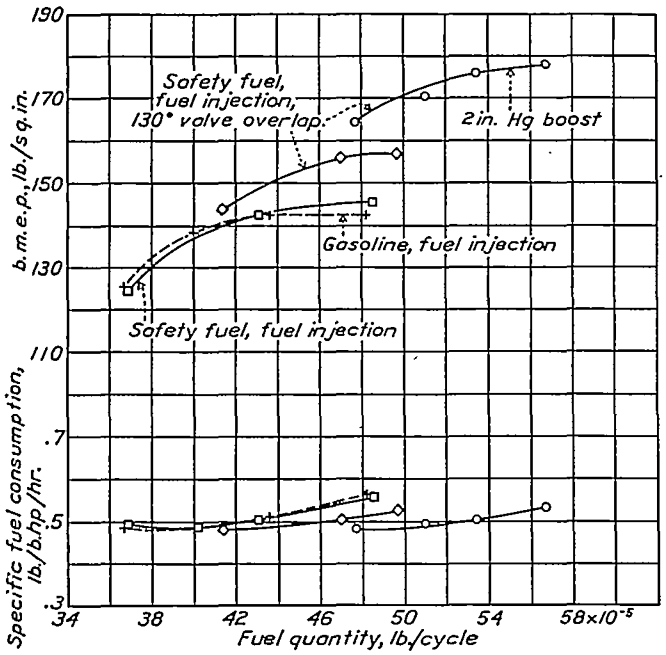

FIGURE 7.—B.m.e.p. and fuel consumption obtained when operating at a compression ratio of 7.0 and an engine speed of 1,750 r.p.m. with gasoline and with safety fuel.

operating with 2 inches of mercury boost pressure a maximum brake mean effective pressure of 170 pounds per square inch is obtained. The large increase in maximum brake mean effective pressure obtained with a small boost pressure is caused principally by the scavenging of the clearance volume. In these tests with safety fuel at a compression ratio of 5.85 there was a small increase in fuel consumption for the scavenged condition, whereas in earlier tests with gasoline on another engine there was a slight decrease in fuel consumption when scavenging (reference 4).

The curves in figure 7 for a compression ratio of 7.0 and no boost pressure show that with safety fuel the brake mean effective pressure is approximately 10 pounds per square inch greater and the specific fuel consumption is 7 to 8 percent lower than for the 5.85 compression ratio. The compression ratio could probably be further increased without the addition of fuel dope to the safety fuel, for there was no indication of detonation in these tests. When operating with a valve overlap of 130 crankshaft degrees and 2 inches of mercury boost, a brake mean effective pressure of 175 pounds per square inch was obtained with a fuel consumption of 0.50 pound per brake horsepower hour. The specific fuel consumption was the same for the scavenged condition as for the condition with no scavenging.

In the comparison of the curves for these two compression ratios it should be borne in mind that a constant fuel quantity per cycle does not mean a constant mixture ratio, because the volume of air inducted per cycle depends on the valve overlap and the boost pressure. A charge that is excessively lean may not give as much power as a smaller charge of about the right proportion of fuel and air for maximum power.

Boost pressure.—Figure 8 shows the comparative brake mean effective pressure and fuel consumption obtained when operating at compression ratios of 5.85 and 7.0 with boost pressures varying from 0 to 6 inches of mercury. Increasing the compression ratio from 5.85 to 7.0 resulted in a reduction of fuel consumption of 10 to 13 percent over this range of boost pressures and an increase in power of 8 percent at no boost pressure and 3 percent at 6 inches of mercury boost pressure. It might be well to mention here that the uni-

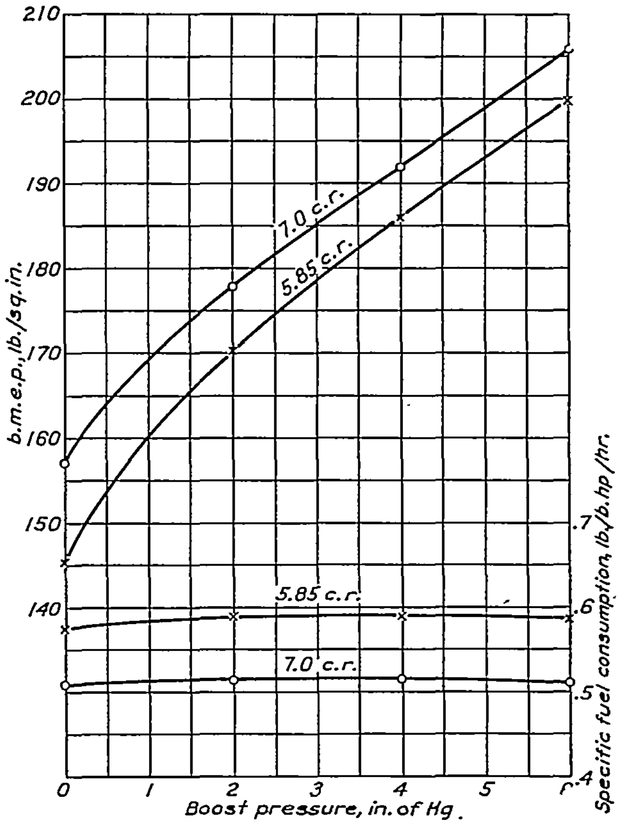

FIGURE 8.—Effect of boost pressure on b.m.e.p. and fuel consumption. Safety fuel; fuel injection; 130° valve overlap; 1,750 r.p.m.

versal test engine, an engine of practically the same design, has been operated with no boost pressure at a compression ratio of 9.0 with this fuel. In the tests with the universal test engine an increase in maximum brake mean effective pressure of 12 pounds per square inch was obtained by increasing the compression ratio from 7.0 to 9.0, even though, to avoid detonation, the spark was retarded 16 crankshaft degrees from the optimum spark setting at a compression ratio of 7.0.

Speed.—Figure 9 shows the fuel consumption and the power obtained at speeds from 1,250 r.p.m. to 2,200 r.p.m. The maximum brake mean effective pressure on this engine is obtained at speeds from 1,700 to 1,900 r.p.m. and there is very little falling off in the brake mean effective pressure at speeds up to 2,200 r.p.m.

Table I is included for a convenient comparision of the power and economy obtained with gasoline and safety fuel and the friction mean effective pressure at each compression ratio for several speeds with normal valve timing. The values given in this table have been taken for the lowest fuel quantities per cycle at which the maximum brake mean effective pressure is obtained. These tabulated results show that the brake mean effective pressure obtained with safety fuel is equal to that obtained with gasoline and that the fuel consumption obtained with safety fuel is only from 5 to 10 percent higher than that with gasoline. As the calculated lower heating value of the safety fuel of 17,560 B.t.u. per pound is 7 to 8 percent lower than that of gasoline, the thermal efficiency for the two fuels would be practically the same. It is believed that the fuel economy obtained with safety fuel as compared with that obtained with gasoline cannot be appreciably improved.

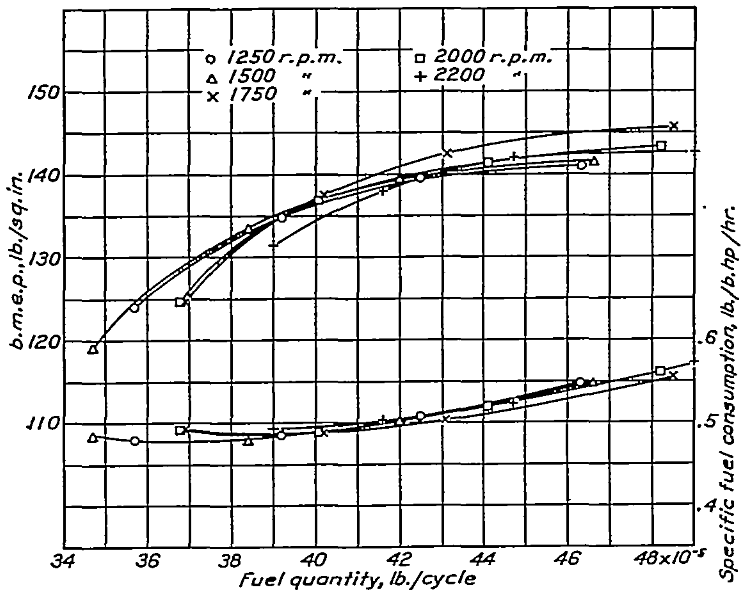

FIGURE 9.—B.m.e.p. and fuel consumption obtained for various engine speeds. 7.0 compression ratio; safety fuel; fuel injection; normal valve timing; no boost pressure.

Coolant temperature.—The tests so far discussed were conducted at coolant temperatures of 150° F. Other tests made at coolant temperatures of 200, 250, and 280° F. showed that no improvement in the brake mean effective pressure or the fuel consumption could be obtained by operating at high coolant temperatures. Some time ago, when operating the universal test engine with another hydrogenated safety fuel, a large improvement in the economy was obtained by operating at high coolant temperatures (reference 3). In those tests the fuel consumption was high at coolant temperatures of 150° F., whereas in the present tests the fuel consumption was normal. High coolant temperatures apparently result in improved economy where the economy is poor at low temperatures, but increasing the coolant temperatures when the economy is already good results in no improvement. When operating with safety fuel, low coolant temperatures (150° F.) are to be preferred because high coolant temperatures impair the antiknock properties of the safety fuel. From the results obtained on the universal test engine with a different safety fuel it is believed that since this safety fuel can be used at 7.0 compression ratio with coolant temperatures of 250° F., it can be used at a compression ratio of 8.5 with coolant temperatures of 150° F.

IDLING AND STARTING

The idling of the engine with normal valve timing when operating with safety fuel is entirely satisfactory. When operating with a large valve overlap the idling is poor with the usual throttle arrangement, because some of the exhaust gases flow into the intake manifold whenever the engine is throttled. The idling with a large valve overlap will be satisfactory if the throttle is placed close to the inlet valve so as to reduce to a minimum the volume between the throttle valve and the intake valves. The universal test engine operating with a valve overlap of 112° would idle at an engine speed of 150 r.p.m. when the throttle valve was close to the intake valve.

Starting with safety fuel was difficult when the engine was cold; that is, when it had been standing overnight at a temperature of 50°–60° F. It has been started cold when motoring at 700 r.p.m. with a compression ratio of 5.85, but starting under these conditions is not satisfactory. In later tests satisfactory starting was obtained by injecting a small quantity of gasoline into the intake manifold while the engine was being motored at speeds as low as 120 r.p.m. and while safety fuel was being injected into the cylinder. Immediately after the engine was started on gasoline it would continue to run on safety fuel. This method of starting requires only the addition of a small gasoline tank, as the priming system is identical with the present priming system used on aircraft engines. On engines equipped with air starters the fuel might be mixed with the starting air just before it is inducted. Both of these methods would require the use of two fuels, but the supply of gasoline carried for starting would be very small.

The engine could be started on safety fuel if the intake air, the fuel, or both were heated. The curve in figure 10 shows approximately the minimum air temperature at which the engine will start with different fuel temperatures. The engine would start consistently at a speed of 350 r.p.m. with air and fuel temperatures as shown in this figure.

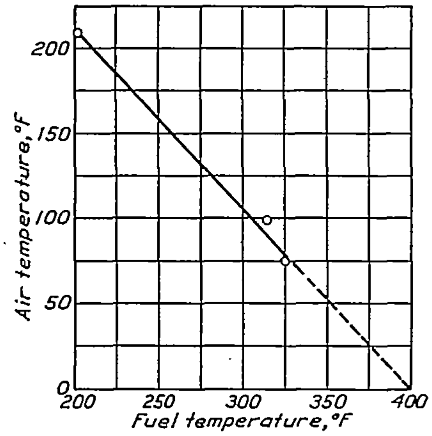

FIGURE 10.—Minimum fuel and air temperatures required for starting at a compression ratio of 7.0 when motored at 350 r.p.m.

THE INJECTION SYSTEM

In these tests the fuel consumption and the brake mean effective pressure were not critically sensitive to the timing of the start of the injection period, the duration of the injection period, the injection pressure, or the valve-opening pressure. The effect of the timing of the start of the injection period on the brake mean effective pressure is shown by the curve in figure 11. With the injection system used in these tests, the best results were obtained when the start of injection was from 70 to 90 crankshaft degrees after top center on the suction stroke.

The curves in figures 12 and 13 show the characteristics of the injection system used. For these tests the pump setting gave a fuel quantity of approximately 0.00058 pound per cycle at an engine speed of 1,750 r.p.m. It will be noted from figure 7 that this is the fuel quantity giving maximum power with a compression ratio of 7.0 and 2 inches of mercury boost pressure. Although the rate of injection shown in figure 12 gave the best performance of the several rates tried, it is believed that some deviation from these rates will not appreciably impair the performance. The length of the injection period increased from 74 to 93 crankshaft degrees with an increase in pump speed from 750 to 1,100 r.p.m. In other tests on the same engine with the injection period varying from 150 to 200 crankshaft degrees the fuel consumption was higher and the power lower. An injection period of approximately 30 crankshaft degrees has been tried on the universal test engine. The results of all these tests indicated that better economy and power could be obtained when the length of the injection period was from approximately 60 to 90 crankshaft degrees.

All data submitted in this report were obtained with a commercial fuel pump, an injection-valve opening pressure of 2,000 pounds per square inch, and injection pressures as shown in figure 13. A few tests, however, have been conducted with other fuel pumps and with gasoline as a fuel. In some of these tests a valve-opening pressure of 800 pounds per square inch and an injection pressure of 1,200 pounds per square inch were used. The results obtained with these low injection pressures were, for practically all conditions, equal to those obtained with high injection pressures. It is believed that an injection system operating with injection pressures as low as 500 pounds per square inch or lower would be satisfactory.

MECHANICAL CONSIDERATIONS

The use of a large valve overlap requires a consideration of several mechanical problems. The overlap must be sufficiently large to give the desired scavenging at sea level with a pressure difference across the valves of from 2 to 5 inches of mercury, but not so large that an appreciable amount of the air is wasted at moderate or high altitudes, when the pressure difference across the valves may be 10 to 15 inches of mercury. The pressure difference across the valves on a supercharged engine increases with the altitude of operation because the pressure at the intake is usually kept constant up to some definite altitude, whereas the atmospheric pressure at the exhaust decreases with altitude. Under these conditions the importance of scavenging the clearance volume decreases with altitude. For instance, the gain obtained by scavenging the clearance volume of an engine at 18,000 feet is only 50 percent of that obtained at sea level because the reduced exhaust pressure permits more of the exhaust gas to escape.

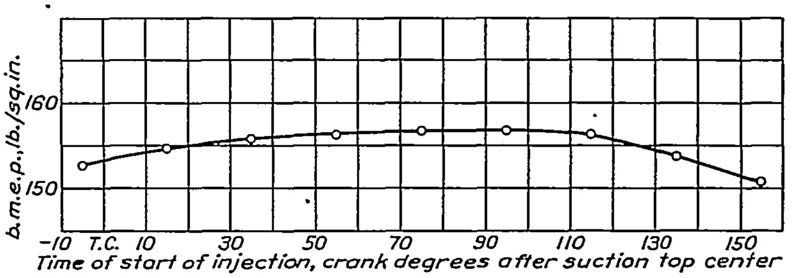

FIGURE 11.—Effect of start of injection period on b.m.e.p.

Exhaust back pressure.—A few tests were made to determine the effect of reduced exhaust back pressures on the volumetric efficiency and the power when operating with a large valve overlap. The results of these tests showed that the air supplied to an engine operating with a valve overlap of 130 crankshaft degrees corresponds to volumetric efficiencies of 110, 117, and 122 percent at engine speeds of 2,200, 1,800, and

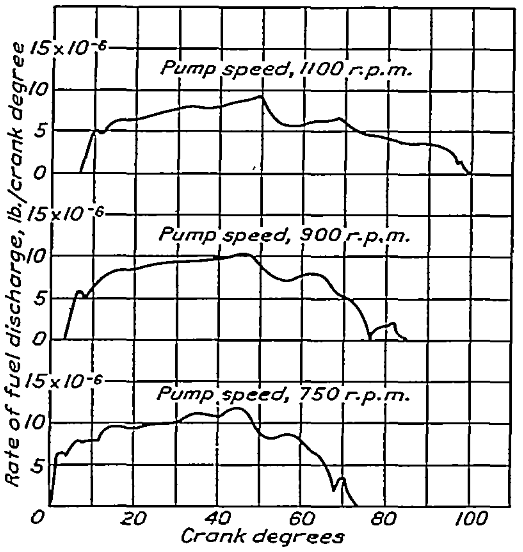

FIGURE 12.—Effect of pump speed on the length of the injection period and the rate of discharge. 12-millimeter plunger diameter; 0.125-inch tube diameter; 32-inch tube length; 2,000 pounds per square inch valve-opening pressure.

1,500 r.p.m., respectively, when operating with atmospheric pressure at the intake and a pressure 8 inches of mercury less than atmospheric at the exhaust. For these conditions the volumetric efficiency increased at

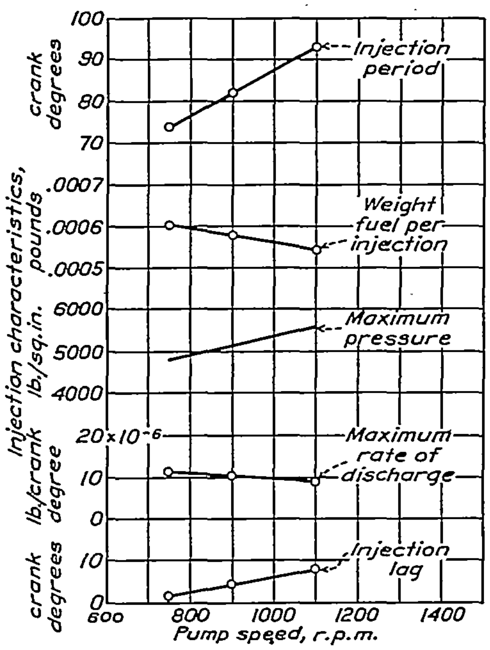

FIGURE 13.—Effect of pump speed on injection characteristics. 12-millimeter plunger diameter; 0.125-inch tube diameter; 32-inch tube length; 2,000 pounds per square inch valve-opening pressure.

a greater rate than the power, indicating that a large amount of the fresh air escaped during the scavenging process. With the exhaust pressure from 3 to 5 inches of mercury lower than the intake, the increase in volumetric efficiency was practically equal to the increase in power, indicating that very little of the fresh air was wasted.

Some tests were also made to determine the effect of exhaust back pressure of 3½ inches of mercury on the maximum brake mean effective pressure when operating with a valve overlap of 130 crankshaft degrees and intake pressures varying from 0 to 10 inches of mercury boost. The results of these tests are shown in figure 14. Note that when the intake pressure is 7 inches of mercury and the exhaust pressure is increased from 0 to 3½ inches of mercury the power decreases 4 percent as compared to 20 percent when the

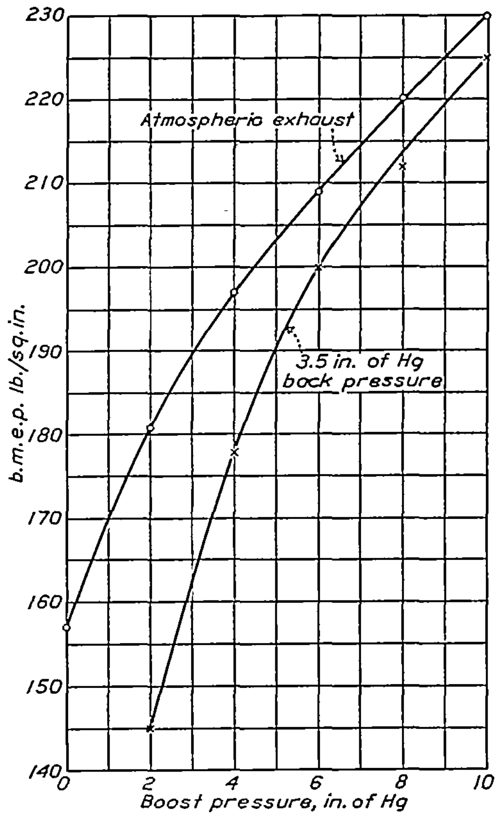

FIGURE 14.—Effect of 3½ inches of mercury exhaust back pressure on the power with the boost pressure varying from 0 to 10 inches of mercury. 1,750 r.p.m.; 7.0 compression ratio; safety fuel.

intake pressure is 2 inches. It is very important that the intake pressure be greater than the exhaust pressure when operating with a valve overlap. If the exhaust back pressure is higher than the inlet pressure the gas flow may be reversed during part of the cycle, and exhaust gas may fill part of the displacement volume and induction pipes.

Air and fuel control.—When applying fuel injection to a spark-ignition engine for aircraft service the air throttle and fuel-quantity control should be interconnected so that the engine will receive air and fuel in the proper proportions over the entire range of loads and speeds. This problem may require the working out of a complicated linkage, particularly if an accurate proportioning of fuel and air is attempted.

Spark setting.—The maximum cylinder pressures were determined for the most severe operating condition tried and are plotted in figure 15. With the spark at 30° before top center, the setting for maximum power, the maximum pressure recorded was 890 pounds per square inch. This pressure could be reduced to 800 pounds per square inch by setting the spark for 22° before top center—a reduction of 10 percent in maximum cylinder pressure with a sacrifice of 2½ percent in brake mean effective pressure.

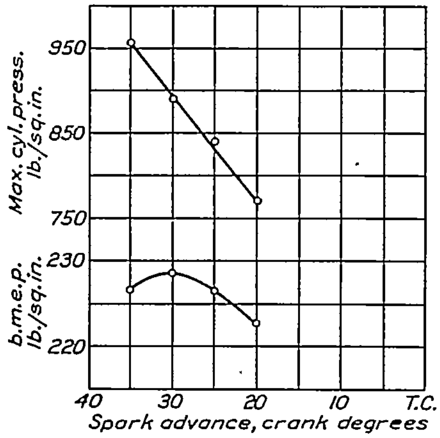

FIGURE 15.—Effect of spark advance on b.m.e.p. and maximum cylinder pressure. 1,750 r.p.m.; 7.0 compression ratio; safety fuel; 10 inches of Hg boost pressure; 3.5 inches of Hg back pressure.

CONCLUSIONS

1. The hydrogenated safety fuel manufactured primarily to eliminate fire hazard in aircraft gives a maximum brake mean effective pressure equal to that with gasoline when using the fuel-injection system, and the fuel consumption is from 5 to 10 percent higher.

2. The high antiknock value of the hydrogenated safety fuel permits a higher compression ratio to be used than can be used with most gasolines without the addition of fuel dope, thus improving both the power and the economy.

3. At present the hydrogenated safety fuel can be used to best advantage by employing fuel injection, and the use of fuel injection makes scavenging by the use of large valve overlap and moderate supercharging feasible.

4. The results of the test with 130 crankshaft degrees overlap show that so long as the pressure difference between the intake and the exhaust is not greater than 5 inches of mercury practically no air is wasted in the scavenging process.

5. Additional measures must be taken to insure starting when using safety fuel. Priming with gasoline is a simple and practical way of solving this problem.

LANGLEY MEMORIAL AERONAUTICAL LABORATORY,
NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS,
LANGLEY FIELD, VA., *June 13, 1933.*

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TABLE I

ENGINE PERFORMANCE WITH GASOLINE AND HYDROGENATED SAFETY FUEL

(Best Setting for Maximum Power; Normal Valve Timing; No Boost)

Engine speed, r.p.m.	5.85 compression ratio							7.0 compression ratio				
	Carburetor with gasoline		Fuel injection				f.m.e.p., lb./sq. in.	Fuel injection				f.m.e.p., lb./sq. in.
			Gasoline		Safety fuel			Gasoline		Safety fuel		
	b.m.e.p., lb./sq. in.	s.f.c., lb./b.hp./hr.	b.m.e.p., lb./sq. in.	s.f.c., lb./b.hp./hr.	b.m.e.p., lb./sq. in.	s.f.c., lb./b.hp./hr.		b.m.e.p., lb./sq. in.	s.f.c., lb./b.hp./hr.	b.m.e.p., lb./sq. in.	s.f.c., lb./b.hp./hr.	
1,250	128	0.485	130	0.550	131	0.575	17.8	139	0.485	140.5	0.535	18.1
1,500	129	.490	132	.550	131	.570	21.7	141	.500	141.5	.535	22.4
1,750	129	.515	133	.530	135 [1] 132.5	.595 .550	25.4	142.5	.510	145.5	.540	26.5
2,000	129	.490	132	.530	132	.560	28.9	141	.525	143.0	.550	32.4
2,200	127	.490	132	.530	----------	----------	33.9	139	.520	142.5	.550	37.4

[1] 2 percent less than maximum power.