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REQUIREMENTS FOR UNIT FUEL-INJECTION SYSTEMS

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SUMMARY

A unit injector was operated under various test conditions with a cam outline giving a high rate of plunger displacement. The rate of discharge of the unit injector followed the plunger displacement for the outwardly opening injection valve (open nozzle) except under conditions of high fuel pressures when the effect of fuel compressibility decreased the rate of discharge as the pressure increased. The rate of discharge for the inwardly opening injection valve (closed nozzle) did not directly follow the plunger displacement. The initial rate of discharge was independent of operating variables for the closed nozzle but varied with the valve-opening pressure and the rate of plunger displacement at the port closing for the open nozzle.

INTRODUCTION

The multiplicity of injection system designs for internal-combustion engines have all been developed for the same purpose; the injection of a controlled fuel charge into the combustion space. The timing, the quantity, the atomization, and the dispersion of the fuel charge are all controlled in part by the injection system. The different injection systems are not equally efficient in producing a desired result because each type has its own pecularities.

The injection characteristics are controlled by a considerable number of interrelated variables. Rate-of-discharge tests performed on various types of injection systems have shown that injection irregularities, such as oscillating discharges and secondary injections, can be explained by the pressure-wave phenomena occurring in the injection tube. A discharge rate that fluctuates widely between the maximum rate and a very low rate is called an oscillating discharge. Fluctuations that are less violent in character produce pulsating rates of discharge. A secondary discharge occurs separately from the controlled discharge and after the controlled discharge has been terminated by the opening of the bypass ports.

The oscillating discharges occur with a differential-area injection valve and result from pressure variations at the injection orifice. Various combinations of variables that produce a restriction to fuel flow or distinct pressure pulsations with resulting oscillating discharges may include: small injection-tube diameter, slow plunger displacement, high valve-opening pressure, low pump speed, and large orifice diameters.

The appearance of a secondary discharge depends primarily upon the injection-tube length, which causes a time interval between the release of pressure at the injection valve and the reappearance of a pressure wave at the injection valve. The intensity of the pressure wave trapped in the injection tube must be sufficient to open the injection valve. Injection variables usually producing secondary discharges are: high pump speeds, high valve-opening pressure, and small orifice diameters. All these conditions increase the intensity of the reflected pressure waves. The secondary discharges appear after cut-off at the pump because of reflection of the initial pressure wave from the closed pump check valve back to the injection valve.

Reducing the length of the injection tube should reduce the time interval between the pressure waves that cause secondary discharges and should result in less variation between the build-up of pressure at the pump plunger and at the orifice. With the discharge orifice attached directly to the pump chamber, the rate of pressure rise should be controlled by the rate of plunger displacement. A unit injection system combines the injection pump and the injection valve in one unit with practical elimination of the fuel-passage length between them. This feature is the primary constructional difference between unit injectors and conventional injection systems.

Rate-of-discharge tests of a simulated unit injection system have been reported in reference 1. The important conclusions presented in that paper are: That the secondary discharges were eliminated, that the rate of discharge changed with the type of injection valve, that the rate of discharge for an open nozzle was controlled by the rate of plunger displacement, and that the rate of discharge of a differential-area valve did not follow the rate of plunger displacement.

Actual control of the rate of discharge by controlling the rate of plunger displacement is the outstanding advantage of the unit injector. Various rates of discharge can be obtained by interchange of properly designed cams. In order to check the extent of this control at high pressures and high rates of plunger displacement, rate-of-discharge tests were made on a unit injection system under extreme operating conditions. The results of the tests should be applicable to the design of cams having accurately predetermined rates of discharge for the purpose of checking the effect of the rate of discharge on engine performance.

METHODS AND APPARATUS

The requirements of a unit injector as presented herein are also, in part, the requirements of a conventional injection system. They are based on rate-of-discharge tests obtained with a General Motors model R-201 unit injector having a 3/8-inch-diameter plunger and a 3/4-inch stroke. This unit injector (fig. 1) has been described in various publications. (See reference 2.) Two plunger-and-sleeve designs having a lapped helix, the rotation of which varied the effective stroke for fuel-quantity variation, were used. One set (plunger-and-sleeve design 1) used round ports having an area of 0.0048 square inch in the sleeve and had a varying start and cut-off of injection with change in throttle position. Plunger-and-sleeve design 2 used helical ports in the sleeve, corresponding to the throttle helix on the plunger, and had a constant start of injection with varying cut-off. The port areas in plunger-and-sleeve design 2 were 0.0109 square inch for the inlet port and 0.0141 square inch for the bypass port. Figure 2 shows the variation in port area with plunger lift for the two plunger-and-sleeve designs. A greater initial increase in the rate of area change for plunger-and-sleeve design 2 can be obtained by using the modified form obtained by filing the round corners. This alteration can easily be made and is necessary only on the opening edge of the port.

Two fuel connections to the unit injector permitted constant circulation of the fuel. A primary pressure of 50 pounds per square inch was used except at the higher speeds and loads when a primary pressure of 100 pounds per square inch was necessary to prevent a rapid drop in the fuel quantity discharged as the speed was increased.

The cam outline used was calculated to give, for an open nozzle at a constantly increasing rate of discharge, an injection period of 15 cam degrees for 0.00100 pound of fuel per injection. The cam outline was designed to give a very slow closing of the ports (for plunger-and-sleeve design 2); with the plunger motion utilized for injection occurring after this slow port closing. With plunger-and-sleeve design 1, the port closing occurred at various plunger velocities, depending upon the throttle setting. The plunger velocity in inches per cam degree can be obtained from the curves of rate of plunger displacement by dividing the rate of plunger displacement by the constant 0.00339.

Several designs of injection valve (fig. 3) were tested. In this report the closed nozzle consists of an orifice closed by a differential-area inwardly opening valve stem (valve A), and the open nozzle is an orifice closed by an outwardly opening valve sealing the fuel passage (valves B, C, and D). The fuel-passage area of valve A was equivalent to a 0.056-inch-diameter passage and of valves B, C, and D, to a 0.09 4-inch-diameter passage. For valve A_1, the area was increased three times in order to investigate the effect of restriction to the fuel flow through the injection valve.

The apparatus used in obtaining the rate-of-discharge curves has been described in reference 3. Operation of the equipment at high speeds was particularly noisy. The noise level appeared to be as high as that due to operation of a single-cylinder test engine. The noise level of the unit-injector equipment when used on an engine would, however, not increase the noise level above that due to the engine-valve mechanism and other engine noise sources. Data were taken at pump intervals of $1/2^{\circ}$ except when very little change in rate occurred; for such cases, data were taken at 1° intervals. The data were connected by straight lines, and fairing the curves was not attempted. The fuel quantity per injection was determined from the number of cycles required to inject one-half pound of fuel. This fuel quantity was checked from the area of the rate-of-discharge curves (figs. 4 to 11), which indicated a precision of within ±5 percent of the fuel weight. The dotted lines on the rate-of-discharge curves indicate the rate of plunger displacement, which was obtained by taking tangents to the lift curve. The positions of port closing and opening are indicated by the vertical lines at the extremes of the rate of plunger displacement. In figures 7 and 11, these positions are indicated for the various throttle settings by the vertical dashes on the base line or on the curve of rate of plunger displacement. These positions and the plunger lift were obtained statically and do not exactly conform to the dynamic positions of the experimental rate-of-discharge curves.

RESULTS AND DISCUSSION

The Injection Valve

It is difficult to say which part of the unit injector is most important. Each part has control over certain characteristics of the injection rate at particular times during the injection cycle. In this report, the characteristic effects on the rate of discharge resulting from changes in the design of the individual parts are discussed and rate-of-discharge curves are presented for the operating conditions shown in table I.

Open nozzle. - The type of injection valve used exercises probably the greatest control over the rate of discharge of the injection system of the variables tested. Parts (b), (c), and (d) of figure 4 give the rate-of-discharge curves for the open-nozzle types of injection valve shown in figure 3. With open nozzles, the rate of discharge closely followed the rate of displacement of the injection-pump plunger. An increasing or a decreasing rate of discharge could be maintained by control of the plunger velocity. The volume of fuel between the valve stem seat and the orifice must be kept at a minimum because, at high injection pressures, the expansion of the volume of fuel under the seat caused trailing at cut-off. The flow passage through the seat need be only slightly greater than the orifice areas. Cut-off with a ball check valve is as effective as with a poppet valve but, because of the larger volume of oil under the seat around the spring, the fuel trailed at cut-off. The most compact injection-valve arrangement tested was obtained by using a Belleville spring with a ball check valve (valve C, fig. 3). A guided poppet valve having a small lift and a stem stop (valve D, fig. 3) gave as sharp a cut-off as any obtained with a differential-area injection valve. No oscillating discharge was obtained with the open nozzle under any combination of variables.

Closed nozzle. - With the differential-area injection valve (fig. 4(a)), the rate of discharge did not directly follow the rate of plunger displacement and oscillating discharges might occur as easily as with long injection tubes. As previously mentioned, oscillating discharges are a result of pressure variations at the orifice. These pressure variations are caused by the oscillating valve stem (reference 4) and by the variable restriction at the valve-stem seat. If the rate of pressure rise at the injection valve does not take care of the pressure drop resulting from the valve lift and from the injection process, pressure variations at the orifice will probably occur with resulting oscillating discharges. With a closed nozzle, injection with a slow initial rate of plunger displacement usually results in oscillating discharges. The advantage of the closed nozzle is in its ability to build up a high initial rate. If a sufficiently high rate of displacement is used over the entire injection period, a very short period can be obtained.

The initial rate of discharge for the closed nozzle may be considerably higher than the rate of plunger displacement. This fact probably accounts for the poor engine performance of open nozzles when the two types are compared in the conventional injection system. For both nozzles to have comparable initial rates of discharge, the rate of plunger displacement should be increased for the open nozzle.

Valve-opening pressure. - The characteristic discharge rates of the two types of injection valve were not materially affected by valve-opening pressure (fig. 4 and reference 3). As the valve-opening pressure was increased, the injection period was slightly shortened owing to the additional compression of the fuel required. Because the injection valve must seal the injection system against the entrance of the combustion gases, the minimum valve-opening pressure of the open nozzle can be made much lower than that of the closed nozzle provided that the minimum valve-opening pressure is high enough to overcome the pressure built up by the resistance to fuel flow through the bypass ports at zero throttle settings. For example, valve A with a valve-opening pressure of 3,400 pounds per square inch would inject at a pump speed of about 750 rpm for plunger-and-sleeve design 1 at zero throttle setting. In order to make use of a lower valve-opening pressure or a higher operating speed, the bypass area would have to be increased to reduce the build-up of pressure in the plunger space.

Orifice diameter. - The orifice diameter is one of the controlling factors of the injection system (fig. 5). For a given rate of discharge, the injection pressure will vary inversely as the fourth power of the orifice diameter and directly as the square of the pump speed. Since the characteristics of the fuel spray depend, in part, upon orifice and pressure values, a summary of the desired spray characteristics would help to determine the orifice and the pressure conditions.

The maximum pressure developed from the use of a small total-orifice area will limit the maximum speed owing to excessive loading of the actuating mechanism. A graph of pressures calculated from the maximum theoretical rate of discharge for the cam used is presented for various orifice diameters and pump speeds (fig. 6). These theoretical maximum pressures are reduced at the orifice during injection because of the compressibility of the fuel, the leakage, and the pressure drop from resistance to flow through fuel passages.

Figure 5 shows the marked reduction in the rate of discharge that occurs with the pressure increase obtained by reducing the orifice area. The change in rate of discharge is explained by the compressibility of the fuel under pressure. Calculation of the instantaneous pressure at the discharge orifice, compressibility being considered but pressure waves being neglected (see reference 5) indicated that the calculated decrease in the rate of discharge obtained agreed closely with the experimental rate of discharge.

Flow passage. - The flow passages through the injection valve should be short to minimize pressure surges. The volume of fuel under compression should be a minimum for rapid cut-off and the volume of fuel between the valve seat and the orifices should be small for rapid pressure rise and fall to prevent dribbling.

The flow-passage area depends upon a low pressure drop for the open nozzles. Any resistance to flow means additional pressure rise in the pump chamber with resulting increased compression of the fuel and a direct decrease in the rate of discharge. For the closed nozzle, however, a threefold increase in the passage area, with a resulting reduction in the calculated pressure drop from 15 to 2 percent, did not effect a material change in the rate-of-discharge curve (fig. 7(a)).

The Injection Pump

Plunger-and-sleeve design. - The external sleeve diameter should be large in order to minimize possible distortion from the nonsymmetrical ports. It is important to watch any dimensional change that tends to increase the clearance because the leakage varies with the cube of the clearance. With the hollow plunger and the sleeve diameters used, the clearance increase is about 0.00006 inch for 20,000 pounds per square inch. The calculated percentage leakage due to this clearance is negligible. Also the rate-of-discharge curves do not fall off when close to the cut-off position, indicating that leakage is negligible even at this position. Because of the high plunger velocity, the lap length at 1° before cut-off may be more than 0.030 inch. Wear is an indeterminate factor and, for the injection of fuel under sustained high fuel pressure at low plunger velocity, the leakage could be an appreciable factor affecting the rate of discharge. The short length of lap sealing the high pressure decreases in length and the pressure and the clearances increase with an increase in effective plunger motion. These factors tend to increase the leakage. Although the fuel viscosity increases with pressure, the leakage increases under the pressure and the dimensional changes.

Under high pressures, the hydraulic surfaces of the plungers should be balanced to prevent side thrust against the ports. A high plunger velocity combined with side thrust may produce scoring from lack of lubrication or scuffing of the ports.

The flow-passage area used in the plunger and the sleeve limits the minimum injection valve-opening pressure and the rapidity of cut-off. A high rate of plunger displacement combined with small flow passages could build up sufficient pressure in the pump chamber to inject fuel after cut-off at the pump. Restricted flow area prolongs injection but, with the open nozzles used, no such prolonging of injection occurred owing to their adequate flow areas.

The rate of opening of the bypass ports is a factor in determining the rapidity of cut-off. Helical ports and poppet valves instead of round ports are simple means of obtaining a rapid initial increase in the rate of opening. (See fig. 2.) Figure 7(a) shows the effect of restriction in the injection-valve passage and figure 7(b) shows the effect of slow opening of the bypass ports. The effect of restriction at the bypass ports is evident (fig. 7(c)) with the closed nozzle under conditions of high pressure by definite relifting of the valve stem during cutoff.

The rate of port closing affects the initial rate of discharge of the open nozzle but does not affect the initial rate of discharge of the closed nozzle (figs. 8 and 9).

Separate inlet and bypass ports are prerequisites if pressure fluctuations are to be eliminated on the intake side of the pump. These pressure variations are the cause of cyclic variations in the fuel discharge that result from lack of filling the plunger space under identical conditions. A check valve placed on the intake side of the pump can be used to prevent the transmission of pressure surges through the intake tubing. Separate ports can also be used to advantage to circulate fuel through the pump to prevent vapor and air accumulation and to cool the unit.

The difficulty of filling the plunger space of injection systems having ported controls is increased at high pump speeds and temperatures by the vacuum formed during the return plunger stroke. Air is released from the fuel and, for the more volatile fuels, vaporization occurs before the inlet ports are opened. The air and the vaporized fuel may be trapped in the dead end of the plunger space and probably could not be removed by circulating fuel. High primary pressure can be used to help fill the plunger space, but a construction that will allow the plunger space to be filled during the down stroke of the plunger will eliminate one of the most potent causes of fuel-quantity irregularity.

Discharge check valve. - In the conventional system, a check valve is usually placed at the injection pump to retain a residual pressure in the injection tube between cycles. If extremely high pressures are obtained and a discharge check valve is used, injection would be prolonged beyond the cut-off at the pump from expansion of the compressed fuel between the check valve and the orifice. Such a condition also results when restriction to flow occurs at the bypass ports. Tests have shown (reference 6) that this valve causes secondary discharges. Its elimination prevented the secondary discharge but produced irregular cycles at high speeds. Because of the small volume of fuel in the unit injector, a pump check valve is unnecessary. Rate-of-discharge tests made with and without a pump check valve showed no difference in the injection characteristics.

Pump speed. - The rate of plunger displacement varies directly with the plunger velocity and with the square of the plunger diameter. With a high rotative speed, a high rate of discharge could be more easily obtained with a large plunger diameter and a small plunger motion, but the dynamic load on the operating mechanism would be high and the rate of opening of the ports, slow. A small-diameter high-velocity plunger could be used at extreme pressures but it would require auxiliary means of controlling the cut-off, as adequate port area could not be obtained through the small plunger. Mechanics would probably limit the maximum stroke and determine the minimum plunger diameter. For the same rates of plunger displacement, the unit injector will give a considerable increase in the rate of discharge over the conventional injection system.

An advantage of the unit injector lies in its ability to inject satisfactorily at high rotative speeds the actuating mechanism should therefore be light, consistent with a minimum of distortion. The mounting must be rigid to prevent strain from shock loading. The decrease in the rate of discharge with increase in speed, shown in figure 10, is accounted for by the effect of compressibility of the fuel.

Cam outline. - The change in injection characteristics desired to accompany a change in throttle setting will determine the use of a variable start or a variable stop or the use of a constant, an increasing, or a decreasing velocity portion of the cam. A slowly increasing initial rate of discharge at the start of injection cannot be obtained by the use of the closed nozzle having a constant orifice area without auxiliary means. Usually, a maximum rate of discharge is obtained at low throttle settings, and an increase in throttle setting maintains the same maximum rate with an increase in the injection period, the injection period varying directly with the fuel quantity discharged (fig. 9). A low initial rate of discharge could be obtained by causing a small orifice to open before the main discharge orifices, or a very small total-orifice area could be used; such means would require extremely high pressures to obtain the maximum rate of discharge. As the closed nozzle is little affected by a change in the initial rate of plunger displacement at port closure, an injection timing in which the beginning of injection was advanced with throttle setting would not change the initial rate of discharge.

The initial rate of discharge of an open nozzle depends upon the injection valve-opening pressure and the rate of plunger displacement at port closure. A low initial rate of injection cannot be obtained with a high valve-opening pressure even with a low initial rate of plunger displacement at port closure (fig. 11). Injection is delayed until the plunger compresses the fuel sufficiently to open the valve; consequently, injection occurs during a period of higher plunger velocity. An initial rate of discharge that does not vary with throttle setting can be obtained by using a varying cut-off so that injection always starts on the same portion of the cam outline (fig. 11).

The effects of the variables tested in this report on the shape of the rate-of-discharge curve of a unit injection system are presented in table II. Other notable variables, not presented, have a considerable influence on the rate of discharge. Among these variables are: volume of oil in the injection system; volume of oil between the valve seat and the orifice; effect of restriction at the valve-stem seat (spring scale); the ratio of valve-opening pressure to valve-closing pressure; and cam outline.

The cam outline can be partially corrected for the preceding variables. The total variation, however, may not be greater than 15 percent, and the lack of understanding of the actual injection characteristics desired for the varying engine cycles may make unfeasible an attempt to apply an adjustment or a correction that would apply only to a constant injection condition.

CONCLUSIONS

The test data and the analysis presented indicate that an assumed rate of discharge for the unit injection system tested may be modified by a number of variables, such as leakage, compressibility of the fuel, fuel viscosity change with pressure, restriction to flow through fuel passages, and minimum valve-opening pressure. When an open nozzle is used, the values of these quantities may be neglected as negligible, within limits, in designing a cam outline to give a rate of plunger displacement identical with the rate of discharge desired.

For short injection periods at high rates of discharge, rates of discharge comparable with the rate of plunger displacement can be obtained with the closed nozzle.

A serious cause of disagreement between the static rate of plunger displacement and the actual rate of discharge can be attributed to the compressibility of the fuel under conditions of high pressure.

Langley Memorial Aeronautical Laboratory,
National Advisory Committee for Aeronautics,
Langley Field, Va., May 22, 1940.

REFERENCES

1. Marsh, Edred T.: Discharge Characteristics of a Simulated Unit Injection System. NACA TN No. 676, 1938.

2. Anon.: The General Motors Diesel. The Auto. Eng., vol. XXIX, no. 382, March 1939, pp. 71-75.

3. Gelalles, A. G., and Marsh, E. T.: Rates of Fuel Discharge as Affected by the Design of Fuel-Injection Systems for Internal-Combustion Engines. NACA Rep. No. 433, 1932.

4. Lutz, O.: The Processes in Spring-Loaded Injection Valves of Solid Injection Oil Engines. NACA TM No. 758, 1934.

5. Rothrock, A. M.: Hydraulics of Fuel Injection Pumps for Compression-Ignition Engines. NACA Rep. No. 396, 1931.

6. Rothrock, A. M., and Marsh, E. T.: Distribution and Regularity of Injection from a Multicylinder Fuel-Injection Pump. NACA Rep. No. 533, 1935.

TABLE I

Operating Conditions for Rate-of-Discharge Tests

Injection valve	Orifice diameter (in.)	Plunger-and-sleeve design	Fuel quantity (lb/injection)	Valve-opening pressure (lb/sq in.)	Pump speed (rpm)	Figure
A_1	0.050	2	1.068×10^{-3}	3,400		4,7,10
B	.050	2	1.042	800		4,11
C	.050	2	1.028	3,200		4
D	.050	2	.981	2,400		4
A	.060	1	1.339	3,400		5
A	.040	1	1.369	3,400		5
A	.0315	1	1.301	3,400		5,7
D	.060	2	.982	2,400		5,7,10
D	.040	2	.949	2,400		5
D	.0315	2	.864	2,400		5
A	.050	2	1.059	3,400	500	7
D	.060	1	.918	2,400		7
A	.0315	1	.999	3,400		7
A	.0315	1	.757	3,400		7
A	.0315	1	.500	3,400		7
A	.0315	1	.324	3,400		7
D	.060	1	1.340	2,400		8
D	.060	1	.918	2,400		8
D	.060	1	.550	2,400		8
D	.060	1	.185	2,400		8
A	.050	1	1.435	3,400		9
A	.050	1	.799	3,400		9
A	.050	1	.451	3,400		9
A_1	.050	2	1.069	3,400	750	10
A_1	.050	2	1.055	3,400	1,000	10
D	.060	2	.961	2,400	750	10
D	.060	2	.893	2,400	1,000	10
B	.050	2	.710	800	500	11
B	.050	2	.491	800	500	11
B	.050	2	.241	800	500	11
D	.060	2	1.382	2,400	750	11
D	.060	2	.955	2,400	750	11
D	.060	2	.505	2,400	750	11
B	.050	2	.989	2,300	1,000	11
B	.050	2	.665	2,300	1,000	11
B	.050	2	.443	2,300	1,000	11
B	.050	2	.215	2,300	1,000	11

TABLE II

SUMMARY OF THE EFFECTS OF VARIABLES TESTED ON THE SHAPE OF THE RATE-OF-DISCHARGE CURVE VARIABLES

S	pump speed
R_v	resistance to flow, valve passage
R_p	resistance to flow, bypass port
O	orifice diameter
VOP	valve-opening pressure
Dp	rate of plunger displacement (or velocity) for the interval

Interval (fig. 12)		Closed nozzle	Open nozzle
Symbol	Action		
A	Injection lag	Increased with increased S and R_v and with decreased D_p; independent of O	Increased with increased VOP, O, and S and with decreased D_p
B	Slope of initial rate of discharge	Slight decrease with decreased D_p and R_v; independent of O and S	Increased with increased VOP, D_p, and O and with decreased S
C	Maximum value of initial rate of discharge	Increased with increased O and D_p and with decreased R_v and S	Increased with increased O and D_p and with decreased S and VOP
D	Time to reach C	Practically independent of variables tested	Increased with decreased VOP and D_p; practically independent of S and O
E	Pressure duration, constant throttle	Increased with increased S and with decreased O	Increased with increased S and with decreased O
F	Variation between D_p and rate of discharge	Oscillations reduced by increased D_p and S and by decreased O and R_v	Effect of compressibility of fuel increased by increased S and by decreased O
G	Time between static and dynamic opening of ports	Increased with decreased O and with increased S and duration of pressure cycle	Increased with decreased O and with increased S and duration of pressure cycle
H	Slope of cut-off	Increased with increased O and with decreased R_p, R_v, and S; practically independent of D_p	Increased with increased O and D_p and with decreased S and R_p
I	Trailing off	Increased with decreased O and with increased R_v and R_p	Increased with increased R_p, S, and volume of fuel between valve seat and orifice

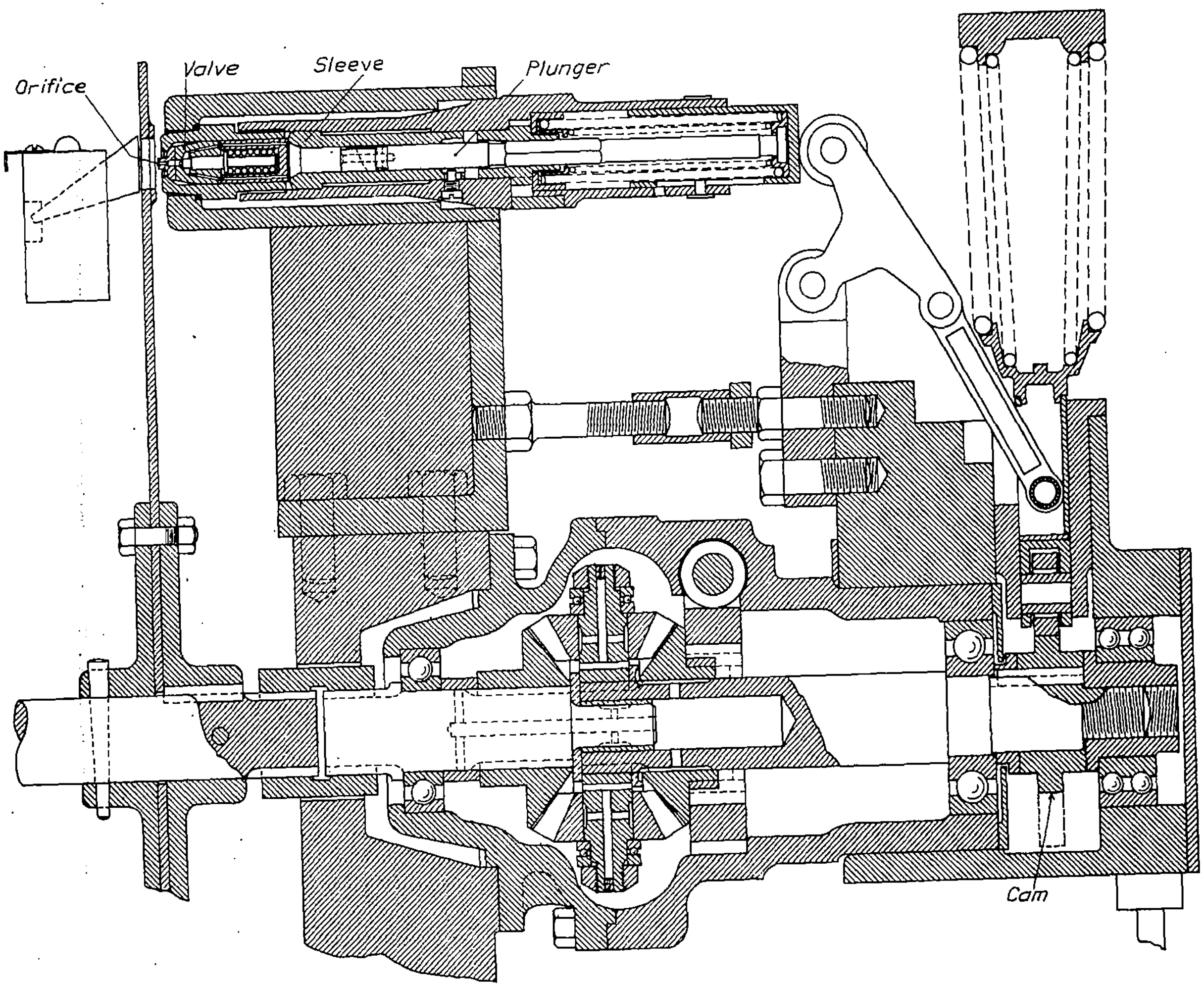


Figure 1.- Unit injector and actuating mechanism.

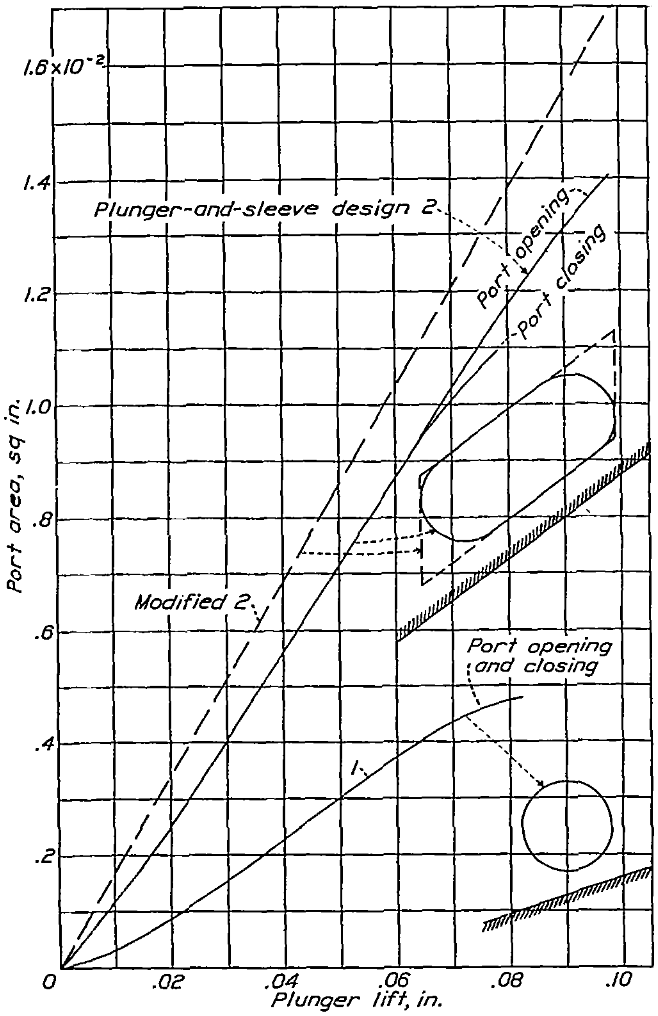


Figure 2.- Variation in port opening and closing area with plunger lift.

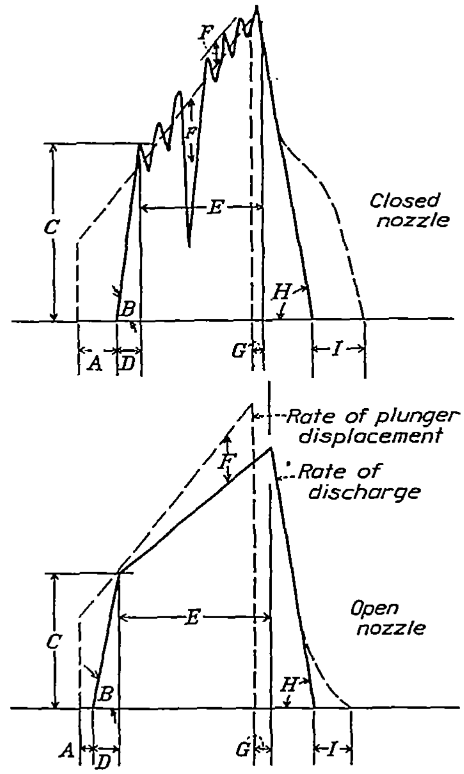
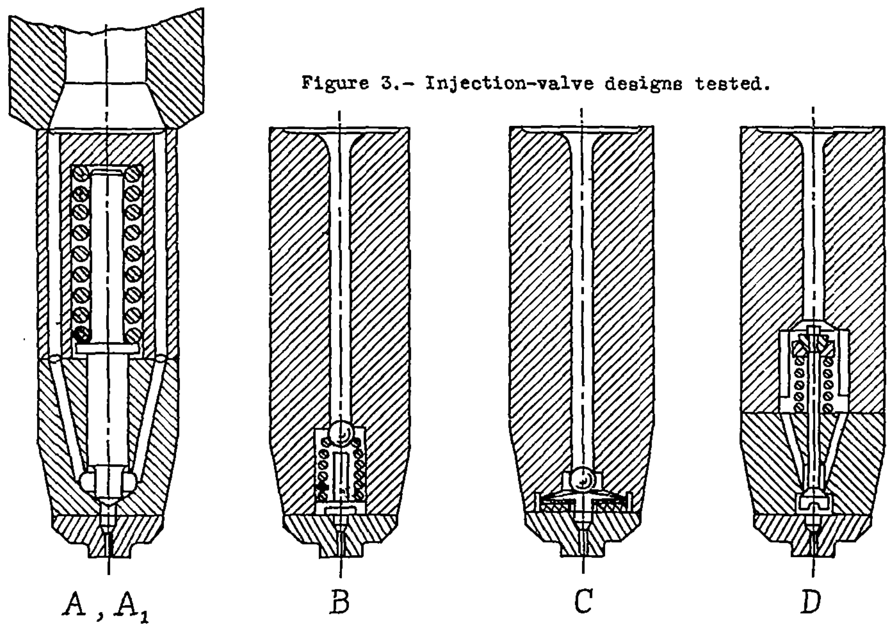


Figure 12.- Diagrammatic rate-of-discharge curves.

Figure 3.- Injection-valve designs tested.



	(a)	(b)	(c)	(d)
Injection-valve design	A_1	B	C	D
Valve-opening pressure, lb per sq in	3400	800	3200	2400

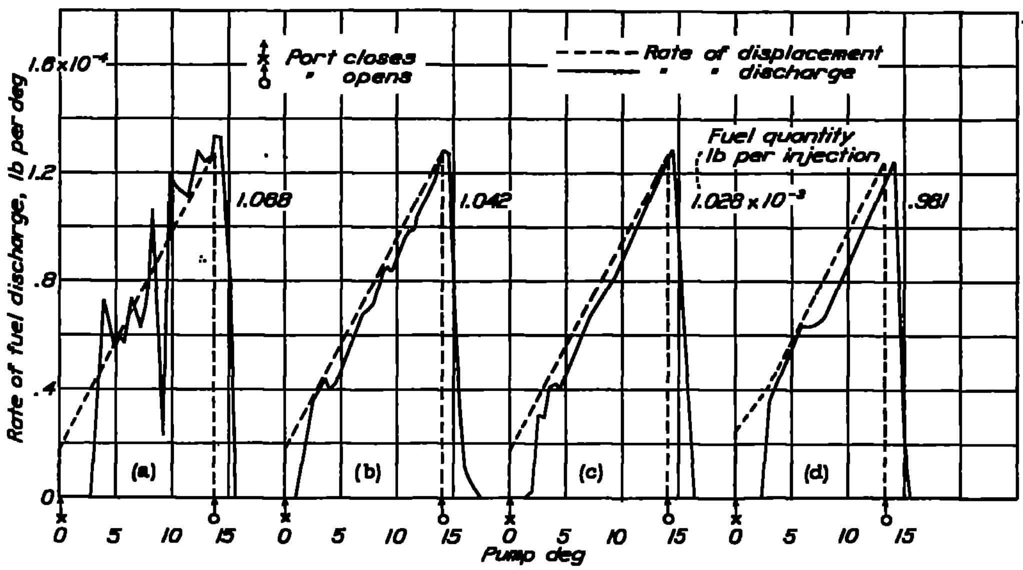


Figure 4.- Effect of injection-valve design on the rate of discharge. Orifice diameter, 0.060 in., pump speed, 500 rpm, Plunger-and-sleeve design 2.

	(a)	(b)
Injection-valve design	A	D
Valve-opening pressure, lb per sq in.	3400	2400
Plunger-and-sleeve design	1	2

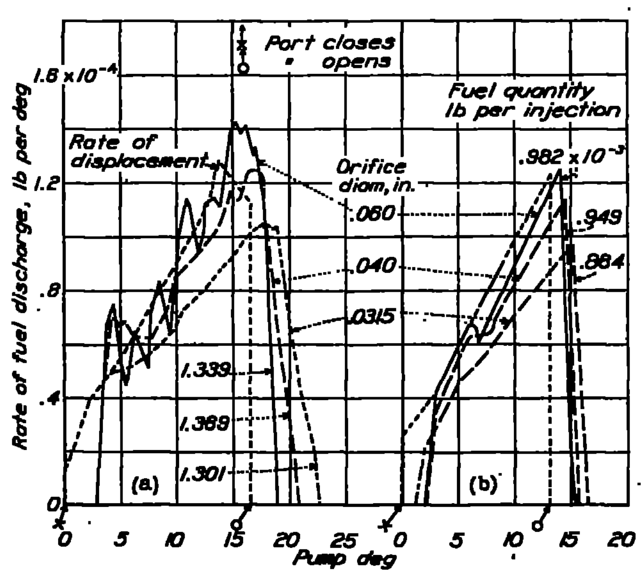


Figure 5.- Effect of orifice diameter on the rate of discharge. Pump speed, 500 rpm

	(a) A, A_1	(b) D	(c) A
Injection-valve design	A, A_1	D	A
Valve-opening pressure, lb per sq in.	3400	2400	3400
Orifice diameter, in.	.050	.060	.0315
Plunger-and-sleeve design	2	1,2	1

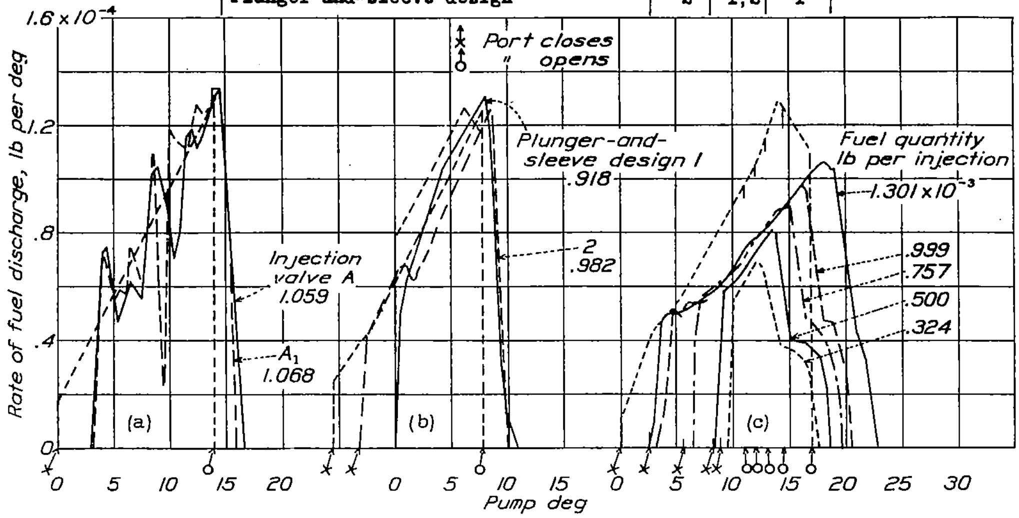


Figure 7.- Effect of fuel-passage area on the rate of discharge. Pump speed 500 rpm

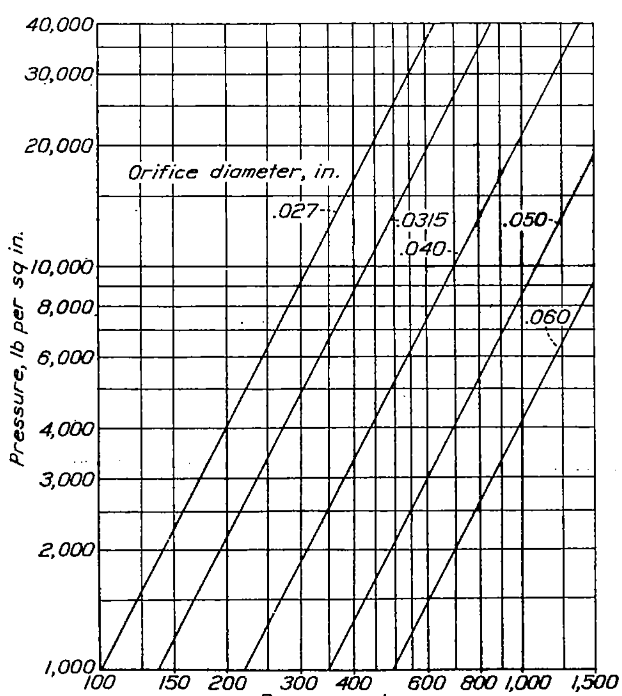


Figure 6.- Maximum injection pressures calculated from the theoretical maximum rate of discharge.