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RATES OF FUEL DISCHARGE AS AFFECTED BY THE DESIGN OF FUEL-INJECTION SYSTEMS FOR INTERNAL-COMBUSTION ENGINES

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SUMMARY

Using the method of weighing fuel collected in a receiver during a definite interval of the injection period, rates of discharge were determined, and the effects noted, when various changes were made in a fuel-injection system. The injection system consisted primarily of a by-pass controlled fuel pump and an automatic injection valve. The variables of the system studied were the pump speed, pump-throttle setting, discharge-orifice diameter, injection-valve opening and closing pressures, and injection-tube length and diameter.

The results show that, for the same orifice diameter, the rate of discharge increased with the pump speed and injection-valve opening and closing pressures. For the same pump speed, the throttle setting had little effect upon the rate of discharge up to the point of cut-off. The rates of discharge conformed approximately to the contour of the cam only with the larger size orifices; with the smaller orifices, because of the excess energy supplied by the pump over that utilized for discharge, the higher intensity reflections and reenforcements altered the discharge to a different conformation. The tube length was found to have little effect on either the rate of discharge, injection period, or injection lag. The data show that the pressure before the injection valve is affected substantially if injection-tube diameters are used that are below the critical diameter.

INTRODUCTION

In the operation of compression-ignition engines it is particularly desirable to control the rate of burning in the combustion chamber. Rapid burning of the fuel is manifested by a rapid rise of pressure in the engine cylinder, resulting in rough running of the engine. High cylinder pressures usually accompany this rapid burning, increasing the stresses on the working parts and often reducing the thermal efficiency of the engine. For the purpose of controlling this rate of pressure rise, efforts are being made by research workers to reduce to a minimum the time that elapses between the start of injection and the ignition of the fuel, i. e., the ignition lag. With a small ignition lag the rate of burning may undoubtedly be influenced to some extent by the rate and manner in which the fuel is injected into the combustion chamber.

With spark ignition and fuel injection the mass rate of fuel discharge is not as important as with compression ignition. However, the injection period and the velocity with which the spray is injected are important because both factors affect the fuel distribution.

Data on the rate of fuel discharge are of considerable assistance in the selection of the size and type of fuel-injection pump and other integral parts of the injection system that are necessary to approach the desired conditions. Data on rate of discharge in conjunction with studies of the hydraulics of an injection system and other experimental data on the factors influencing spray characteristics give the engine designer information (1) on the type of fuel spray to anticipate at different conditions of operation, (2) on the relative merits and limitations of various injection systems, and (3) on the changes to be made in an injection system to obtain as near as possible the desired results. Information (3) is particularly desirable with respect to standardization of pumps and injection valves.

The analytical and experimental investigations of Sass (reference 1) and Rothrock (reference 2) have shown that changes in the rate of fuel discharge can be effected through minor changes in the injection system. Some data on the rates of fuel discharge have been given by De Juhasz. (Reference 3.) He showed that the rate of discharge varied considerably with both the injection pressure and orifice size. Gerrish and Voss of this laboratory (reference 4) have determined the rate of discharge of a fuel pump controlled by a by-pass valve. The significant observation made on these tests was that, with this pump and an automatic injection valve, there was a period at the beginning of injection during which discharge took place at a very low rate. These rate-of-discharge tests, together with the tests by Rothrock (reference 2) on the hydraulics of fuel-pump injection systems, indicated the need to continue investigations on rates of fuel discharge.

The purpose of the present work was to furnish supplementary information on the rate of fuel discharge with the same pump and practically the same set of conditions used in the tests by Rothrock and to determine how the rate of fuel injection with a given pump can be varied by changes in the injection valve and tubing. Tests were made with a by-pass port-controlled fuel-injection pump at various pump-shaft speeds, throttle settings, orifice diameters, injection-valve closing pressures, and injection-tube lengths and diameters. As the rate of fuel injection depends entirely on what actually occurs in the injection system, in the explanation of the results presented in this report, continual mention will be made of reference 2 where the hydraulics of a fuel-injection system are treated extensively.

The investigation was conducted by the National Advisory Committee for Aeronautics at Langley Field, Va., during the summer of 1931.

METHODS AND APPARATUS

In this investigation the fuel discharged from an injection valve during a small interval of the injection period was collected in a receiver and weighed. This method was first used for this type of research by De Juhasz. (Reference 3.) The apparatus (fig. 1) consisted essentially of a fuel-injection system with a fuel pump and an injection valve to produce the spray, a rotating steel disk with a slot near its outside circumference to interpose a portion of the discharge during a time interval of 1 pump degree, a receiver to collect the fuel passing through the disk slot, and an epicyclic gear train that reduced the pump speed to one-half the disk speed and with provisions to alter and record the angular phase of the disk with respect to the pump.

The pump employed in these tests is shown diagrammatically in Figure 2 and is completely described by Wild in reference 5. When the plunger advances, the inlet ports are covered and a pressure is built up in the injection tube, culminating in the opening of the injection valve and the commencement of discharge. At a later part of the stroke, the slanted groove on the plunger communicating to the pressure side of the pump is brought opposite the inlet or suction port,

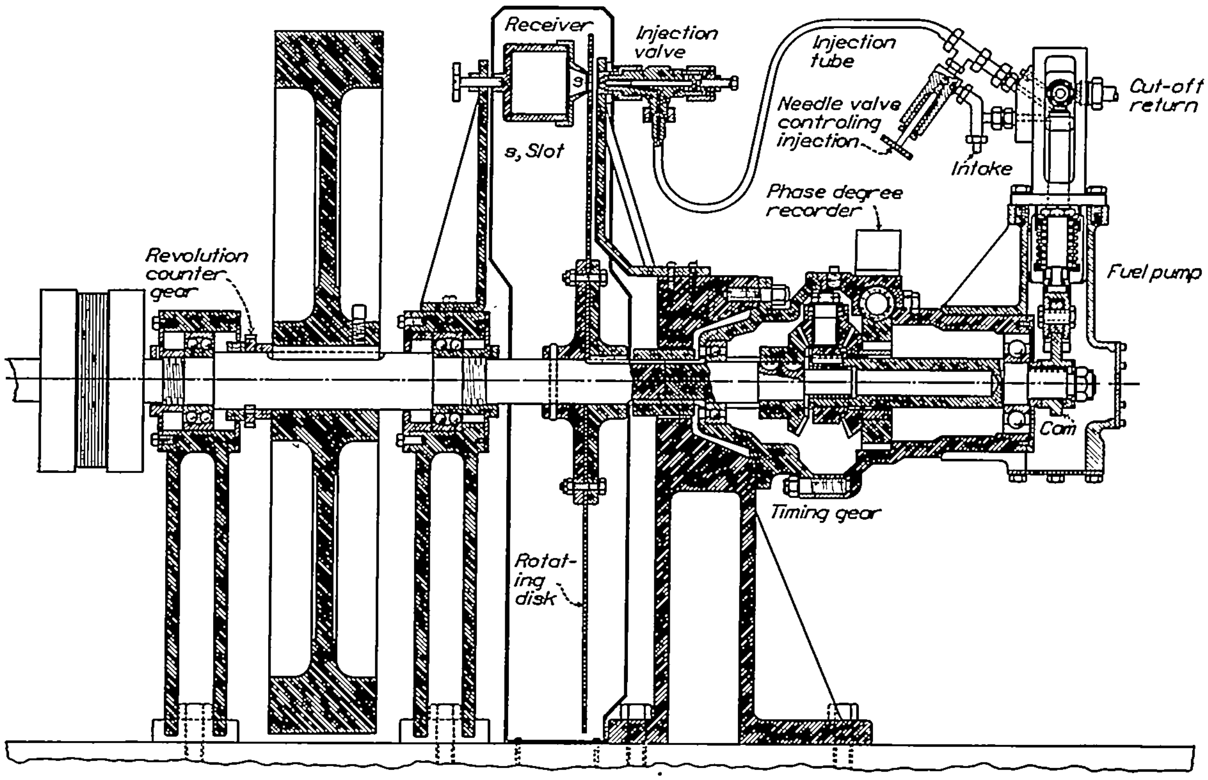

FIGURE 1.—Apparatus for measuring rate of fuel discharge

thus releasing the pressure in the injection tube and stopping the discharge. The amount of the effective stroke is regulated by the angular rotation of the plunger, which causes the slanted groove to be brought opposite the suction port earlier or later in the stroke.

The fuel discharged by the pump was returned to the pump when the needle valve (fig. 1) was open, and was discharged through the injection valve when the needle valve was closed. This valve was located as near to the injection line as possible to avoid any alterations to the conditions of flow that existed without the valve in place.

The time-recording apparatus (not shown in fig. 1) consists of an electrically operated revolution counter and a stop watch for determining the rate and number of injections for which fuel was collected. The weighing of the collected fuel was made on an analytical balance.

The test procedure was as follows: The weighed empty receiver was mounted on the bracket opposite the injection nozzle and as near to the nozzle as the rotating disk would permit. The apparatus was brought to the test speed. The needle valve controlling the beginning and end of discharge was closed and at the same instant an electric push-button was depressed, starting simultaneously the revolution counter and the stop watch. After a fixed time, usually one minute, the discharge-control valve was opened and the electric switch was again depressed, thus simultaneously ending the discharge and the counting of the number of discharges and of the time. The receiver and contents were weighed to determine the weight collected. The angular phase of the disk with respect to the pump was then altered and the procedure was repeated.

Except when the orifice diameter and the injection-tube length and diameter were varied to study their effect on the rate of discharge, a nozzle having a single 0.022-inch orifice and an injection tube 34 inches long with an inside diameter of 0.125 inch were used.

The speeds at which the pump was tested were varied from 100 to 1,000 r.p.m., the throttle settings from one-fourth to full load, orifice diameters from 0.008 to 0.042 inch, injection-valve closing pressures from 500 to 3,000 pounds per square inch (the injection-valve opening pressure was 1.4 times the closing pressure), injection-tube lengths from 20 to 74 inches, and injection-tube inside diameters from 0.041 to 0.188 inch. The fuel oil used had a specific gravity of 0.859 and an absolute viscosity of 0.022 poise (approximately 5 Say olt seconds Universal) at 100° F. and atmospheric pressure.

The effective injection pressure behind the nozzle was determined from the relationship:

$$\Delta W = VAc\gamma t = Act\sqrt{2gp\gamma} \qquad (1)$$

in which

ΔW, weight collected per increment of discharge, 1 pump degree.

V, velocity of flow through the orifice, assumed to be constant over the whole cross section of the orifice. (Reference 6.)

A, area of the discharge orifice.

c, coefficient of discharge, 0.92. (Reference 6.)

t, time interval of discharge during which fuel was collected.

γ, weight of fuel per unit volume.

Solving for p

$$p = K\left(\frac{\Delta W}{t}\right)^2 \qquad (2)$$

in which

$$K = \frac{1}{2g\gamma(Ac)^2}$$

PRECISION OF RESULTS

The precision with which the weights were collected depended on what portion of the actual fuel discharge was collected in the receiver during the interval of injection intercepted by the slot of the disk or its equivalent—1 pump degree. There was no other possible criterion than collecting a number of discharges in a bottle, weighing them, determining the amount of fuel per discharge, then comparing the weight thus obtained with that obtained by integrating the curve of increment weights that were collected with the receiver per increment of injection against degrees. The amount of fuel collected by discharging in a bottle was more than that collected in the receiver of the rate-of-discharge tests but the excess was rarely more than 2 per cent. A sufficient number of injections were collected in both tests to obtain a good average weight per injection. Increment weights were collected in the receiver every half pump degree when the rate at which the fuel was discharged increased or decreased rapidly and every 1 or 1.5 pump degrees when the rate remained approximately constant. The accuracy of the analytical balance under the conditions used in these tests was ± 0.02 gram; errors from this source were therefore negligible.

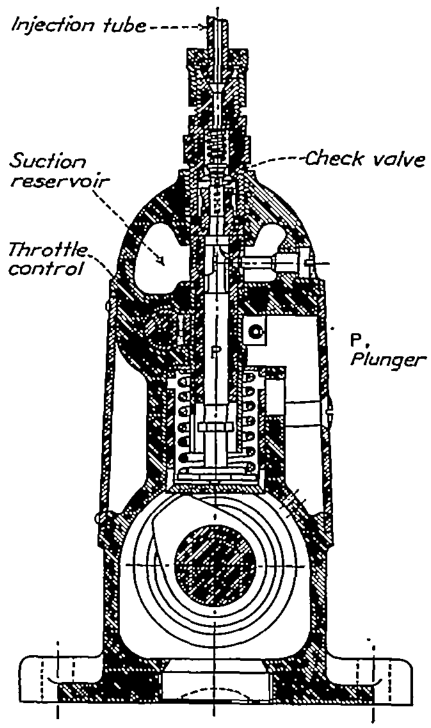

FIGURE 2.—Diagrammatic sketch of fuel pump used in tests

It has been observed by Rothrock and Marsh (reference 7) in their investigation of injection lags for pump-injection systems with an oscilloscope using approximately the conditions and system of the present tests, that there was an interval at the beginning and end of injection during which fuel was discharged at a very low rate, as was evidenced by the lightness of the spray produced. The duration of this light foredischarge and afterdischarge varied with the conditions, and was as long as 25 per cent of the total period at full throttle. Discharges were shown with negative injection lags; that is, spray appeared before the by-pass ports were fully covered at the pump. During only a portion of this foredischarge or afterdischarge period was it possible to collect fuel in the present tests. At no condition was there a negative lag observed in the present tests. As the amount of fuel collected by discharging into the bottle was rarely more than 2 per cent greater than that collected in the cup, this foredischarge or afterdischarge must have been too small to merit any consideration.

The precision of the computed pressures depends on how accurately the factors of equations (1) and (2) were determined and how closely the conditions represented by these equations were approached. The assumptions under which these equations were derived and the accuracy with which each of the factors involved were determined were discussed by Gelalles in reference 6. The same accuracy obtains in this work. Where effective pressures far below the closing pressure of the valve are shown in the results, the probability was that the injection-valve stem either bounced on its seat (reference 7) or remained barely lifted off its seat, thus throttling the flow. The computed pressures under these conditions can not therefore represent the actual pressures behind the nozzle. Where pressures equal to or above the closing pressure of the valve are shown, they represent with a fair degree of accuracy the actual pressures behind the nozzle, subject to the limitations under which equations (1) and (2) were derived.

RESULTS AND DISCUSSION

Rates of discharge at different pump speeds.—In Figures 3 to 6 are given the rates of dicharge and the computed effective injection pressures at varying pump speeds. For the purpose of easier comparison of the data, the zero points for the abscissa scales were selected arbitrarily so that the point at which the by-pass ports of the pump were covered by the advancing plunger (resulting in the commencement of discharge through the injection valve) comes at a fixed interval after the zero under all conditions of plotting.

Speeds below 215 r. p. m. at an injection-valve closing pressure of 2,500 pounds per square inch and below 108 r. p. m. at a closing pressure of 500 pounds per square inch were not used with the 34-inch injection tube because the pump could not build sufficient pressure to maintain the valve fully open throughout the effective stroke of the plunger. The amount of fuel discharged per injection diminished, and became irregular immediately below the foregoing speeds.

An inspection of the figures shows the following characteristics:

1. The interval in fractions of a second between the point at which the by-pass ports were closed and the commencement of discharge, or the injection lag, was increased with the decrease of speed.

2. The rate of discharge constantly fluctuated at the lower speeds. The speed at which these fluctuations commenced varied with the closing pressure of the injection valve.

3. The interval between the by-pass port opening and the cessation of discharge, or the cut-off lag, was decreased with the decrease of speed at the lower closing pressure tested.

4. The period of injection in fractions of a second increased and in degrees decreased with the decrease in speed.

In reference 2 the pressure before the nozzle was shown to vary with the pump-plunger velocity. It was shown there that, as the pump speed was decreased, the intensity of the pressure waves transmitted to the injection valve was decreased. Computations for the higher speeds of these tests by the Allievi method of treatment show the intensity of the primary waves to be of sufficient magnitude to open the injection valve with either no reenforcement by reflected waves or with only small reenforcement. As the speed was decreased, however, the number of reflected waves necessary to build up sufficient pressure to open the injection valve was increased. Taking the speed of a pressure wave as equivalent to 5.95×10^4 inches per second, the time required by a pressure wave to traverse the 34-inch injection tube was 5.7×10^{-4} seconds. This time interval in pump degrees is given in Table I for the different pump speeds. For the closing pressure of 2,500 pounds per square inch (figs. 3 and 5) it is seen that (1) at the pump speed of 1,000 r. p. m. the valve opened shortly after the first wave front reached the injection valve, (2) at 750 r. p. m. the valve did not open until after the first wave was reflected from the injection valve but did open before the reflected wave again reached the valve, (3) at 470 r. p. m. sufficient time elapsed for the first wave to be reflected from the injection valve and reenforce the primary pressure wave twice before the valve opened, and (4) at 215 r. p. m. the first pressure wave was reflected and reenforced the primary three times before the valve was opened. For the injection-valve closing pressure of 500 pounds per square inch (figs. 4 and 6), (1) at 1,000 and 750 r. p. m. the valve opened when the first pressure wave reached the injection valve, (2) at 470 r. p. m. the valve did not open until after the first wave reached the injection valve, but before the reflected wave reached the valve again, (3) at 215 r. p. m. the first pressure wave was reflected and reenforced the primary wave two times before the valve opened, and (4) at 108 r. p. m. the first wave was reflected and reenforced the primary nearly five times before the valve opened.

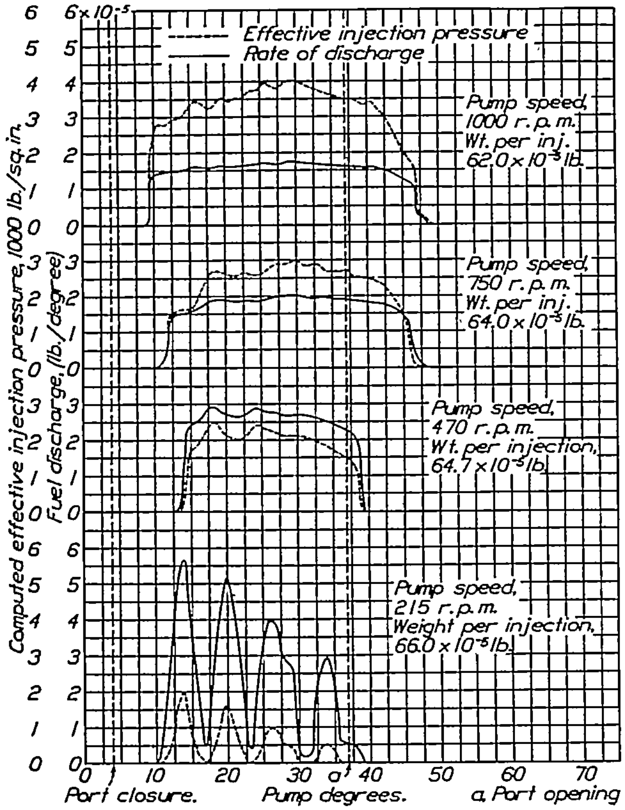

FIGURE 3.—Rates of discharge and effective injection pressures with various pump speeds: Throttle setting, full load; injection-tube length, 34 inches; inside tube diameter, 0.125 inch; injection-valve closing pressures, 2,500 pounds per square inch; orifice diameter, 0.022 inch

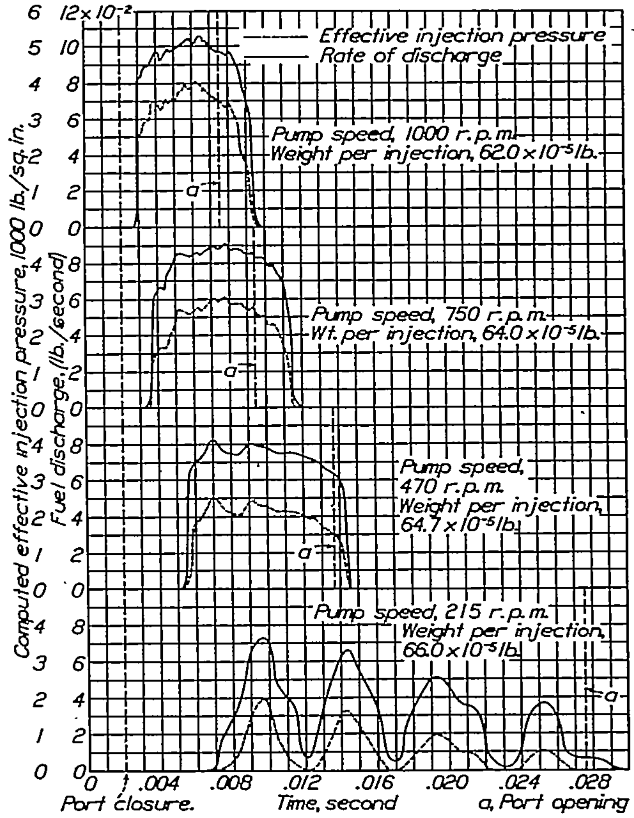

FIGURE 5.—Rates of discharge and effective injection pressures with various pump speeds: Throttle setting, full load; injection-tube length, 34 inches; inside tube diameter, 0.125 inch; injection-valve closing pressure, 2,500 pounds per square inch; orifice diameter, 0.022 inch

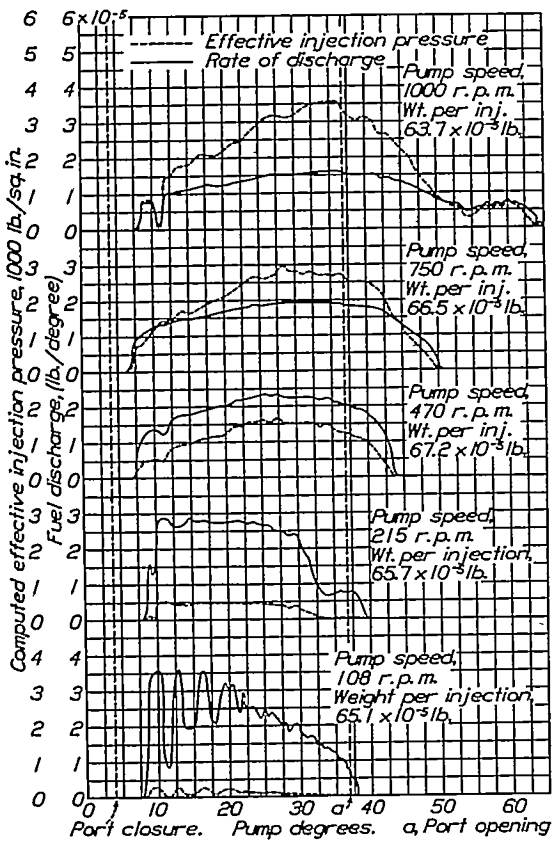

FIGURE 4.—Rates of discharge and effective injection pressures with various pump speeds: Throttle setting, full load; injection-tube length, 34 inches; inside tube diameter, 0.125 inch; injection-valve closing pressure, 500 pounds per square inch; orifice diameter, 0.022 inch

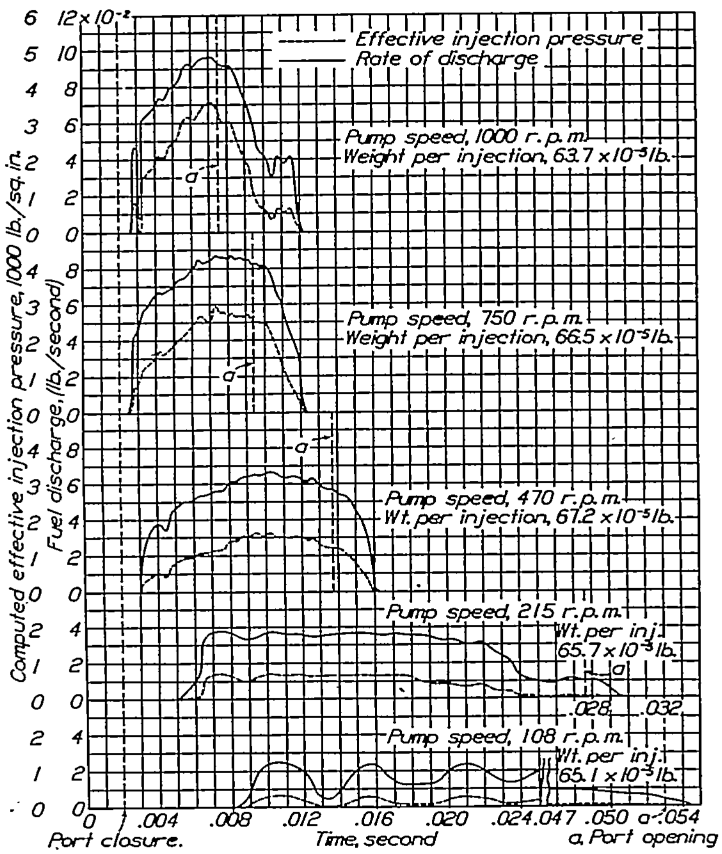

FIGURE 6.—Rates of discharge and effective injection pressures with various pumps speeds: Throttle setting, full load; injection-tube length, 34 inches; inside tube diameter, 0.125 inch; injection-valve closing pressure, 500 pounds per square inch; orifice diameter, 0.022 inch

TABLE I

EQUIVALENT INTERVAL IN PUMP DEGREES REQUIRED FOR A PRESSURE WAVE TO TRAVERSE A 34-INCH TUBE AT DIFFERENT PUMP SPEEDS

L, tube length = 34 inches.
S, velocity of a pressure wave = 59,500 inches per second.
$L/S = 5.72 \times 10^{-4}$ seconds.

Speed r. p. m.	108	215	470	750	1,000
Equivalent L/S in degrees	0.37	0.74	1.62	2.58	3.44

For the same pump, it was shown experimentally and analytically in reference 2 that the reflected pressure waves from the valve end of the tube, after the valve opened, changed from a positive to a negative value as the speed was reduced from 750 to 470 r. p. m. It can be readily shown by the same analysis that as the speed was reduced from 1,000 to 750 r. p. m., the intensity of the reflected positive pressure waves was decreased; at 470 r. p. m. the reflected waves changed to comparatively weak negative waves; and, as the speed was reduced further, the reflected negative waves increased in intensity.

The results shown in Figures 3 to 6 confirm the previous test and the analytical results. For the speeds of 1,000 and 750 r. p. m., the rate of discharge and the computed effective pressures increased steadily until shortly after the pump plunger started riding over the decelerating part of the cam lift. The same phenomenon appears to be true for the 470 r. p. m. at 500 pounds per square inch valve-closing pressure. (Figs. 4 and 6.) For 470 r. p. m. at the valve-closing pressure of 2,500 pounds per square inch and for the lower speeds at both closing pressures, however, the rate of discharge and the computed effective injection pressures reached the greatest values soon after the opening of the injection valve.

At the lower speeds, computed effective pressures below the closing pressures of the valve are shown, caused by restriction to the flow through the orifice because of the near approach of the valve stem to its seat. As was stated previously, sufficient pressure was built at the lower speeds of these tests to open the valve either by successive reenforcements of the primary pressure waves, wholly reflected initial waves, or by higher intensity waves emanating from the pump as the plunger was riding on to a steeper portion of the cam. Owing to the negative reflected waves and the weak primary waves at these speeds, however, the pressure behind the valve could not be maintained above the closing pressure of the valve. The result was that the injection-valve stem was forced by the spring toward its seat, thus throttling the flow. Following this restriction, and especially if there was an instantaneous complete closure of the valve, sufficient pressure was again built up to force the stem away from its seat. For the speed of 215 r. p. m. at a valve-closing pressure of 500 pounds per square inch, there was a restriction to the flow through the orifice, but the pressure fluctuations were sufficient to maintain the valve stem floating near its seat. For the speed of 215 r. p. m. at closing pressure of 2,500 pounds per square inch, and the speed of 108 r. p. m. at closing pressure of 500 pounds per square inch, the valve stem was alternately seated and forced away from its seat until the uncovering of the by-pass ports at the pump ended the discharge.

No definite trend in the cut-off lag is shown in Figure 5 with the valve at 2,500 pounds per square inch closing pressure. The explanation of this phenomenon can be found in the fact that there was not much difference between the closing pressure of the valves and the pressure behind the valve at the point of cut-off at the pump. When such small differences in pressure exist there can be little or no decompression period. The cut-off lag therefore depended largely on the lag of the check-valve and injection-valve closures, which are independent of the pump under these conditions.

In Figures 4 and 6 the cut-off lag is shown to decrease with the decrease of speed. This decrease was larger between the speeds of 1,000 and 470 r. p. m. for which the difference between the closing pressure of the valve and the pressure in the tube at the point of cut-off was comparatively large. This lag can be attributed to the inertia lag in the closing of the injection valve and to the trapping of fuel under pressure in the tube by the closure of the check valve. At the higher speeds the inertia lag in the valve closing was of appreciable magnitude, for the needle was some distance away from its seat. Trapping of fuel under pressure in the tube was possible because the difference of pressure in the tube and the primary pressure was so large that the check valve was forced to close quickly before the pressure in the tube was reduced appreciably. The pressure under which the fuel was trapped depended on the pump speed as the variation in the cut-off indicates.

There was only a small difference in the weights discharged per injection at full load for the whole speed range tested. This feature of the pump is especially desirable with power plants operating at low speeds and full load. These conditions are met in the operation of motor trucks, where conditions of traffic necessitate running at low engine speeds a large portion of the time. The disadvantage in the operation at low speeds and at full load is that, because of the lower effective pressure behind the valve, the spray delivered lacks penetrating ability and its atomization and dispersion are poor enough to result in a lowered combustion efficiency. In aircraft engines this fluctuation of discharge at low speeds is not so objectionable because the low speeds are only used for idling and starting. Under these conditions the combustion efficiency of the engine is not of importance except for the disagreeableness of a smoky exhaust. This disadvantage,

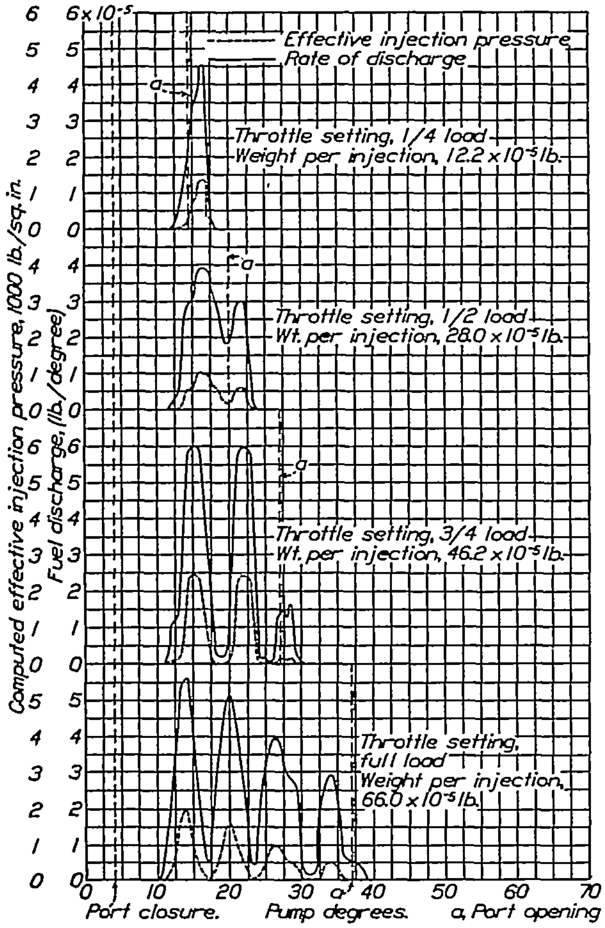

FIGURE 7.—Rates of discharge and effective injection pressures with various throttle settings: Pump speed, 215 r. p. m.; injection-tube length, 34 inches; inside tube diameter, 0.125 inch; injection-valve closing pressure, 2,500 pounds per square inch; orifice diameter, 0.022 inch

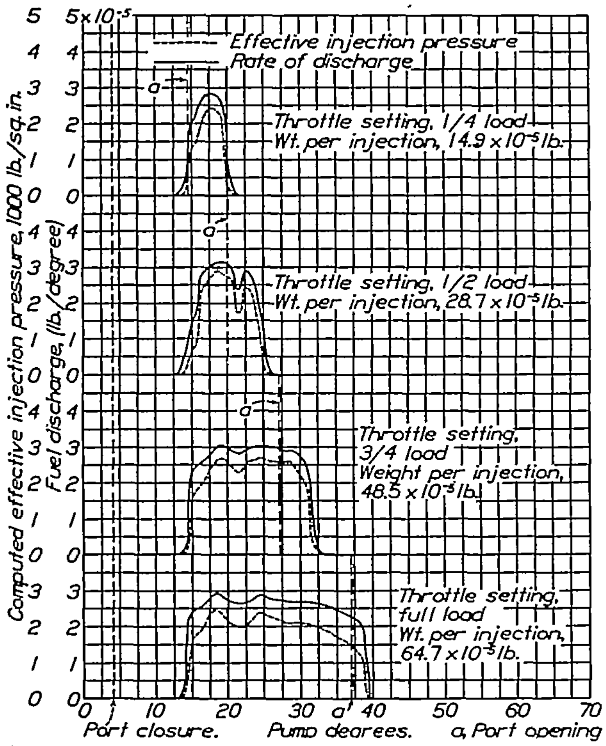

FIGURE 8.—Rates of discharge and effective injection pressures with various throttle settings: Pump speed, 470 r. p. m.; injection-tube length, 34 inches; inside tube diameter, 0.125 inch; injection-valve closing pressure, 2,500 pounds per square inch; orifice diameter, 0.022 inch

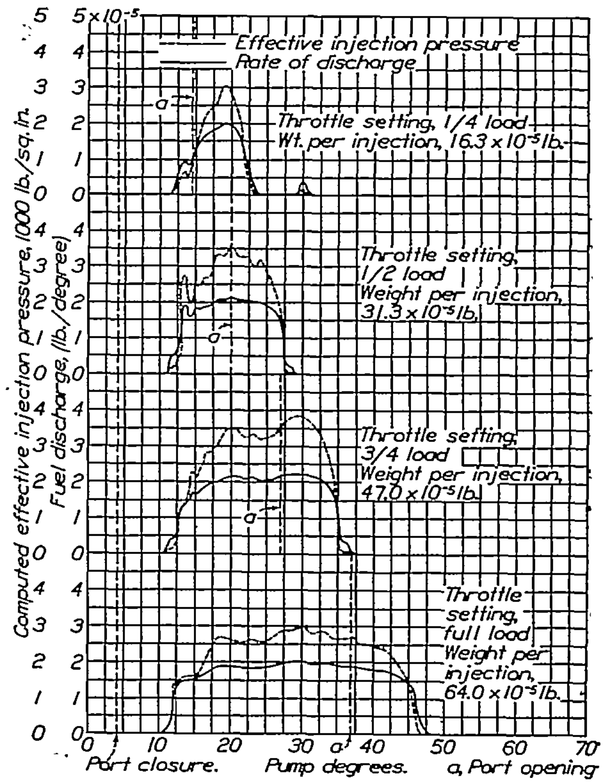

FIGURE 9.—Rates of discharge and effective injection pressures with various throttle settings: Pump speed, 750 r. p. m.; injection-tube length, 34 inches; inside tube diameter, 0.125 inch; injection-valve closing pressure, 2,500 pounds per square inch; orifice diameter, 0.022 inch

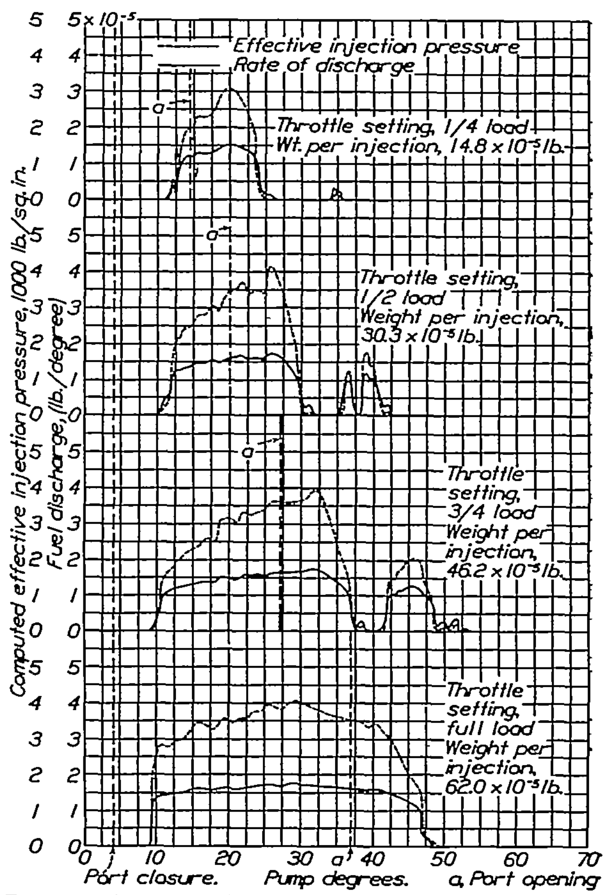

FIGURE 10.—Rates of discharge and effective injection pressures with various throttle settings: Pump speed, 1,000 r. p. m.; injection-tube length, 34 inches; inside tube diameter, 0.125 inch; injection-valve closing pressure, 2,500 pounds per square inch; orifice diameter, 0.022 inch

however, can be somewhat overcome by raising the opening and closing pressure of the valve. As was shown previously, the pressure behind the valve at the lower speeds remains, for the greater part of injection, very near the closing pressure of the valve. If the pressure falls below the closing pressure of the valve, the valve stem approaches its seat and a throttling takes place, which is followed by a pressure building up above the closing pressure of the valve as shown by the results. The pressure behind the valve therefore is kept slightly above or below the closing pressure of the valve and there is only a reduction in the effective opening through which discharge takes place.

Rates of discharge at different throttle settings.—In Figures 7 to 10 the rates of discharge at different throttle settings and speeds are given. At part loads, throttle settings which gave weights per injection in exact proportion to full load at 750 r. p. m. were used.

At part loads and at higher speeds, the cut-offs took place at an interval at which high-intensity pressure waves were propagated to the injection valve; the plunger was riding on a steep portion of the cam and there were high-intensity reflections reenforcing the primary wave. At the lower speeds, as was explained previously, the pressure difference between the closing pressure of the valve and the hydraulic pressure was small throughout the injection period. The valve stem, therefore, was close to its seat throughout the period of injection. Consequently, when the cut-off at part load took place at the pump and a rarefaction wave was transmitted to the injection valve, the delay in stopping the discharge at the lower speeds was small as compared to that at the higher speeds.

At 1,000 r. p. m. (fig. 10) there were secondary discharges after cut-off at part loads. These discharges were probably due to the high-intensity wave reflections which persisted in the tube after the by-pass port was uncovered and the injection and check valves had closed. The duration of these discharges was greater at one-half and three-fourths load because the point of cut-off at the pump took place when the plunger was advancing at or near its maximum velocity. This phenomenon of secondary discharges was also observed previously with this injection system at full load and at 750 r. p. m., and is shown in the results presented in references 2 and 7.

Up to the point of cut-off, there was a small difference in the rates of discharge and in the effective pressures regardless of the throttle setting. There was only a small increase in the injection lag as the load was reduced at 215 and 750 r. p. m. The maximum variation between the loads for the speeds tested is 5, 10, and 25 per cent for the three-fourths, one-half, and one-fourth load, respectively.

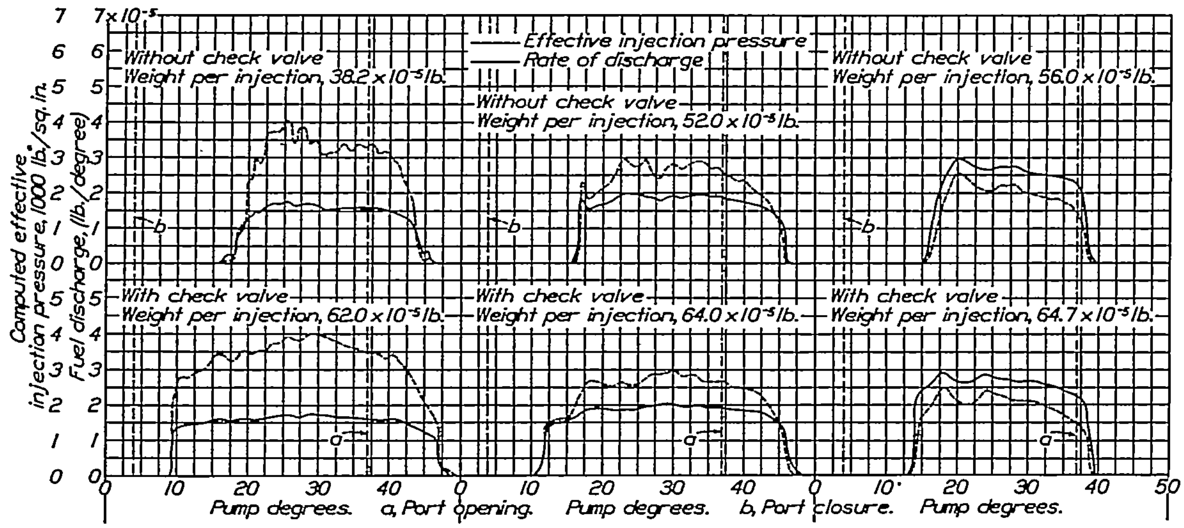

FIGURE 11.—Rates of discharge and effective injection pressures with and without check valve: Pump speed, 1,000 r. p. m.; throttle setting, full load; injection-tube length, 34 inches; inside tube diameter, 0.125 inch; injection-valve closing pressure, 2,500 pounds per square inch; orifice diameter, 0.022 inch

FIGURE 12.—Rates of discharge and effective injection pressures with and without check valve: Pump speed, 750 r. p. m.; throttle setting, full load; injection-tube length, 34 inches; inside tube diameter, 0.125 inch; injection-valve closing pressure, 2,700 pounds per square inch; orifice diameter, 0.022 inch

FIGURE 13.—Rates of discharge and effective injection pressures with and without check valve: Pump speed, 470 r. p. m.; throttle setting, full load; injection-tube length, 34 inches; inside tube diameter, 0.125 inch; injection-valve closing pressure, 2,500 pounds per square inch; orifice diameter, 0.022 inch

Rates of discharge with and without check valve at the pump outlet.—In Figures 11, 12, and 13 the results obtained with and without the check valve at the pump outlet are shown for pump speeds of 1,000, 750, and 470 r. p. m. A comparison of the curves shows that for the same speed, both the injection period and the weight discharged per injection were smaller without than with the check valve, and as the speed was increased the weight discharged per injection without the check valve decreased while that with the check valve remained approximately constant. The explanation for these differences can be found by examining the conditions existing in the injection system at the commencement and end of discharge, resulting in longer injection lags and shorter cut-off lags when no check valve was used at the pump.

The longer injection lag without the check valve can be explained by the fact that the pressure in the tube between injections was reduced to the pump intake pressure, as compared to the tube pressure with the check valve which was very much higher because of the trapping of the fuel by the closure of the check valve. As a consequence of this reduction in the initial pressure without the check valve, the injection valve did not open until the primary pressure wave was reenforced by the initial reflections once at 1,000 and 750 r. p. m. and almost three times at 470 r. p. m. The difference in the injection lags, with and without the check valve at 1,000 and 750 r. p. m., was equivalent to exactly the time necessary for a pressure wave to traverse the tube twice, showing that when no check valve was used an additional reenforcement of the primary wave by the initial reflections was required before sufficient pressure was built up to open the injection valve at these speeds.

The cut-off lags without the check valve at 1,000 and 750 r. p. m. were slightly longer than was necessary for the rarefaction impulse originating at the pump end by the uncovering of the by-pass ports to reach the valve end of the tube plus the inertia lag in the injection-valve closing as computed by the method given in reference 8. The observed longer cut-off lags than anticipated indicate that the uncovering of the by-pass ports did not release the pressure in the tube instantaneously. At 470 r. p. m. the discharge closed immediately as the rarefaction wave from the pump reached the valve end of the tube, for at the point of cut-off the pressure before the valve was very near the closing pressure of the valve.

Rates of discharge with different orifice diameters.—In Figure 14 the rates of discharge at different orifice diameters are given. Considerably higher pressures were built behind the valve with the smaller sizes of orifices. With the 0.008, 0.015, and 0.022 inch orifices the pressure, consequently the rates of discharge, continued to increase until sometime after the plunger began the decelerating motion of its upstroke; the smaller the orifice size the longer this pressure rise continued. For the 0.031 and 0.042 inch orifices the pressure and rate of discharge variation conformed more closely to the acceleration and deceleration of the pump plunger. The effective injection pressures decreased with the increase of the orifice size.

Owing to the small discharge opening with the smaller orifices, the amount of the pressure-wave energy utilized in the discharge of the fuel through the orifice was smaller than with the larger orifices. With the smaller orifices tested the pressure-wave reflections (reference 2) from the injection valve were of high intensity. The effects of these reflections and the subsequent reenforcements of the primary wave are shown clearly in the curve of the effective pressure behind the valve with the 0.008-inch orifice. There was a sudden increase in the effective pressure with every arrival of the reenforced wave at the injection valve. As the orifice size was increased, however, more of the pressure-wave energy was utilized in the discharge of the fuel, and the intensity of the reflected waves was decreased.

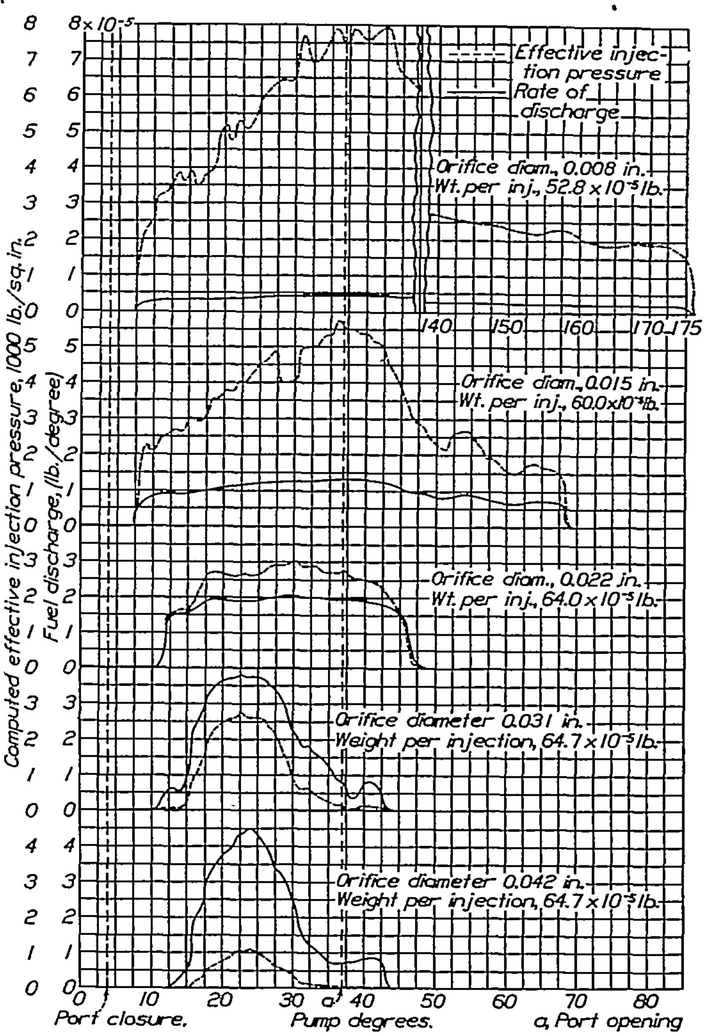

FIGURE 14.—Rates of discharge and effective injection pressures with various orifice diameters: Pump speed, 750 r. p. m.; throttle setting, full load; injection-tube length, 34 inches; inside tube diameter, 0.125 inch; injection-valve closing pressures, 2,500 pounds per square inch

The prolonged discharge after cut-off with the smaller orifices was caused by the fuel under pressure that was trapped in the fuel line by the closure of the check valve. Computations using the method given in reference 2 show that the actual time for the prolonged discharge after cut-off agrees with the computed decompression period for the 0.015-inch orifice. Similar computations, however, account for only one-half of the decompression period for the 0.008-

inch orifice. During the tests with the 0.008-inch orifice the injection tube was observed to heat excessively. Although no measurements of the increase in the oil temperature and no computations were made, taking into account the compressibility of the fuel, which is known to change rapidly with temperature (reference 9), the difference between the computed and the actual decompression period can be safely attributed to the changed physical properties of the fuel in the injection tube.

The trapping of the fuel under pressure in the tube after cut-off explains the increased injection lag with the increase of orifice size. With the smaller orifices, the injection valve was given an opportunity to approach its seat slowly, and, when closed, leave the fuel in the tube between injections at a pressure very near the closing pressure of the valve. With the larger orifices, the pressures before the valve immediately preceding cut-off were near the closing pressures, and at the interval between cut-off and the closure of the injection valve sufficient fuel was discharged through the orifice and past the check valve to leave the pressure in the tube between injections much lower than the closing pressure of the valve. At the start of injection with the larger orifices, therefore, sufficient time elapsed until enough pressure was built up to open the valve.

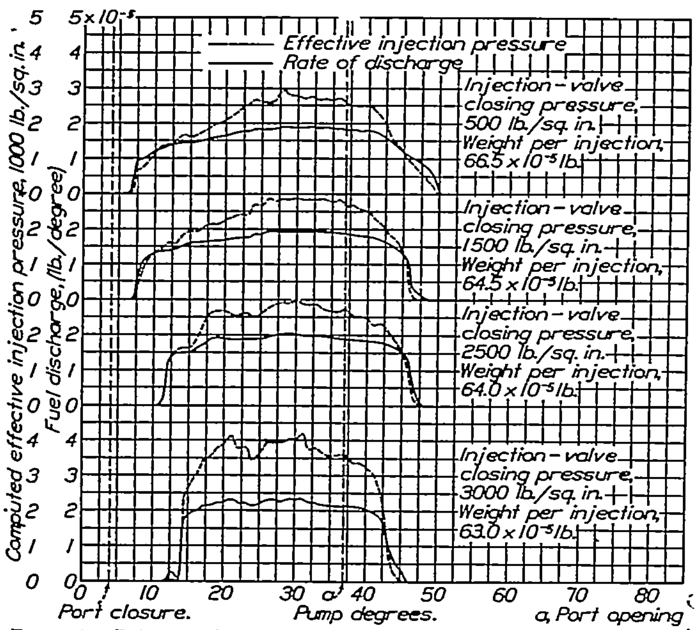

FIGURE 15.—Rates of discharge and effective injection pressures with various injection-valve closing pressures: Pump speed, 750 r. p. m.; throttle setting, full load; injection-tube length, 34 inches; inside tube diameter, 0.125 inch; orifice diameter, 0.022 inch

There was an 18 per cent difference in the total weight discharged per injection between the 0.042 and 0.008 inch orifices, a 7 per cent difference between the 0.042 and 0.015, but only 1 per cent difference between the 0.042 and 0.022 inch.

From the results of Figure 14 it is seen that, for a given pump capacity, normal operating speeds, injection-valve design, and period of injection (which conditions are fixed by the engine capacity and design) the useful range of effective orifice areas is limited. In the selection of a suitable effective discharge orifice area, it would be advisable to make a set of computations for the mean pressure before the discharge orifice with different orifice areas according to equation (1) in reference 2. Graphs of injection pressure against orifice areas at the normal operating speeds, loads, and other important conditions may be prepared. A suitable effective orifice area may then be selected that gives a spray with the desired penetrative power, dispersion, and atomization.

Rates of discharge at different valve-closing pressure.—In Figures 3 to 6 the results with two closing pressures and different speeds were given. In Figure 15 the rates of discharge at various closing pressures at a speed of 750 r. p. m. are shown. As the valve-closing pressure was increased, the injection lag increased and the cut-off lag decreased, resulting in a shorter period of injection. In addition, the effective injection pressures and the period during which higher injection pressures existed behind the valve increased.

At the valve-closing pressures of 500 and 1,500 pounds per square inch, the valve opened immediately as the first pressure wave reached that end of the tube. At the closing pressure of 2,500 pounds per square inch, the valve opened after the initial pressure wave was reflected from the injection valve. At the closing pressure of 3,000 pounds per square inch, the valve did not open until after the complete reflection of the initial pressure wave reenforced the continuously emanating primary wave and had reached the injection valve again.

The smaller cut-off lag at the higher closing pressures can be explained by the fact that the difference between the pressure behind the valve and the closing pressure of the valve was small. Therefore, the valve closed quickly as soon as a rarefaction reached that end of the tube after the opening of the by-pass ports at the pump. At the lower closing pressures, this difference in pressure was comparatively large; as was shown previously, fuel under pressure was trapped in the line with a consequent slower closing of the valve.

Computations according to reference 2 show that because there is no energy used for injection through the orifice during the early part of the effective pump stroke higher pressures are built behind the valve with higher valve-closing pressures. Owing to the larger injection lag when higher closing pressures are used, the pressure waves propagated to the injection valve after the valve opens are reenforced by the wholly reflected portions of the initial primary. In addition, the waves emanating from the pump at that point are of a higher pressure intensity, for the plunger rides on a steeper portion of the cam. Thus, as the experimental results show, a higher injection pressure is maintained behind the injection valve.

Owing to this higher injection pressure, with the highest closing pressure tested in which the injection period was 30 per cent smaller, the pump was able to deliver a total amount of fuel per injection only 5 per cent less than was obtained when the lowest closing pressure of the valve was used.

Rates of discharge with different tube lengths.—In Figure 16 the results obtained with tubes of four different lengths are shown. In previous investigations of the committee (reference 10) for injection lags with common-rail injection system, an increase of injection lag was shown with the increase of tube length. This variation was normally what should have been expected, if all other conditions were kept constant throughout the tests and if the inside diameter of the injection tube was of the proper size so that there were no appreciable losses to friction or compressibility. In the present tests with the pump injection system, the injection lags did not increase appreciably with the increase of the tube length.

The number of pump degrees at 750 r. p. m. for a pressure wave to travel through the length of the 20-inch tube is 1.5°, of the 34-inch tube 2.6°, of the 53-inch tube 4°, and of the 74-inch 5.6°. With the exception of the 20-inch tube, the injection valve opened some time after the first reflection from the injection valve but before the primary wave was reenforced by any reflections. With the 20-inch tube, the time required was long enough for the initial reflected wave to reenforce the primary twice before sufficient pressure was built behind the valve to open it. The possibility is suggested that the pressure in the line between injections was lower with the shorter tube than with the longer. The inertia lag of closing of the injection valve and the check valve being approximately the same for all tubes, the amount of fuel that left the tube during the interval of closing the by-pass ports and closing of both these valves was proportionately larger with the shorter tubes, thus leaving lower pressures between injections in the tube.

The slightly increasing cut-off lags with the increase of tube length can be attributed to the longer time required for the rarefaction wave (propagated from the pump after the by-pass ports are uncovered) to traverse the longer tubes.

Rates of discharge with different tube diameters.—Figures 17 and 18 give the rates of discharge with tubes of different diameters at speeds of 750 and 470 r. p. m., respectively. Thus far an injection tube was used having an inside diameter of 0.125 inch. As was shown in reference 2, this diameter was above the critical tube diameter for the conditions of these tests. With the use of the smaller tube diameters a disturbing factor was introduced, that is, the energy losses in the tube caused by the turbulent flow. It is difficult to form an equation representing the actual condition with a smaller tube because the friction losses become appreciable.

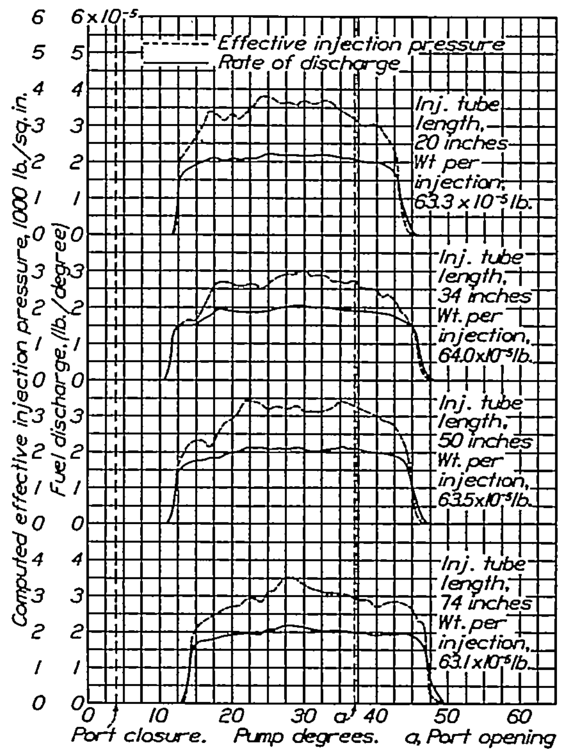

FIGURE 16.—Rates of discharge and effective injection pressures with various injection-tube lengths: Pump speed, 750 r. p. m.; throttle setting, full load; inside tube diameter, 0.125 inch; injection-valve closing pressure, 2,500 pounds per square inch; orifice diameter, 0.022 inch

At 750 r. p. m. (fig. 17), sufficient pressure was created to open the valve some time after the initial pressure wave was reflected from the valve end of the tube, and slightly before or a little after the reenforced primary reached the valve again, with all tube diameters tested. After the opening of the valve, however, because of the resistance losses in the tubes having inside diameters below the critical diameter, sufficient pressure could not be maintained to keep the injection valve fully open. Immediately after the opening, the discharge decreased, which indicates that probably the valve stem was forced toward its seat with a consequent throttling of the flow.

At 470 r. p. m. (fig. 18) larger injection lags and short cut-off lags are shown with the increase of tube diameter. Computations according to reference 2 for the 0.125 and 0.187 inch tubes indicated that because of the low speed, the reflected pressure waves during the interval of injection were all negative and that the pressure behind the valve actually dropped far below the closing pressure of the valve before the effects of

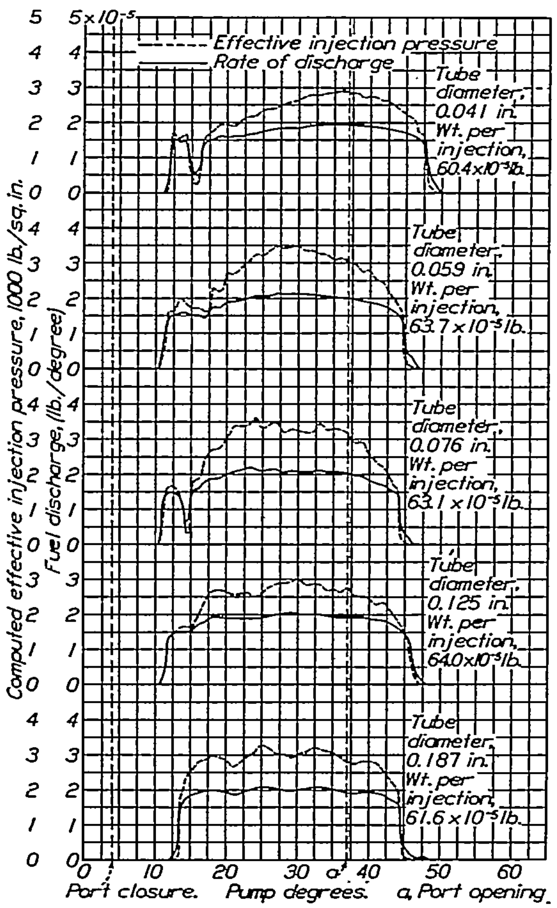

FIGURE 17.—Rates of discharge and effective injection pressures with various inside tube diameters: Pump speed, 750 r. p. m.; throttle setting, full load; injection tube length, 34 inches; injection-valve closing pressure, 2,500 pounds per square inch; orifice diameter, 0.022 inch

the by-pass port closure were felt at the injection valve. The effect of friction losses on the rate of discharge with the 0.041-inch tube diameter are apparent by the ragged appearance of the curve. Nevertheless, because of the smaller losses to compressibility, the discharge did not end until after the rarefaction, due to opening of the by-pass ports at the pump, reached the injection valve.

CONCLUSIONS

The rates of discharge confirm previous analytical and experimental results for pressure variation before the injection valve with a fuel pump injection system.

The rate of discharge in weight of fuel discharged per unit time increased with the increase of pump speed.

Up to the point of cut-off, there was a small difference in the rates of discharge, irrespective of the throttle setting at any one speed.

The rates of discharge conformed approximately to the contour of the cam only with the larger size orifices. With the smaller orifices, the high-intensity pressure wave reflections and reenforcements altered the rates of discharge to a different conformation. There was evidence of the flow being throttled by the injection-valve stem when orifices were used that were larger in diameter than those for which the pump was normally designed.

With the increase of the opening and closing pressure of the injection valve, there was a decrease in the total injection period and an increase in the interval during which higher effective pressure prevailed before the injection valve.

The tube length was found to have little effect on either the rate of discharge, injection period, or injection lag with the injection system and conditions tested.

Small tube diameters caused fluctuations in the rates of discharge.

At part throttle settings the load variation between different speeds was larger than at full throttle settings, and increased with the decrease of load.

The general conclusion from these tests is that a single pump design can satisfy the several requirements of many engines, having approximately the same load

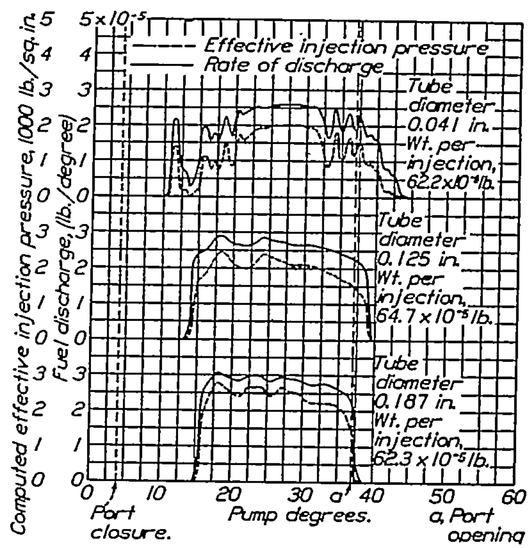

FIGURE 18.—Rates of discharge and effective injection pressures with various inside tube diameters: Pump speed, 470 r. p. m.; throttle setting, full load; injection-tube length, 34 inches; injection-valve closing pressure, 2,500 pounds per square-inch; orifice diameter, 0.022 inch

demands, by using a suitably designed injection valve and other integral parts of the injection system; and that to obtain the best performance from an engine the injection system must be designed as a unit for any specific engine design.

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NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS,
LANGLEY FIELD, VA., *May 15, 1932.*

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