REPORT No. 243

A PRELIMINARY STUDY OF FUEL INJECTION AND COMPRESSION IGNITION AS APPLIED TO AN AIRCRAFT ENGINE CYLINDER

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National Advisory Committee for Aeronautics

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SUMMARY

This report summarizes some results obtained with a single cylinder test engine at the Langley Field Laboratory of the National Advisory Committee for Aeronautics during a preliminary investigation of the problem of applying fuel injection and compression ignition to aircraft engines. For this work a standard Liberty engine cylinder was fitted with a high compression, 11.4 : 1 compression ratio, piston, and equipped with an airless injection system, including a primary fuel pump, an injection pump, and an automatic injection valve.

The results obtained during this investigation have indicated the possibility of applying airless injection and compression ignition to a cylinder of this size, 5-inch bore by 7-inch stroke, when operating at engine speeds as high as 1,850 R. P. M., although the unsuitability of the Liberty cylinder form of combustion chamber for compression ignition research probably accentuated the difficulties to be overcome. No difficulty was experienced in metering and injecting the small quantities of fuel required. A minimum specific fuel consumption with Diesel engine fuel oil of 0.30 pound per I. HP. hour was obtained when developing about 16 B. HP. at 1,730 R. P. M. Specific fuel consumption increased for higher loads at these speeds. A maximum power output of 29.7 B. HP. at 1,700 R. P. M. was obtained but could not be maintained for more than one-half minute due to piston failure. Mean effective pressures approaching standard aircraft engine practice could not be obtained, due in part, it was attributed, to the unsuitable form of the Liberty cylinder combustion chamber. Excessive maximum combustion pressures were encountered when developing only about 60 pounds B. M. E. P. at 1,700 R. P. M., and piston life was very short. The engine could be idled with regular firing at 400 R. P. M., but acceleration under load was not satisfactory, due probably to the fixed timing of injection during any particlar run.

INTRODUCTION

The application of airless fuel injection and compression ignition to aircraft engines is attractive because it enables the use of less inflammable and cheaper fuel, and gives, theoretically, better fuel economy at part loads than is obtained with the carburetted engine. While compression ignition eliminates the electric ignition system with its attendant troubles, it can not be said, as yet, that the fuel injection apparatus would be any less troublesome.

Since there appeared to be no published data available on the results of tests of an aircraft engine operating with fuel injection and compression ignition, this preliminary investigation was undertaken in order to learn something of the problem of airless, or solid, fuel injection as applied to aircraft engines.

The tests herein reported were conducted under the direction of Robertson Matthews.

DESCRIPTION OF APPARATUS AND METHODS

A standard Liberty cylinder, 5-inch bore and 7-inch stroke, without alterations, and having standard Liberty valve timing, was mounted on a single cylinder base and coupled to a 45/75 Sprague electric cradle dynamometer for the present tests. A photograph of the engine, showing the location of the injection valve and injection pump, is given in Figure 1.

To adapt the engine for use with compression ignition, it was necessary to remove the electric ignition system, to provide a drive shaft for the injection pump, mounted in the previous location of the ignition distributor head, and to substitute a high compression piston for the standard piston.

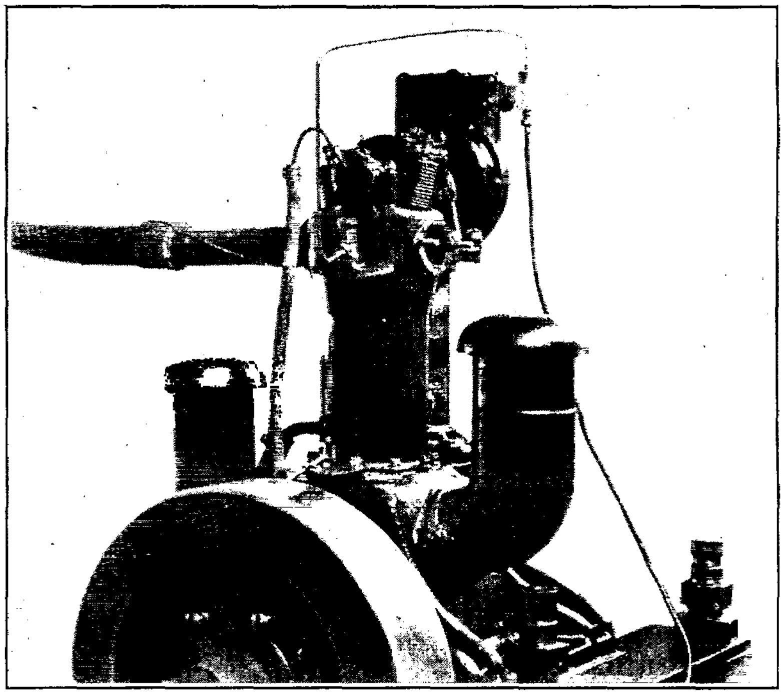

FIG. 1.—Single-cylinder Liberty engine equipped with airless injection system

Additional equipment for the engine included the primary fuel system, the injection pump, and the automatic injection valve.

PISTON DESIGN

In selecting the compression ratio for the Liberty cylinder operating on compression ignition, it was desired to use a ratio that would insure compression ignition when starting with a cold engine. At the same time, it was desired to use the lowest practical ratio in order to minimize the cranking effort required in starting, to keep the piston weight a minimum, to provide the largest space over the piston for the injection of fuel, and to keep combustion pressures as low as possible.

A fairly high compression ratio, 11.4:1, was finally chosen, being based only on a consideration of those conditions within the cylinder itself which might have an appreciable influence on the final compression pressure and temperature. Other factors which might logically have influenced the selection of the compression ratio, such as providing a compression ratio that would insure reliable engine operation under altitude conditions, or using a higher compression ratio than required from the standpoint of compression ignition only in order to provide a given temperature and pressure at the time the fuel would be injected, were not considered, as it was thought unnecessary at this stage of the investigation.

In order to secure the necessary compression pressure, it was required, with the Liberty form of combustion chamber, that the piston extend almost to the top of the cylinder. A conventional form of piston head, such as a dished head, could not be used without sacrificing compression pressure. Further, as the injection valve would be located in one of the spark-plug holes, which in the Liberty cylinder are off center, it appeared that the best distribution of fuel would be obtained by directing the fuel spray across the top of the piston. These conditions necessitated the adoption of an irregular form of head, having fluting or shallow grooving which would conform more or less to the spray shape, thus reducing deposition of fuel on the piston head. As the standard Liberty connecting rod was used, it was necessary

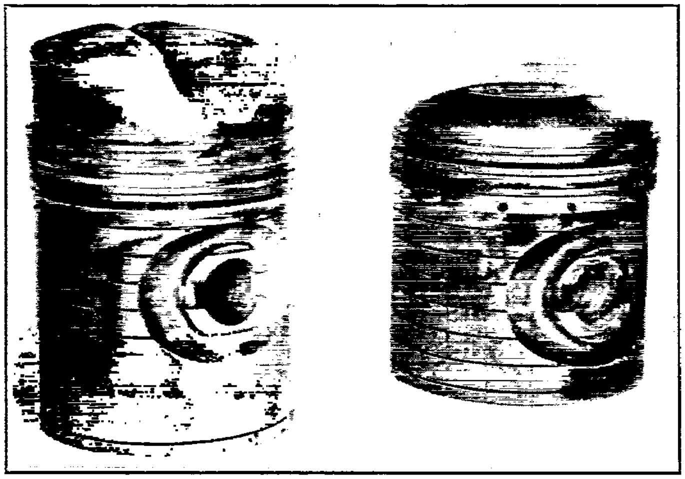

FIG. 2.—Comparison of high-compression piston (11.4:1) with the standard Liberty Army-type piston (5.4:1.)

to make the piston longer above the wrist pin bosses, as shown in Figure 2. Sectional drawings of the high compression pistons are shown in Figures 3 to 7, inclusive. Figures 3 and 4 show the form of piston head with the deep fluting which was first used. Figures 5, 6, and 7 show the form of head later adopted in an attempt to lower the rim of the piston and obtain better distribution of fuel to the air around the piston.

Aluminum alloy, magnesium alloy, and cast-iron pistons were tried in an attempt to obtain a piston material that would withstand the severe conditions of temperature and pressure obtaining during these tests. The standard Liberty piston, 5.4 ratio, with three standard rings weighs 3.75 pounds, whereas the high-compression pistons with rings had the following weights: Aluminum alloy piston 2, 5.13 pounds; aluminum alloy pistons 3 and 4, 4.92 pounds; cast-iron pistons 7 and 8, 8.89 pounds; magnesium alloy piston 12, 3.82 pounds; and magnesium alloy piston 14, 4.62 pounds.

PRIMARY FUEL SYSTEM

A gear pump and a three-throw plunger pump, both of commercial types, were used. A small volumetric fuel tank, having a capacity sufficient for a two to three-minute run at maximum load and speed, was used in making fuel consumption tests. This tank could be refilled from the by-passed overflow of the primary pump without stopping the engine. The only strainer in the fuel system was placed between the primary and injection pumps, and consisted of a double layer of 150-mesh copper screen.

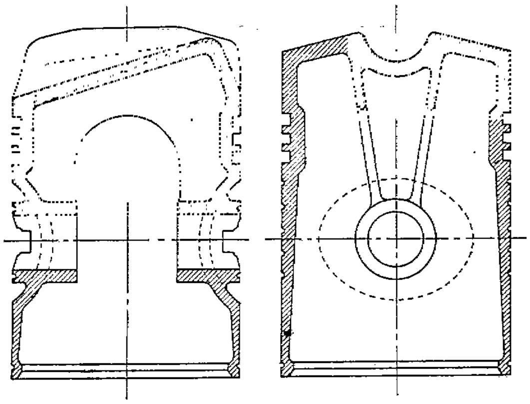

FIG. 3.—Sectional views of pistons 1 and 2 (aluminum alloy)

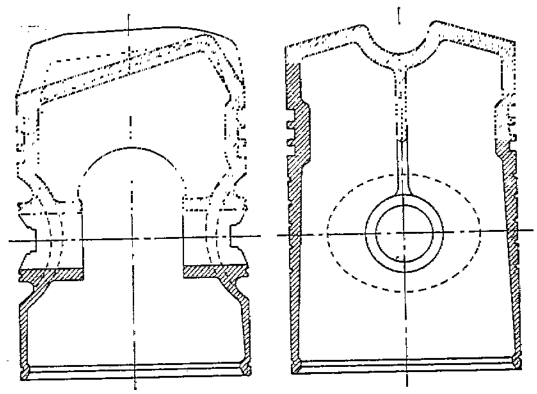

FIG. 4.—Sectional views of pistons 3 and 4 (aluminum alloy)

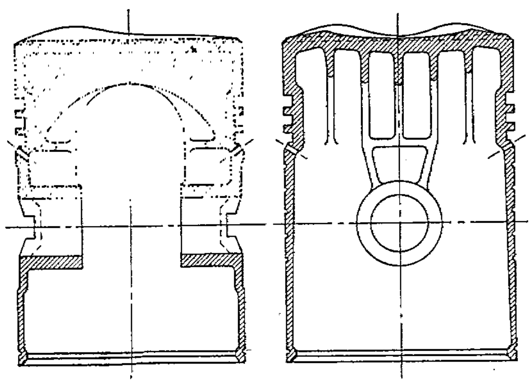

FIG. 5.—Sectional views of pistons 5 and 8 (aluminum alloy and cast iron)

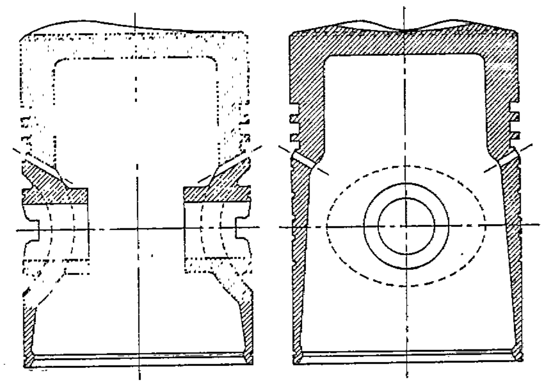

FIG. 6.—Sectional views of piston 12 magnesium alloy)

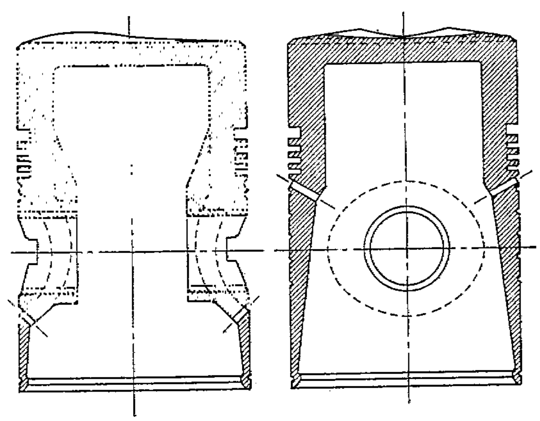

FIG.—7. Sectional views of piston 14 (magnesium alloy)

INJECTION PUMP

A photograph of the injection pump, and a sectional drawing of the pump as originally constructed are given in Figures 8 and 9, respectively. This pump was used to meter and inject the fuel, the amount of fuel injected being controlled by adjusting the length of the pump plunger stroke. The floating plunger, $\frac{7}{32}$ inch in diameter, was a lapped fit and was used without packing. It was driven forward during the injection stroke by the spring, previously compressed by means of the radial cam, and was returned solely under the action of the primary fuel pressure. The plunger stroke began when the rocker arm follower dropped off the radial cam, and ended when the spring cap came against the adjustable stop. A 173-pound helical spring was used, which exerted a force of 285 pounds on the pump plunger at

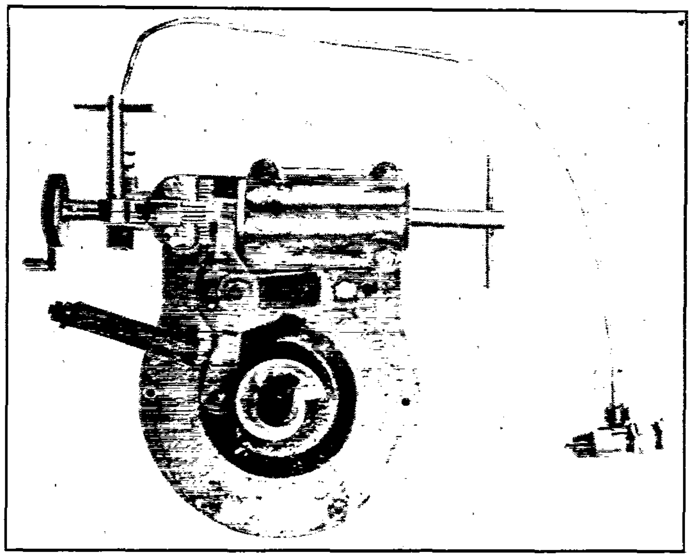

FIG. 8.—Injection pump and injection valve

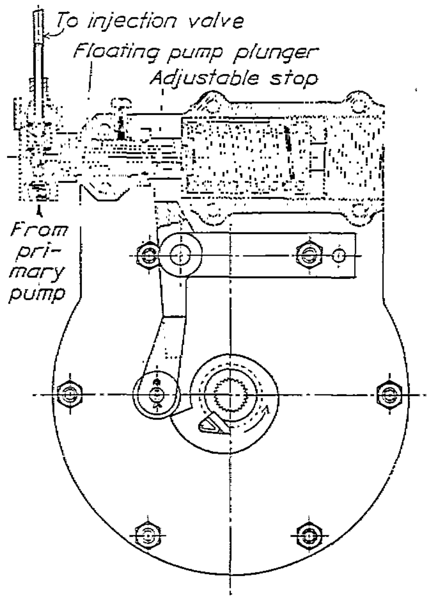

FIG. 9.—Spring-actuated injection pump

the beginning of the plunger stroke. Two spring-loaded ball check valves in series were used both for the intake side and for the discharge side of the pump. Seamless steel tubing, ⅛ inch inside, was used for the high-pressure fuel line between the injection pump and injection valve.

INJECTION VALVE

Certain limitations were placed upon the design of the injection valve: Outside dimensions were limited as it was planned to locate the valve in the metric spark plug hole, which had a diameter at the root of the thread of only 0.64 inch; it was necessary that the moving parts of the valve be light in order to meet high-speed requirements (15 injections per second at 1,800 R. P. M. of the engine); it was desirable to have a minimum volume of fuel in the valve in order to reduce the amount of fuel to be compressed before injection occurred, and to reduce the tendency for "dribbling"; and the direction of the fuel jet had to be at an angle with the axis of the valve in order to spray across the piston.

A drawing showing a section through the injection valve is shown on Figure 10. The valve stem, which was a lapped fit in the valve body, lifted and seated automatically with change of pressure in the high-pressure fuel line, and was free to lift until the spring coils closed. The opening pressure was fixed by the spring load on the valve stem. Small changes in opening pressure could be obtained by adjusting the spring cap, but large changes in opening pressure were effected by using a series of graded springs, which gave a range of opening pressures from 1,200 pounds per square inch to 5,400 pounds per square inch. These opening pressures were determined on a gauge tester; the maximum pressures in the high-pressure fuel line probably would be much higher.

Two types of nozzles (fig. 10) were used with the above injection valve: The plain orifice and the impact or lipped type. The plain orifice type, having but a single orifice, 0.021 inch in diameter, was first tried. A similar tip, having a single 0.016-inch diameter orifice was next tried, as was also the two-hole type, having two 0.012-inch diameter orifices. The plain nozzles, however, had nothing to recommend them except simplicity, and conformity between shape of jet and piston fluting. The fuel injected from this type nozzle was not finely atomized. Moreover, the plain jet gave too much penetration for the size cylinder used, as there was evidence that fuel was being deposited on the relatively cold combustion chamber wall opposite the injection valve. Consideration of these conditions led to the adoption of the lipped nozzle which gave less penetration and better atomization and distribution. As the Diesel engine fuel oil burned cleanly, the lipped nozzle could be used without excessive amounts of carbon being deposited thereon. No erosion of the lip was noticeable after running for several hours with Diesel engine fuel oil. With a lip of the approximate proportions shown in the figure, the following orifice diameters were tried: 0.012, 0.014, 0.018, 0.023, and 0.026 inch.

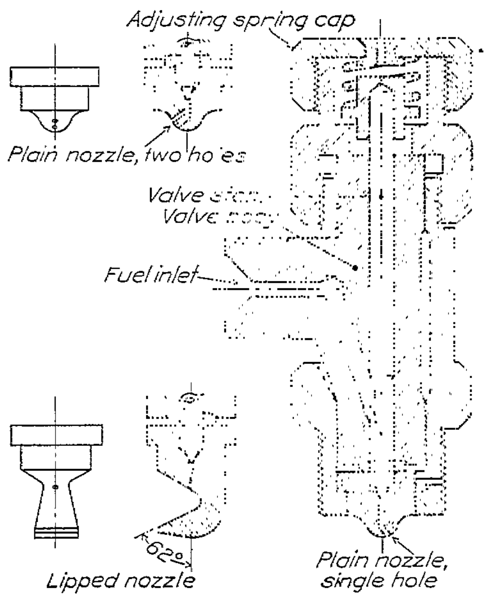

FIG. 10.—Automatic injection valve

FUEL

A high-grade Diesel engine fuel oil, having a Saybolt viscosity of 43 seconds and a specific gravity of 0.87 (34° B.) at 60° F., was used for most of the tests. Kerosene was tried in order to obtain a comparison with Diesel engine fuel oil.

PRECISION AND METHODS

Care was exercised to obtain brake horsepower, friction horsepower, and fuel consumption with a minimum of error possible with the apparatus used. In presenting the data, however, no corrections for barometer or air temperature have been applied.

Torque was measured with a Kron scale having minimum graduations of 0.2 pound. Revolutions were measured with a revolution counter and stop watch synchronized by means of a solenoid switch. With a dynamometer torque arm of 15.75 inches the equation for power is

$$\text{HP.} = \frac{\text{Scale reading (lb.)} \times \text{R. P. M.}}{4{,}000}$$

Indicated power was obtained by the addition of brake power and friction power, the latter obtained by motoring the engine with the dynamometer. Recent tests (Reference 1) made elsewhere have shown that this method of securing indicated power may be in error by as much as 10 per cent.

Fuel consumption was obtained by observing the time required to empty a small volumetric fuel tank.

Maximum combustion pressures were measured by means of a Mader microindicator, and a spring-loaded relief valve. Values for maximum pressures obtained by these methods were considered inaccurate but indicative.

An Okill indicator was used to measure compression pressures.

The method of starting, without resorting to spark ignition, with compression pressures above 300 pounds, and jacket water temperature about 100° F., consisted in motoring the engine by means of the dynamometer at speeds as low as 500 R. P. M. and then starting the injection of fuel. By this method, the engine started firing within a few revolutions after fuel was admitted. The difficulty experienced in starting the engine with some of the pistons, which gave compression pressures as low as 270 pounds, was overcome by attaching a long pipe to the engine intake, thus obtaining an increase in final compression pressures. (Reference 2.)

Variation in the timing of injection was obtained by removing and resetting the injection pump cam on its splined shaft. This shaft had 100 splines, so that a minimum adjustment of 7.2 degrees, measured in terms of crank angle, could be obtained. Optimum injection timing was obtained by making trial runs.

The lag of the fuel jet behind the pump cam timing and the interval of injection were obtained by injecting the fuel, while motoring the engine, against paper targets shellacked to the flywheel.

DISCUSSION OF RESULTS

Some preliminary tests were made using the plain nozzle type injection valve previous to adopting the lipped nozzle. During these tests using pistons of the deep-fluted type, a maximum of 22 B. HP. at 1,800 R. P. M., or 109 pounds I. M. E. P., was obtained with the plain nozzle having a single 0.018-inch orifice; but the fuel consumption was excessive, about 0.84 pound per B. HP. hour, and there was considerable exhaust smoke. The best combination of power and fuel consumption, 20 B. HP. at 1,800 R. P. M., and 0.62 pound per B. HP. hour, was also obtained with the 0.018-inch orifice. Tests with both the single 0.016-inch orifice and the nozzle having two 0.012-inch orifices gave 18 B. HP. at 1,800 R. P. M. with a fuel consumption of 0.64 pound per B. HP. hour. In order to obtain the above power it was found necessary to advance the timing of the injection pump cam to 61 degrees before T. D. C. Of this necessary advance, about 17 degrees was lag of injection behind pump cam timing. The injection interval for the pump stroke necessary to obtain 22 B. HP., when using the 0.018-inch orifice, was found from spray patterns to be 73 degrees. Consequently, injection occurred for 29 degrees after T. D. C. For the fuel quantity necessary to obtain 20 B. HP., the injection interval was 55 degrees, so that fuel was injected for 11 degrees after T. D. C. The low power output and the high fuel consumption with the plain nozzles indicated that good distribution of the fuel throughout the air charge and fine atomization were not being obtained. Also, inspection of the cylinder showed that fuel was being deposited on the combustion chamber wall opposite the injection valve, indicating too much penetration.

Following the tests with the plain nozzles, the lipped nozzle was adopted with a view to securing better atomization and distribution of the fuel with less penetration With this type nozzle it was found that deposition of fuel on the combustion chamber wall was reduced, the power output was increased and the fuel consumption lowered. The outstanding results obtained with this type nozzle are given in Table I, and are referred to in the following discussion. These tests were not continued until definite conclusions, as regards the interrelation and influence of the several variables on the engine performance could be made, as it was evident that a prolonged investigation with this cylinder would be inadvisable

MAXIMUM POWER

The maximum power developed during the tests using the lipped nozzle was 29.7 B. HP. at 1,700 R. P. M., or about 131 pounds I. M. E. P., run No. 138. Other high-power runs gave 29.1 B. HP. at 1,820 R. P. M., No. 147, and 28.3 B. HP. at 1,800 R. P. M., No. 148. The above runs were of short duration, being sustained only long enough to ensure a steady torque, in order to favor the piston as much as possible. During the longest run maintaining high power, No. 39, 25.2 B. HP. was developed for about 10 minutes. These tests were made with light alloy pistons. With a cast-iron piston, a maximum output of 21.8 B. HP. at 1,500 R. P. M. was obtained, No. 131. Because of the excessive weight of the cast-iron pistons, the engine speed was arbitrarily limited to 1,500 R. P. M. In order to obtain maximum power it was found necessary to advance the pump timing as much as 84 degrees before T. D. C. In each of the above runs, after reaching the above stated maximum power, further increase in the amount of fuel supplied had no noticeable effect on the torque. Attempts to obtain maximum power usually resulted in failure of the piston, either from burning of the head or cracking of the wrist-pin bosses.

The reason for the failure to approach more nearly the power output of this cylinder obtained when using carburetor has been attributed in part to the unsuitable form of combustion chamber. (Fig. 11.)

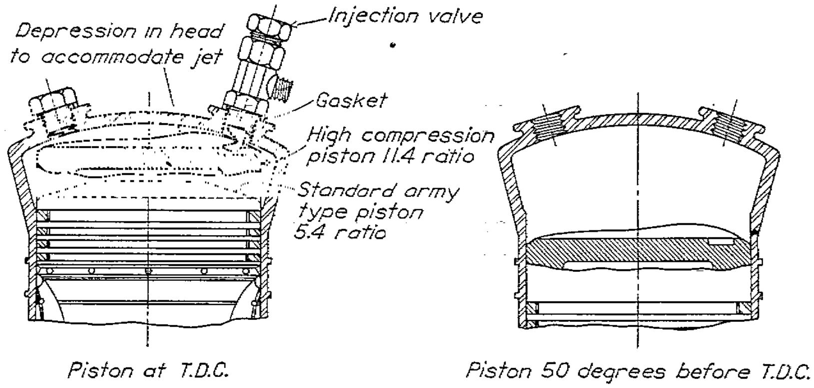

FIG. 11.—Single-cylinder Liberty engine form of combustion chamber

FRICTION POWER

The friction horsepower of the engine, when using the high compression pistons, was somewhat higher than that obtained with the standard Liberty piston. The friction varied for the several pistons used, and was sensitive to changes in jacket water and oil temperatures. In computing indicated power, it was endeavored to obtain a value for friction power taken under conditions simulating those existing during the respective power runs.

FUEL CONSUMPTION

Best fuel economy was obtained when developing only 12 to 15 B. HP., or less than one-half the normal output of this cylinder when using a carburetor. Run No. 112 gave a fuel consumption with Diesel engine fuel oil of 0.47 pound per B. HP. hour, and run No. 111 gave 0.49 pound per B. HP. hour, the indicated specific fuel consumption being about 0.30 pound per I. HP. hour, in both cases.

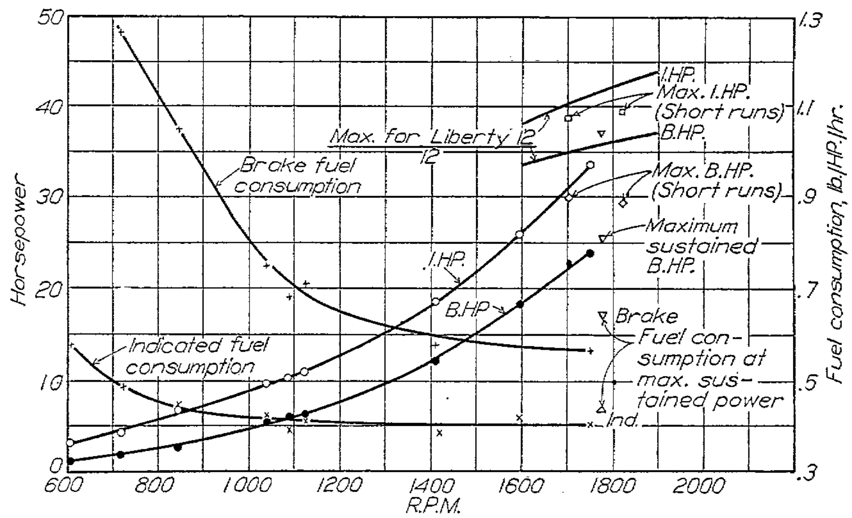

FIG. 12.—Single-cylinder Liberty engine compression ignition. Power and fuel consumption at part loads. Propeller load: 24 B. HP. at 1,750 R. P. M. taken as normal. Lipped nozzle with 0.018-inch diameter hole. Diesel engine fuel oil. 11.4 compression ratio

The specific fuel consumption increased with increase in power output above 15 B. HP. as shown in run No. 114, which gave a fuel consumption of 0.52 pound per B. HP. hour, when developing about 21 B. HP. at 1,750 R. P. M. under conditions directly comparable to those existing in run No. 112. The fuel consumption for the highest sustained power run No. 39, was about 0.64 pound per B. HP. hour showing again that increased power was obtained at the expense of fuel economy.

Part load runs were made for the purpose of determining fuel consumption over a wide range of loads. Table I gives the data for these runs, Nos. 42 to 60. A value of 24 B. HP. at 1,750 R. P. M. was taken arbitrarily as normal power and a series of runs made at an approximately constant speed of 1,740 R. P. M. with varying torque, and at varying speeds and torque corresponding to a propeller load. In the latter runs, data for which are plotted on Figure 12, the speed was varied as the cube root of the power, and the torque adjusted to give the required power at this speed. Test points for maximum power obtained, Nos. 138 and 147, and for maximum sustained power, No. 39, are plotted on this figure; the power developed in one cylinder of a Liberty 12 engine using carburetor is also shown. The specific fuel consumptions obtained during these series of part load runs were not as low as obtained during some other runs at the same load. However, all data for the part load runs were taken under the same engine conditions, except for variations in jacket water and oil temperatures, so that the results obtained during the individual runs of the part load series are comparable. The curves of specific fuel consumption against power for the series of part load runs at constant speed, shown on Figure 13, indicate that the consumption increased with power, the best economy being obtained at 12 B. HP. output. It can be seen that the fuel consumption for 24 B. HP. is lower than would be expected from the trend of the curves. This run was made some time before the other runs of the series, and the apparent discrepancy may have been due to conditions within the engine changing in the interim.

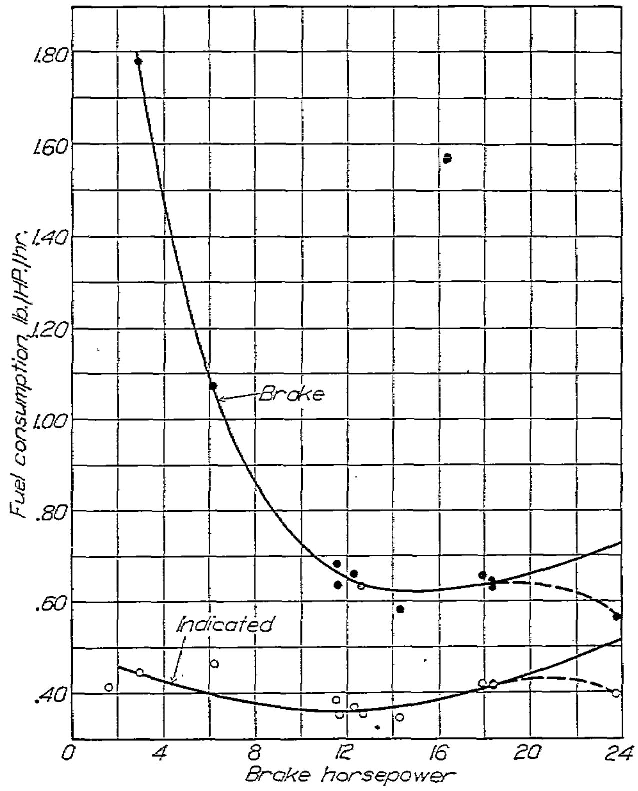

FIG. 13.—Single-cylinder Liberty engine compression ignition. Fuel consumption at part loads. Constant speed. Average speed, 1,740 R. P. M. Lipped nozzle with 0.018-inch diameter hole. Diesel engine fuel oil. 11.4 compression ratio

Tests were made to determine the variation, if any, in fuel consumption during a run of appreciable duration maintaining constant power output. In run No. 116, 16.3 B. HP. at 1,740 R. P. M. was held for approximately one hour, during which the average fuel consumption was 0.51 pound per B. HP. hour with not more than a 3 per cent variation during any of the 7-minute intervals when the rate of fuel flow was timed. In run No. 130, using a cast-iron piston, 16 B. HP. at 1,520 R. P. M. was maintained for nearly 2 hours with an average fuel consumption of 0.66 pound per B. HP. hour. The specific brake fuel consumption for the first 6-minute interval was 0.69 pound, and for the last time interval was 0.66 pound; the lowest consumption during any time interval being 0.65 pound. During run No. 116 the aluminum alloy piston No. 4 failed, due to burning of the head; and during run No. 130 the cast-iron piston No. 7 failed, due to cracking of the wrist pin bosses.

Runs No. 40 and 41 give comparative power and fuel consumption figures for the engine operating under the same conditions with kerosene and with Diesel engine fuel oil. These two runs were made without stopping the engine, the change in fuel being effected while under power. Diesel engine fuel oil gave a lower specific fuel consumption and higher power.

Tests with the lipped nozzle to determine the effect of nozzle orifice diameter on specific fuel consumption showed that best economy was obtained with a 0.014-inch orifice. Decreasing the orifice diameter still further, to 0.012 inch, increased the fuel consumption due, probably to the longer interval of injection with the smaller orifice. The 0.014-inch orifice was, apparently, a good compromise between the somewhat finer atomization obtained with a small orifice and the shorter interval of injection with a large orifice.

FLEXIBILITY

The engine could be idled smoothly at 400 R. P. M. when using Diesel engine fuel oil, and at 630 R. P. M. when using kerosene. The acceleration under load was not satisfactory as compared with the carbureted engine due, in part, to the fact that injection timing could not be varied when the engine was running.

STARTING CONDITIONS

No difficulty was experienced in starting with compression ignition only, when the compression pressure was over 300 pounds and the jacket water temperature about 100° F. With

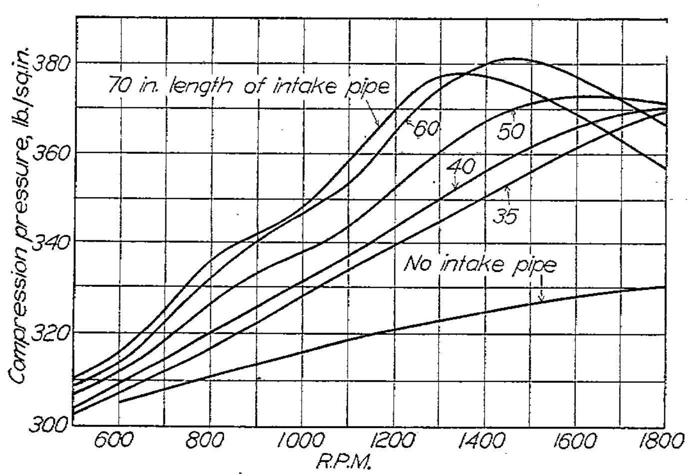

FIG. 14.—Single-cylinder Liberty engine. Increase in compression pressure obtained with long intake pipe. Pipe 3 inches diameter

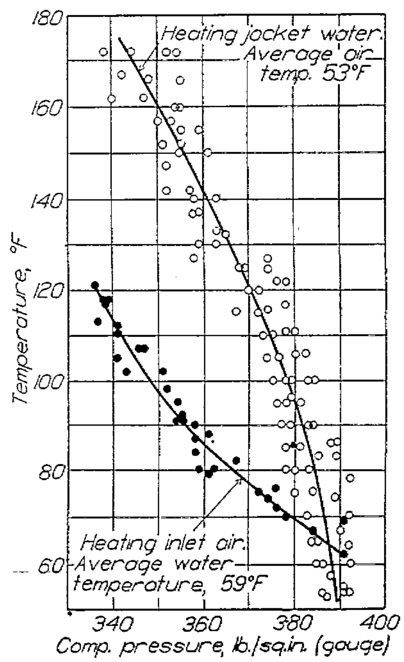

FIG. 15.—Single-cylinder Liberty engine compression ignition. Starting condition. Lipped nozzle with 0.014-inch diameter hole. Diesel engine fuel oil. 11.4 compression ratio

lower compression pressures, it was found necessary to employ some auxiliary means for starting; and it was desirable not to use spark ignition. The engine could be started by increasing the jacket water or inlet air temperature or by increasing the compression pressure through increasing the initial pressure. Heating the jacket water caused delay and, besides, was least effective in facilitating starting. Heating the inlet air was very effective but necessitated the use of some form of heater. However, it was found that the engine could be started, with jacket water and inlet air at room temperature, simply by attaching a long pipe to the intake, thus obtaining an increase in compression pressure. (Reference 2.) In order to realize the advantages of this method, it was necessary to motor the engine at fairly high speeds. By using a suitable length of pipe for a given engine speed the compression pressure was increased as much as 17 per cent. Figure 14 shows the increase in compression pressure, as measured by the Okill gauge, that was obtained by using the long intake pipe.

With a constant compression ratio, and using a long intake pipe to obtain variation in compression pressure, a short investigation was made to determine the relation between jacket water temperature, with inlet air at room temperature, and inlet air temperature, with jacket water at room temperature, and the compression pressure at which the engine would start firing. (Reference 3.) The results are plotted on Figure 15.

These starting tests indicate the sensitiveness of this compression ignition engine to changes in final compression pressure, with constant compression ratio and, further, suggest the need for providing some means, such as supercharging, to compensate for the reduced air pressures and temperatures that would be encountered when operating at high altitudes, in order to insure reliable operation.

MAXIMUM PRESSURES

Excessively high maximum combustion pressures were encountered during this investigation. Records taken with a Mader microindicator, indicated maximum pressures of the order of 1,600 pounds per square inch; and a spring-loaded relief valve, located in the second spark plug hole and set to open at 1,300-pound gauge pressure, "popped" regularly when the engine was developing only moderate torque, about 60 pounds B. M. E. P.

Antiknock dopes, and cooled exhaust gases introduced with the entering air charge were tried in an attempt to reduce these high maximum pressures but, although somewhat smoother running and a slight increase in power output for given engine settings were obtained, maximum pressures were not reduced appreciably. The doped fuels tried consisted of fuel oil and 5 per cent by volume of aromatic amines, and fuel oil and 0.3 per cent by volume of tetra-ethyl lead. An increase in power was obtained by directing a jet of exhaust gases, cooled to 110° F., into the engine intake through a section of ½-inch pipe, but further investigation indicated that this increase in power may have been due to the increased turbulence resulting from directing the jet into the intake rather than to smoother combustion resulting from using the exhaust gases as a diluent.

EFFECT OF TURBULENCE ON POWER

While experimenting with exhaust gases introduced into the engine intake through a small pipe, it was found that variations in power occurred when the jet of gases was directed toward different parts of the intake. The injection valve was located to the left of the intake opening and nearly in the same horizontal plane. Directing the jet toward the right side of the intake, or away from the injection valve, gave no increase in torque or speed, but directing the jet to the left, or toward the injection valve, gave an appreciable increase in torque and speed. In one test, while developing about 11 B. HP. at 1,650 R. P. M., the torque and speed were both increased about 10 per cent when the jet of exhaust gases was directed toward the injection valve. The increase in power thus obtained was attributed to the increased turbulence in the vicinity of the injection valve. However, another explanation might be that, supposing combustion to start in the vicinity of the injection valve, directing the jet toward the injection valve may have resulted in placing the inert gases in that part of the combustion chamber where they could be most effective in acting as a diluent, thus slowing up the rate of burning of the portion of the combustible mixture first ignited, whereas, directing the jet away from the injection valve may have resulted in preventing the inert gases from coming into the region where ignition first occurred.

A further investigation of the effect of turbulence was made by using a throttled inlet port. A metal plate, that covered all but a 90° sector of the intake opening, was clamped over the intake. When the opening was changed from the lower right position to the lower left position an increase in torque of about 9 per cent and an increase in speed of about 13 per cent was obtained.

A further study of the effect of the direction of the entering air stream on power and fuel consumption was not made, but such an investigation, with a more suitable cylinder, might be profitable.

FUEL PUMP CHARACTERISTICS (Reference 4)

An investigation of the spring-actuated fuel pump was made, using the spray pattern method previously described, to determine the effect on the lag and interval of injection, of injection valve orifice size, injection valve opening pressure and fuel pump stroke, or fuel quantity. Spray patterns were taken, with the engine motored at a constant speed of 1,600 R. P. M., for nozzle orifices 0.012, 0.014, 0.018, and 0.022 inch in diameter, for opening pressures of 2,200, 3,200, and 5,000 pounds per square inch and for varying fuel quantities. The results obtained with three sizes of nozzle orifice are shown on Figures 16, 17, and 18. Figure 19 is a cross plot

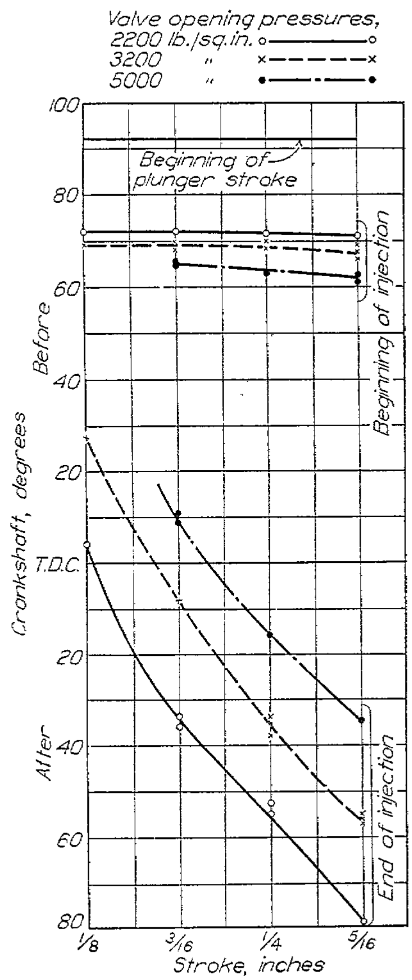

FIG. 16.—Lipped nozzle with 0.012-inch diameter hole

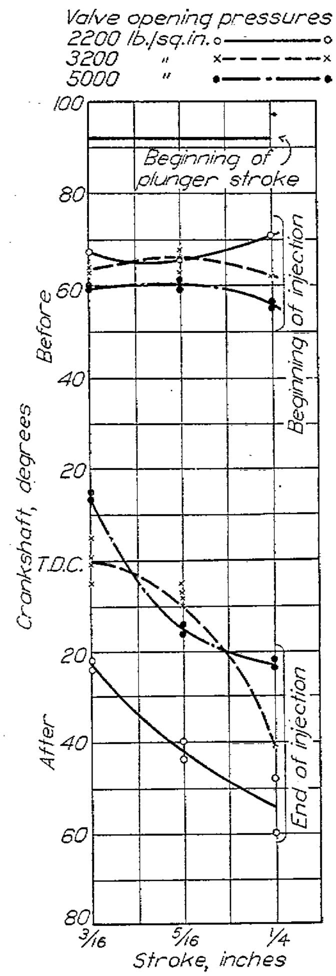

FIG. 17.—Lipped nozzle with 0.014-inch diameter hole

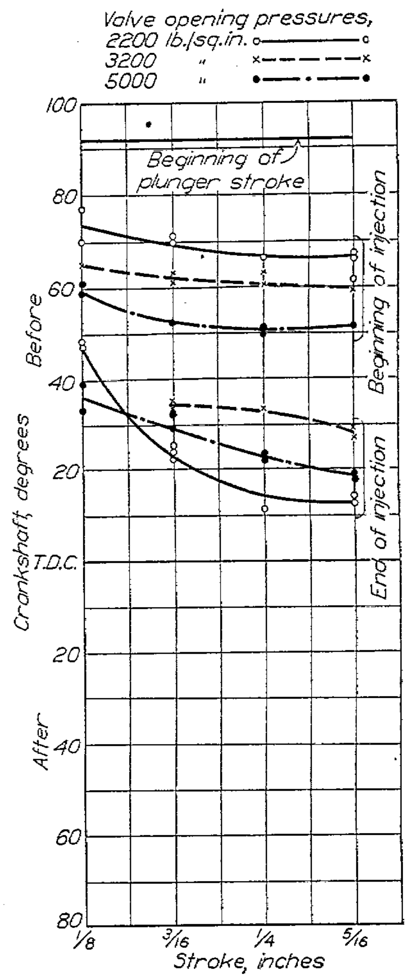

FIG. 18.—Lipped nozzle with 0.018 inch diameter hole

Characteristics of spring actuated fuel pump. Effect of pump stroke and valve opening pressure on lag and interval of injection. Diesel engine fuel oil. R. P. M., 1,600

showing the effect of orifice diameter on lag and interval of injection for one opening pressure and pump stroke. Although somewhat erratic, due to the difficulty experienced in defining the beginning and ending of the spray pattern, the results show, in general that:

> Lag increases with increase in orifice size, injection valve opening pressure, and pump stroke. Interval of injection decreases with increase in orifice size and opening pressure, and with decrease in pump stroke.

Distribution of fuel on the spray patterns indicated, in some cases, that the injection valve stem rebounded from its seat once or twice during a single injection. The above information can not be used in determining the lag and interval of injection for the power runs given in Table I, as data on opening pressure and pump stroke were not definitely determined during these power runs.

The spring-actuated injection pump, for given conditions, gives an approximately constant interval of injection, in terms of absolute time, independent of engine speed. When measured in degrees of crank travel, however, the injection interval varies directly with engine speed.

CONCLUSIONS

The results obtained during this investigation can be used only as a general indication of the possibilities of, and the difficulties to be overcome in applying airless injection with compression ignition to aircraft engines. The combustion chamber of the Liberty cylinder was not suitable for compression ignition research, and probably accentuated the difficulties encountered in attempting to obtain high mean effective pressures.

It appears that a reasonably low fuel consumption can be obtained at reduced loads, as it was possible, with an unfavorable form of combustion chamber, to obtain an indicated specific fuel consumption as low as 0.30 pound per I. HP. hour when developing about 16 B. HP. at 1,730 R. P. M. With this engine, specific fuel consumption increased materially with power output above 16 B. HP. at these speeds. The maximum power output obtained, 29.7 B. HP. at 1,700 R. P. M., may be considered encouraging in view of the unsuitable combustion chamber in which good distribution of the fuel could not be obtained. However, high power output could be maintained for only very short periods of time due to piston failure. Excessive exposed area of the piston to combustion temperature accentuated piston trouble. No difficulty was experienced in metering and injecting the small quantities of fuel required when operating at engine speeds as high as 1,850 R. P. M. The extreme injection advance found necessary accounts in part for the high maximum combustion pressures encountered, as practically the entire fuel charge was in the cylinder at the time ignition occurred, so that constant volume combustion of most of the fuel charge resulted. This indicates the need for reducing the lag of ignition to a minimum, in order to obviate the need for extremely early injection. The brief tests indicating the effect of turbulence on power may be considered valuable in pointing out what may be obtained on this score. The tests showing the sensitiveness of compression ignition to changes in temperature and pressure of the induced air indicate clearly the need for supercharging in order to obtain reliable engine operation at altitude.

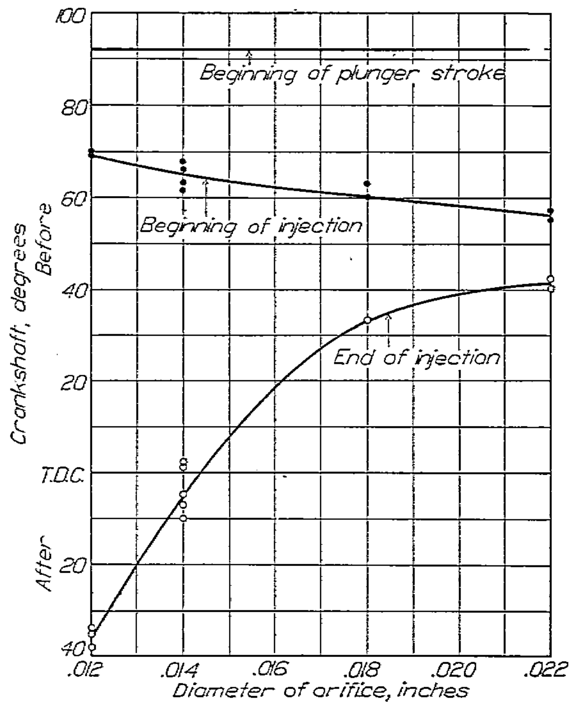

FIG. 19.—Injection-valve opening pressure, 3,200 pounds. Pump stroke, ¼ inch. R. P. M., 1,600. Characteristics of spring-actuated fuel pump. Effect of orifice size on lag and interval of injection

REFERENCES

The information given in References 2, 3, and 4 was obtained during the present investigation, but has been previously reported.

1. H. Moss and W. J. Stern — Mechanical Efficiency of an Internal Combustion Engine. From "Automobile Engineer" (Eng.) March, 1925.
2. Robertson Matthews and Arthur W. Gardiner — Increasing the Compression Pressure in an Engine by Using a Long Intake Pipe. N. A. C. A. Technical Note No. 180–1924.
3. Robertson Matthews and Arthur W. Gardiner — The Influence of Inlet Air Temperature and Jacket Water Temperature on Initiating Combustion in a High-Speed Compression Ignition Engine. N. A. C. A. Technical Note No. 185–1924.
4. Robertson Matthews and Arthur W. Gardiner — The Time Lag and Interval of Discharge with a Spring-Actuated Fuel Injection Pump. N. A. C. A. Technical Note No. 159–1923.

TABLE I

TEST RESULTS. RUNS WITH LIPPED NOZZLE INJECTION VALVE AND DIESEL ENGINE FUEL OIL

Run No.	Piston No.	Duration, minutes	R. P. M.	B. H P.	F. HP.	I. HP.	B. M. E. P., lb./sq. in.	I. M. E. P., lb./sq. in.	Mechanical efficiency, per cent	Fuel consumption, lb./B. HP. hr.	Fuel consumption, lb./I. HP. hr.	Nozzle orifice diameter, in.	Fuel pump timing deg. before T. D. C.[1]	Water out temperature, °F.	Oil out temperature, °F.
32	2	10	1,743	19.5	10.0	29.5	64.5	96	66	0.601	0.397	0.018	74	140	60
39	2	10	1,779	25.2	10.3	35.5	81.6	115	71	.637	.452	.018	74	145	58
40	3	12	1,724	21.2	10.4	31.6	71.0	106	67	.596	.400	.018	74	140	122
41	3	12	1,750	23.7	10.6	34.3	78.1	113	69	.566	.391	.018	74	140	121
42	3	34	608	1.17	2.1	3.27	11.0	31	36	1.620	.580	.018	60	170	116
43	3	16	1,040	5.44	4.1	9.54	30.2	53	57	.750	.428	.018	60	170	116
44	3	20	846	2.86	3.8	6.66	19.4	45	43	1.050	.451	.018	60	176	101
45	3	9	1,410	12.2	6.2	18.4	49.9	75	66	.578	.382	.018	53	168	121
46	3	5	1,595	18.1	7.8	25.9	65.5	94	70	.60	.419	.018	60	158	122
47	3	16	1,090	5.9	4.4	10.3	31.2	54	57	.682	.391	.018	60	152	121
48	3	15	1,124	6.15	4.5	10.65	31.6	55	58	.710	.410	.018	53	166	121
49	3	15	720	1.62	2.6	4.22	13.0	34	38	1.270	.488	.018	53	165	119
50	3	7	1,730	2.98	9.0	11.98	9.9	40	25	1.78	.443	.018	67	169	132
51	3	10	1,745	6.2	9.3	15.5	20.5	51	40	1.07	.466	.108	67	168	120
52	3	8	1,680	11.56	9.0	20.56	39.7	70	56	.686	.386	.018	67	156	112
53	3	8	1,760	12.6	10.0	22.6	41.2	74	56	.635	.355	.018	60	153	120
55	3	9	1,740	11.6	9.7	21.3	38.4	70	54	.642	.349	.018	67	141	125
56	3	8	1,725	12.3	9.8	22.1	41.3	74	56	.660	.368	.018	67	135	123
57	3	6	1,760	17.9	10.0	27.9	58.7	91	64	.655	.421	.018	60	145	120
58	3	6	1,760	18.3	9.8	28.1	59.9	92	65	.646	.420	.018	60	155	122
59	3	6	1,753	18.3	9.5	27.8	60.1	91	66	.638	.420	.018	60	162	124
60	3	14	1,749	1.62	9.3	10.92	5.35	36	15	2.82	.418	.018	60	170	126
111	4	10	1,722	13.6	8.7	22.3	45.5	75	61	.488	.298	.014	77	120	105
112	4	9	1,731	15.7	8.8	24.5	52.3	82	64	.471	.302	.014	77	123	115
114	4	6	1,748	20.8	9.0	29.8	68.5	98	70	.524	.366	.014	77	120	123
116	4	59	1,737	16.3	8.9	25.2	54.4	84	65	.510	.330	.014	77	120	124
130	7	114	1,516	16.0	7.1	23.1	60.7	88	69	.662	.458	.014	77	100	128
131	8	2	1,500	21.8	7.0	29.8	83.6	115	73	--------	--------	.022	77	120	112
138	12	½	1,700	29.7	9.0	38.7	101.	131	77	--------	--------	.018	77	125	110
147	14	½	1,820	29.1	10.2	39.3	92.4	125	74	--------	--------	.026	84	140	--------
148	14	1	1,800	28.3	10.0	38.3	90.9	123	74	--------	--------	.026	84	140	--------

[1] Fuel pump cam setting. Time when fuel enters cylinder not given.

Pistons 2, 3, and 4 aluminum alloy; pistons 7 and 8 cast iron; pistons 12 and 14 magnesium alloy.
Primary pump pressures up to 250 pounds. Injection valve opening pressures from 1,200 to 5,400 pounds.
Run 39, maximum sustained power; cracked water jacket during run.
Run 40, kerosene fuel used.
Run 112, best fuel economy.
Run 116, injection nozzle clean at end of run; failure of piston.
Run 130, injection nozzle clean at end of run; failure of piston.
Run 131, piston bosses cracked during run.
Run 138, maximum power, piston bosses cracked during run.