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NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

REPORT No. 363

PRESSURE FLUCTUATIONS IN A COMMON-RAIL FUEL INJECTION SYSTEM

By A. M. ROTHROCK

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AERONAUTICAL SYMBOLS

1. FUNDAMENTAL AND DERIVED UNITS

	Symbol	Metric		English	
		Unit	Symbol	Unit	Symbol
Length	l	meter	m	foot (or mile)	ft. (or mi.)
Time	t	second	sec	second (or hour)	sec. (or hr.)
Force	F	weight of one kilogram	kg	weight of one pound	lb.
Power	P	kg/m/sec		horsepower	HP.
Speed		km/hr		mi./hr	M. P. H.
		m/sec		ft./sec	f. p. s.

2. GENERAL SYMBOLS, ETC.

W, Weight, $=mg$

g, Standard acceleration of gravity $=9.80665$ m/sec.$^2=32.1740$ ft./sec.^{2}

m, Mass, $=\frac{W}{g}$

ρ, Density (mass per unit volume).

Standard density of dry air, 0.12497 (kg-m^{-4} sec.^{2}) at 15° C and 760 mm $=0.002378$ (lb.-ft.$^{-4}$ sec.^{2}).

Specific weight of "standard" air, 1.2255 kg/m$^3=0.07651$ lb./ft.^{3}

mk^2, Moment of inertia (indicate axis of the radius of gyration, k, by proper subscript).

S, Area.

S_w, Wing area, etc.

G, Gap.

b, Span.

c, Chord length.

b/c, Aspect ratio.

f, Distance from $c.\ g.$ to elevator hinge.

μ, Coefficient of viscosity.

3. AERODYNAMICAL SYMBOLS

V, True air speed.

q, Dynamic (or impact) pressure $=\frac{1}{2}\rho V^2$

L, Lift, absolute coefficient $C_L=\frac{L}{qS}$

D, Drag, absolute coefficient $C_D=\frac{D}{qS}$

C, Cross-wind force, absolute coefficient $C_C=\frac{C}{qS}$

R, Resultant force. (Note that these coefficients are twice as large as the old coefficients L_C, D_C.)

i_w Angle of setting of wings (relative to thrust line).

i_t, Angle of stabilizer setting with reference to thrust line.

γ, Dihedral angle.

$\rho\frac{Vl}{\mu}$, Reynolds Number, where l is a linear dimension.

e. g., for a model airfoil 3 in. chord, 100 mi./hr. normal pressure, 0° C: 255,000 and at 15° C., 230,000;

or for a model of 10 cm chord 40 m/sec, corresponding numbers are 299,000 and 270,000.

C_p, Center of pressure coefficient (ratio of distance of $C.\ P.$ from leading edge to chord length).

β, Angle of stabilizer setting with reference to lower wing, $=(i_t-i_w)$.

α, Angle of attack.

ϵ, Angle of downwash.

REPORT No. 363

PRESSURE FLUCTUATIONS IN A COMMON-RAIL FUEL INJECTION SYSTEM

By A. M. ROTHROCK
Langley Memorial Aeronautical Laboratory

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

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REPORT No. 363

PRESSURE FLUCTUATIONS IN A COMMON-RAIL FUEL INJECTION SYSTEM

By A. M. Rothrock

SUMMARY

The tests reported herein were conducted at the Langley Memorial Aeronautical Laboratory, Langley Field, Va., to determine experimentally the instantaneous pressures at the discharge orifice of a common-rail fuel injection system in which the timing valve and cut-off valve were at some distance from the automatic fuel injection valve, and also to determine the methods by which the pressure fluctuations could be controlled.

The instantaneous pressures at the discharge orifice of a common-rail fuel injection system were determined by analyzing the stem-lift records of an automatic injection valve. The fuel injection was obtained by releasing fuel from a reservoir under high pressure by means of a cam-operated timing valve. The period of injection was controlled by the opening of a cam-operated by-pass valve which reduced the fuel pressure between the timing valve and the injection valve. An injection system of this type assures the same rate of fuel discharge regardless of engine speed. The results show that pressure wave phenomena occur between the high-pressure reservoir and the discharge orifice, but that these pressure waves can be controlled so as to be advantageous to the injection of the fuel. The results also give data applicable to the design of such an injection system for a high-speed compression-ignition engine.

INTRODUCTION

In order to control the injection of fuel into the combustion chamber of a compression-ignition engine, it is necessary to know the pressure variations at the discharge orifice as well as the time interval of injection. These pressure variations are controlled by the design of the fuel injection system and the physical properties of the fuel. The system should be designed to meet the requirements imposed by these physical properties in such a manner that the instantaneous pressures at the discharge orifice cause injection of the fuel according to the requirements of efficient combustion. Since fuel oils are compressible, they are subject to pressure wave phenomena. Because of the fluctuations of these pressures in the injection system, the instantaneous pressures can not be recorded with a static gauge. In fact, the pressure fluctuations may be so rapid and so violent as to make it impossible to predict the rates of discharge from the orifice by means of such pressure gauge readings.

The quantity of fuel delivered by a common-rail fuel injection system is controlled either by the lift of a mechanically operated fuel injection valve or by the time interval during which the oil in the high-pressure reservoir is released to the injection valve. The first type has not been used extensively with high-speed engines, because of the difficulty of controlling the rate of fuel flow by controlling the lift of a mechanically operated valve. With the second type it is necessary to have one or two mechanically operated valves in conjunction with either an automatic injection valve or an open nozzle. When one mechanically operated valve is employed, the fuel under pressure is released to the discharge orifice for the time interval during which the mechanically operated valve remains opened. When this valve closes the fuel continues to discharge until the oil pressure between the mechanically operated valve and the injection valve drops to a value less than the injection valve opening pressure, or to the pressure in the combustion chamber if an open nozzle is used. This results in a comparatively slow cut-off of the fuel spray. If, however, an automatic injection valve is used and a second mechanically operated valve is employed, which causes the stop of fuel discharge by releasing the pressure between the high-pressure reservoir and the injection valve to some value less than the valve closing pressure, a sharp cut-off of the fuel spray is assured. Such a fuel injection system should have the following characteristics:

1. Constant time for a given fuel quantity to be discharged regardless of engine speed.
2. Sharp start of fuel spray.
3. Sharp cut-off of fuel spray.
4. Constant fuel dispersion and penetration regardless of engine speed, except for the effect of air flow.

To design a fuel injection system of this type with a definite rate of fuel discharge, the effect of the different parts of the injection system on the instantaneous pressures must first be determined. If these are known, the pressure fluctuations throughout the injection system and the rate at which the fuel is discharged can be varied almost at will.

The instantaneous injection pressures in an injection system have been measured either directly or indirectly by several investigators. Hesselman (Reference 1), using a combination common-rail and pump injection system, recorded the movement of the automatic injection valve stem, and from the area exposed for discharge into the combustion chamber computed the rate of fuel discharge. Ricardo (Reference 2) measured the instantaneous pressures from a fuel pump by means of the R. A. E. indicator. A third investigator (Reference 3) recorded the stem movement of an automatic injection valve and from it attempted to predict the pressure variations at the discharge orifice. Unfortunately his analysis is incomplete and consequently the pressure curves presented are not justified.

Gelalles and the author (Reference 4) have recorded the movement of the timing valve stem on the injection system of the N. A. C. A. Spray Photography Equipment when the stem was operated hydraulically and, in addition, have computed the stem movement from an analysis of the flow of oil through the valve. The computed movement of the stem was in close agreement with the actual movement. From these results the movement of the stem of an automatic injection valve was computed, but due to limitations of the apparatus at the time, the actual movement was not determined.

The author (Reference 5) has determined experimentally the time interval between the opening of the timing valve and the start of the fuel spray from an injection valve for the injection system of the N. A. C. A. Spray Photography Equipment. The results of these tests showed that the automatic injection valve was opened by a pressure wave.

METHODS AND APPARATUS

The injection system of the N. A. C. A. Fuel Spray Photography Equipment (Reference 6) was used for this investigation. It is illustrated diagrammatically in Figure 1. The timing valve cam was operated by a clutch which, when engaged, caused the cam to make one revolution. As the timing-valve needle was lifted from the seat the oil under pressure in the high-pressure reservoir was released through the injection valve tube to the injection valve. The oil pressure acting on the annular area of the valve stem forced the stem from the seat and the oil was sprayed into the chamber. The by-pass or spray cut-off valve then opened, the hydraulic pressure in the injection valve tube dropped due to the rapid flow of the oil through the by-pass valve, and the injection valve spring returned the stem to its seat, cutting off the fuel spray. The time interval between the start of opening of the timing valve and by-pass valve could be varied as shown. The by-pass valve was so adjusted that it opened before the timing valve started to close.

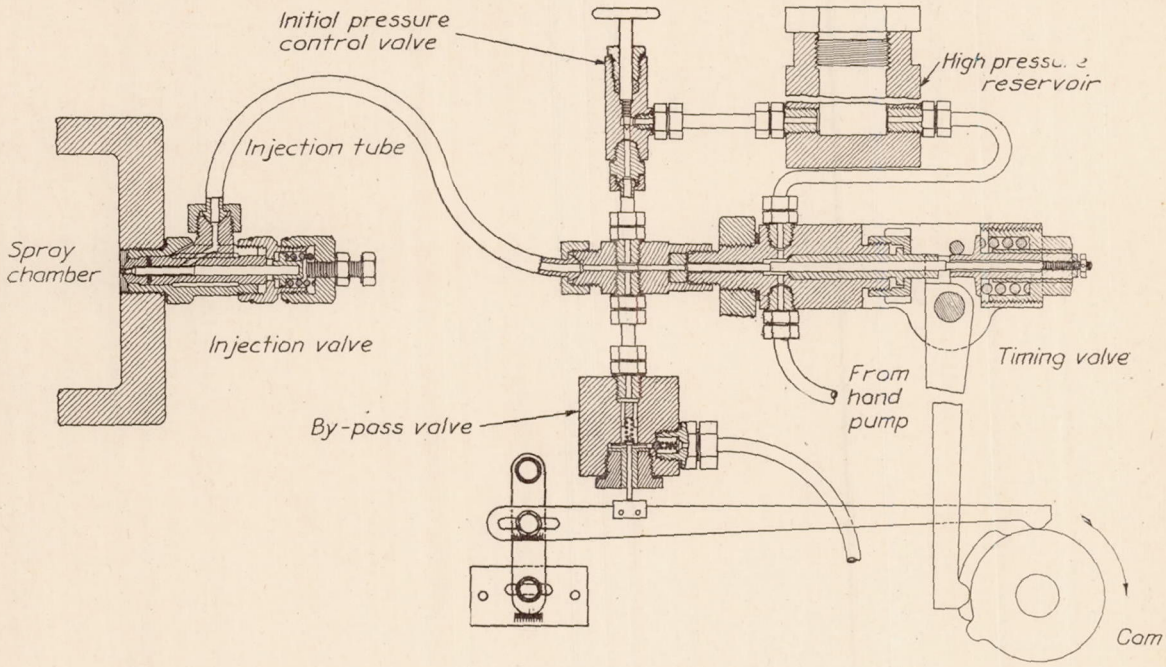


FIGURE 1.—Fuel spray injection system

The initial pressure in the injection valve tube was adjusted to any desired value by means of the initial pressure control valve. The hydraulic pressures were obtained by a hand pump. The static pressures were indicated by a Bourdon spring gauge mounted in the line between the hand pump and the high-pressure reservoir. An initial pressure of 300 pounds per square inch and a camshaft speed of 1,140 r. p. m. were used in all the tests except where otherwise

stated. Particular care was taken to remove all air from the injection system before each test was made. The fuel used was a high grade Diesel fuel oil with a specific gravity of 0.86 at 80° F., and an absolute viscosity of 0.048 poises at 80° F., and atmospheric pressure.

The injection valve shown in Figure 1 was altered as shown in Figure 2, so that the stem movement could be recorded. The timing valve was also altered in the same manner. The movement of each valve stem was recorded by directing a beam of light from a point source onto a mirror operated by the valve stem and focusing the reflected beam onto a film mounted on a revolving drum. The two film drums were mounted on the same shaft, and driven by a synchronous motor at a peripheral speed of 1,038 inches per second. Electromagnetic shutters operated by the camshaft were placed in front of the film drums so that the beam of light fell on them for not more than three revolutions. This was done so that the line of zero lift would not appear as a heavy band. During the preliminary part of the investigation the injection valve was mounted in the spray chamber as shown in Figure 1, and records were taken simultaneously of the timing valve stem movement, the injection valve stem movement, and the development of the fuel spray. In order to synchronize the three records, small spark gaps were placed in front of the two stem movement films. These were connected in series with the main spark gap for taking the high-speed motion pictures of the fuel spray. Hence, for each spray photograph there appeared a short line on each of the two films. Figure 3 shows records of the movements of the valve stems and of the development of the fuel spray taken in this manner. For this particular test the stem stop of the injection valve was set to limit the maximum lift to 0.021 inch.

An examination of the figure shows that both the start of the spray and the start of the injection valve stem movement occurred between the seventh and eighth photographs, and that spray cut-off came between the fifteenth and sixteenth photographs. The stem did not stay against the stop because of the pressure fluctuations. The three records taken simultaneously showed that the start and stop of the spray followed the start and stop of the injection valve stem movement within a few hundred thousandths of a second, that is, within the accuracy of the experimental data. The effect of the several variables on the time interval between the opening of the timing valve and the start of the fuel spray had been determined previously (Reference 5). Therefore, spray and timing valve records were not taken for the majority of the tests, and the injection valve was mounted in theholder shown in Figure 2 so that calibration records could be taken after each stem lift diagram.

In order that the stem movement could be used as the instantaneous pressure indicator, all of the injection-valve records for the pressure analysis were taken with the stem stop backed off, so that the stem did not strike it. A typical record is shown in Figure 4. It

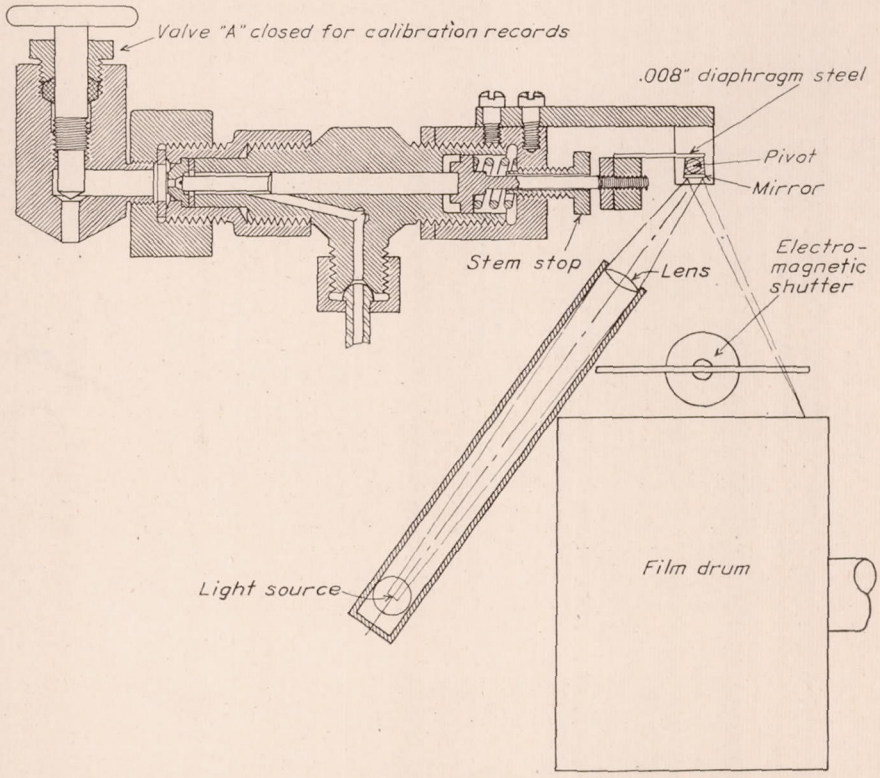
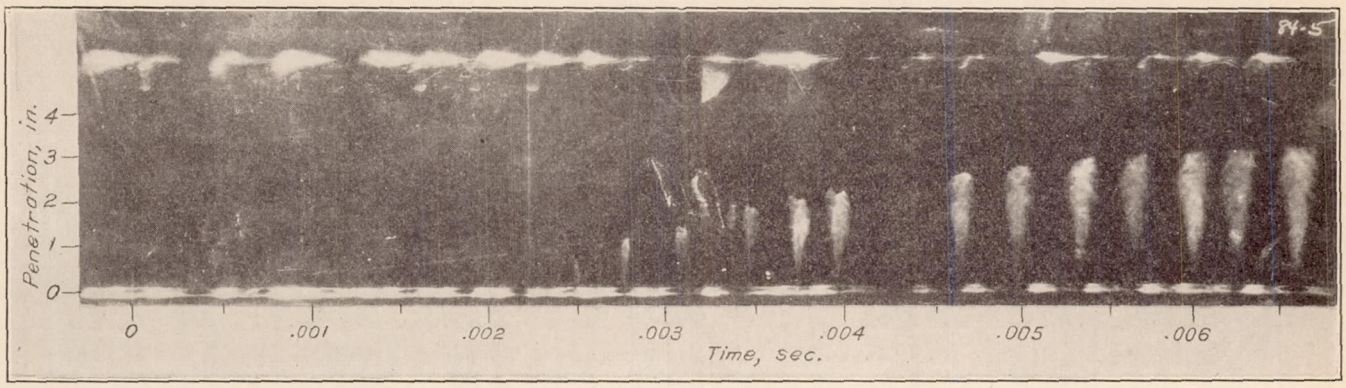


FIGURE 2.—Automatic injection valve and apparatus for recording valve stem movement

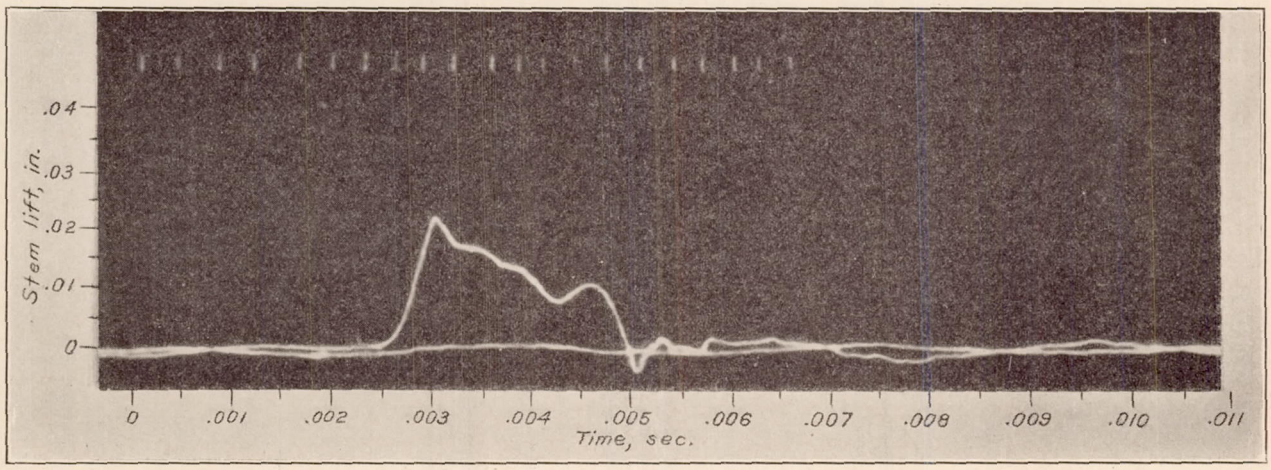
is seen that the stem came in contact with the seat after the first oscillation, but did not touch it again until spray cut-off, at which time it struck the seat and made a series of short bounces. Such bounces are the cause of the discharges which Beardsley defines as secondary discharges and discusses in his report on the reproducibility of spray data. (Reference 7.) They occur when the pressure in the injection valve tube is great enough during the bouncing of the stem to cause discharge. They are not due to a pressure wave oscillating between the by-pass valve and the injection valve, since their frequency is too high for such an oscillation.

The calibration of the spring in the injection valve was obtained by closing the valve A shown in Figure 2. The stem recording mechanism was removed, and a dial indicator graduated in thousandths of an inch

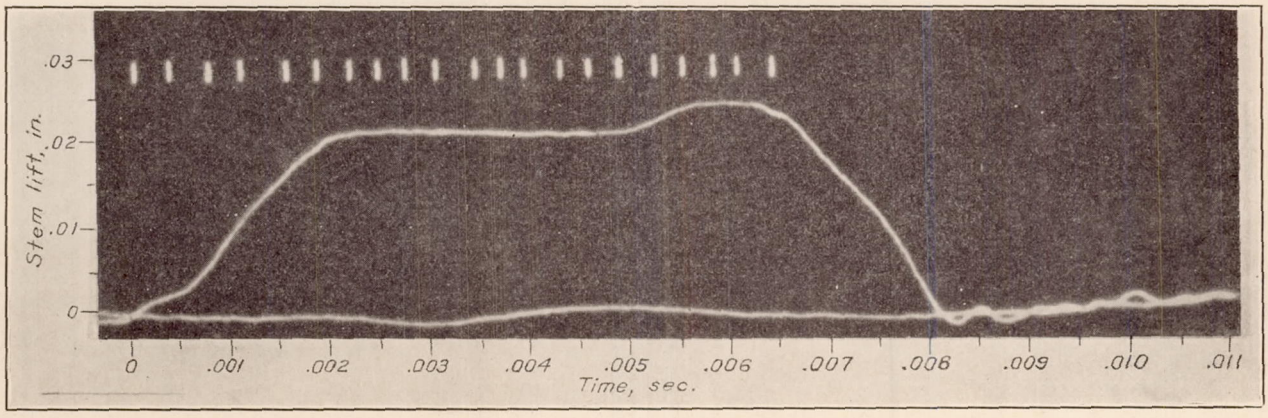
was placed against the spring follower of the injection valve. The hydraulic pressure was built up in stages of 500 pounds per square inch and the lift readings recorded. From these data the spring scale of 3,600 pounds per square inch was computed. The calibration of the lift as recorded on the film was obtained by replacing the lift recording mechanism and building up the pressures in stages of 500 pounds per square inch. At each pressure increment the point source light was flashed on for an instant, so that a record was obtained on the revolving film with straight lines representing the lift for each increment. This record was converted into actual lift of the valve stem in inches from the spring calibration. It was necessary to take film calibration records after each lift record, for, in changing the conditions between successive tests, the angle of the point source light was changed and any difference in this angle changed the recorded lift scale. The calibration record for Figure 4 is shown in Figure 5. The corresponding pressures for



Fuel spray photographs



Record of injection valve stem motion with stem stop set at 0.021 inch lift



Record of timing valve stem motion

FIGURE 3.—Fuel spray development, injection valve stem motion, and timing valve stem motion recorded simultaneously

each line have been marked on the film. In this particular case the valve opening pressure was 2,500 pounds per square inch; consequently, there was no lift recorded until this pressure was reached.

In order to obtain the frequency of the diaphragm steel link, the diaphragm was deflected slightly and then released, and at the same time a photographic record was taken of the vibration of the reflected light beam. The record, Figure 6, shows that the frequency of vibration was 5,000 per second. In the stem records the effects of these vibrations are noticeable to an appreciable degree only when the stem strikes the seat during the bouncing after cut-off, Figure 4.

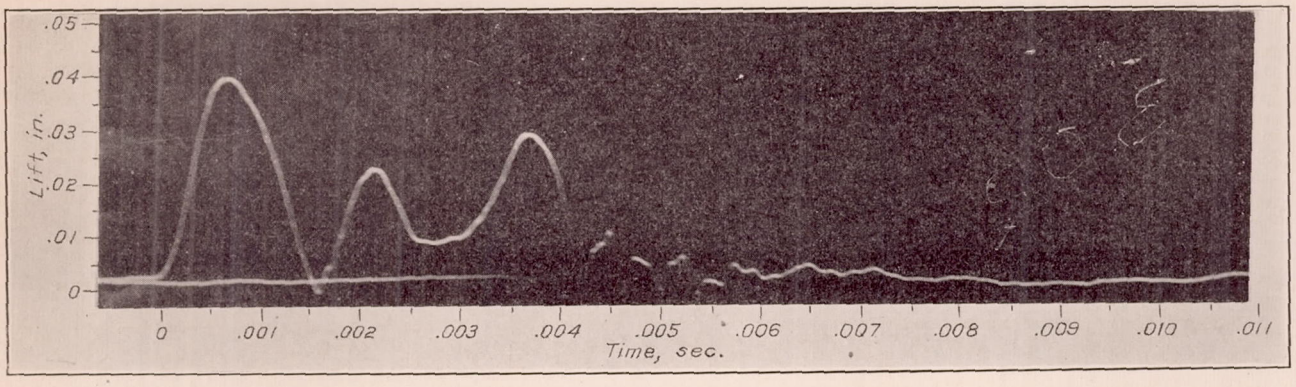


FIGURE 4.—Record of injection valve stem motion with no stop

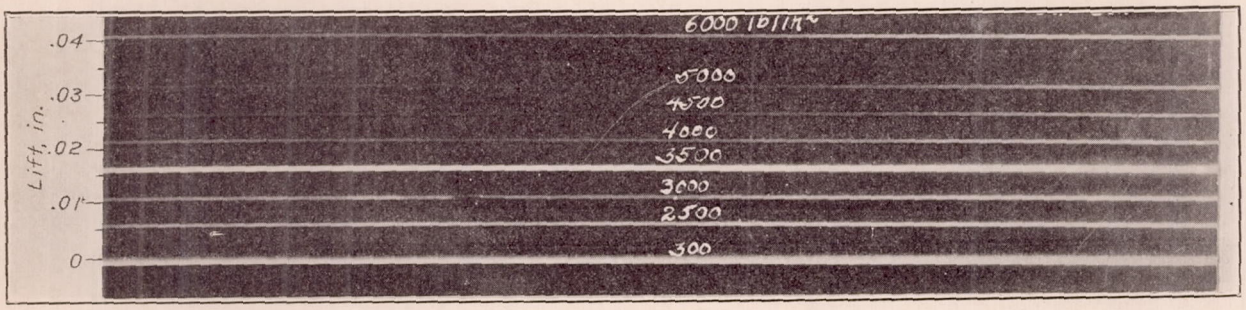


FIGURE 5.—Pressure calibration record

DERIVATION OF THE INSTANTANEOUS PRESSURES AT THE DISCHARGE ORIFICE FROM THE STEM LIFT RECORDS

At every instant the hydraulic force on the valve stem was opposed by the resisting force of the valve spring plus the product of the mass of the moving parts and their acceleration at that instant plus or minus the friction of the stem in its guide. With the injection valve used the lapped stem slid freely in the valve body when the valve was assembled, and consequently the friction force was neglected. The force equilibrium equation is therefore:

$$f = \lambda s + ma \qquad (1)$$

in which f is the hydraulic force at any time t, λ the spring scale, s the compression of the valve spring at the time t, m the mass of the moving parts, and a the acceleration of the stem at the time t. The spring scale has already been given. The mass of the moving parts was taken as the mass of the valve stem, plus the mass of the spring follower and its attachments plus half the mass of the valve spring and the mirror and its support. For the valve tested the total moving mass was 0.069 pound. The spring compression at any instant was the sum of the initial compression of the spring, obtained from the calibration record, plus the stem lift at the instant under consideration, obtained from the stem lift record. The acceleration at any instant was obtained by first drawing the time velocity curve from the tangents to the time lift curve taken at 0.0001-second intervals and then drawing the time acceleration curve from the tangents of the velocity curve taken at the same intervals. The mean pressure on the stem was the total force divided by the area of the stem −0.0398 square inch.

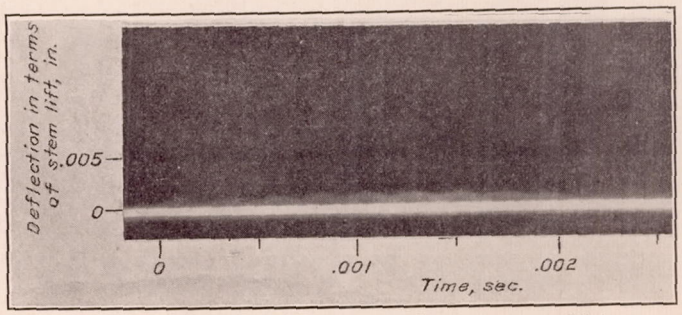


FIGURE 6.—Record of vibrations caused by diaphragm steel link

Figure 7 shows the velocity, acceleration, and pressure curves, together with the lift curve for the experimental record shown in Figure 4. The bouncing of the stem after cut-off is omitted since it was not the purpose of these tests to investigate secondary discharges. The time scale in Figure 7 was enlarged over that in Figure 4 so that the tangents for the velocity curve were for angles less than 45 degrees, since the rate of change of the tangent is less for the smaller angles. The curves in Figure 7 are characteristic of the curves obtained for the stem movement in the majority of the tests. There is first an oscillation of high amplitude, then one of small amplitude, followed by one of a slightly greater amplitude. These oscillations indicate that the stem movement was controlled by two harmonics imposed upon each other. One was the fundamental harmonic of the valve spring. Its period was $2\pi\sqrt{\frac{m}{\lambda}}$ which for the spring used was 0.0014 second, approximately the period of the first oscillation, 0.0018 second. The other was the fundamental harmonic of the oil column between the discharge orifice and the high-pressure reservoir. Its period was $2\pi l\sqrt{\frac{\rho}{2Eg}}$ in which l is the length of the oil column between the discharge orifice and the high-pressure reservoir, 26.7 inches for the 13-inch tube, ρ the density of the Diesel oil, 53 pounds per cubic foot, E the modulus of elasticity of the oil, 284,000 pounds per square inch (Reference 8), and g the gravitational constant. Substituting these values in the expression gives a value of 0.0020 second. The actual period as obtained from the figure is 0.0026 second. The difference between the actual period and the theoretical period can be accounted for in part by the fact that the value of E has not been accurately determined, but chosen from the values of other oils of similar properties. The curves show in nearly all cases a sharp drop in pressure about 0.0004 second after the start of injection caused by the start of the discharge from the orifice. There is a sharp increase in pressure due to the restriction to oil flow whenever the stem reaches or nearly reaches the seat during its oscillations.

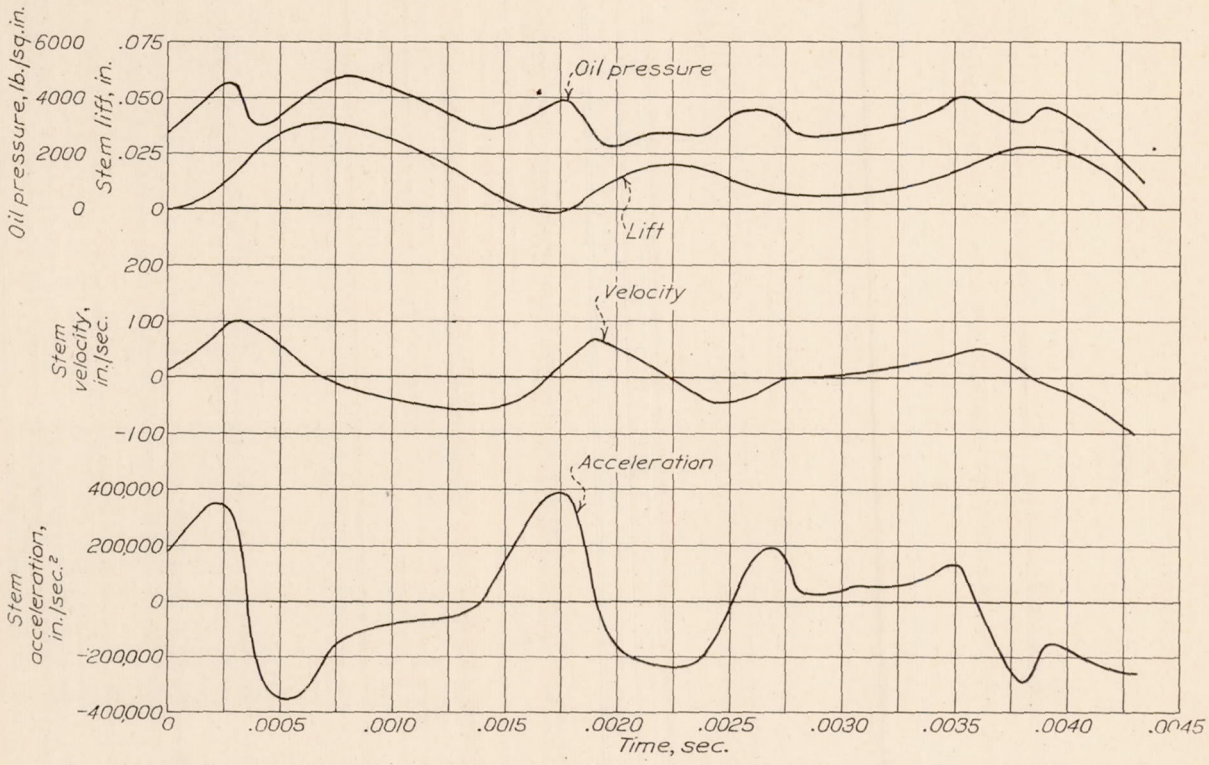


FIGURE 7.—Stem lift record shown in Figure 4 plotted together with stem velocity and acceleration curves and pressure at discharge orifice

In order to determine the accuracy of the tangent method for obtaining the accelerations and pressures on the injection valve stem, an equation was derived by means of Fourier's series for the record in which the fluctuations were most violent. In the computation it was assumed that the lift curve was symmetrical to the time axis and that the stem movement during the complete injection period represented a half period of a harmonic curve. The equation was determined for the first seventeen odd harmonics. The computed points checked the experimental points within 0.0002 inch. The second derivative of the equation was determined and transformed into terms of acceleration in inches per second per second. These values of acceleration were then used in deriving the instantaneous pressures from Equation (1). The results are shown in Figure 8. The end points are not given, since in assuming a curve symmetrical to the time axis the

computed curve must necessarily deviate from the actual curve for 0 and 180 degrees because of the difference in slope at these two points. The computed points shown, however, fall sufficiently close to the curves obtained by the tangent method to indicate the accuracy with which the tangents were drawn.

The flow of oil through the discharge orifice was computed from the pressure curves and the conventional flow formula, $Q=avt$. The total mass M discharged in any time t is

$$M=\rho Q$$
$$=\rho avt$$
$$=\pi\frac{d^2}{4}Ct\sqrt{2P\rho g} \qquad (2)$$

in which P is the effective pressure, d the diameter of the orifice, and C the coefficient of discharge. The effective pressure was taken as the square of the mean of the square roots of the instantaneous pressures. It was obtained by plotting the square roots of the instantaneous pressures, integrating the curve with a planimeter, dividing the integral by the total time of injection, and squaring this mean of the square roots to obtain the effective pressure of injection. Coefficient of discharge tests made on the 0.008 and 0.020 inch orifice with the apparatus employed by Joachim (Reference 9) showed that the coefficients of discharge of these orifices over the range of pressures investigated did not vary materially from 0.94.

The actual discharges were obtained by screwing a small container onto the injection valve and weighing 10 injections. In each weighing the nozzle and end of the valve were cleaned with benzol before attaching the container, and the nozzle and end of the valve were cleaned with a piece of cotton when the container was removed. The weight of the oil collected on the cotton was added to that collected in the container. A certain amount of oil vapor escaped while the container was being removed from the valve and the cap was being screwed on, and it was impossible to wipe all the oil off the nozzle. The exact magnitude of these two losses is difficult to estimate. In every case two or three sets of 10 discharges were weighed and the mean taken. In no case did the individual weights vary more than 3 per cent from the mean.

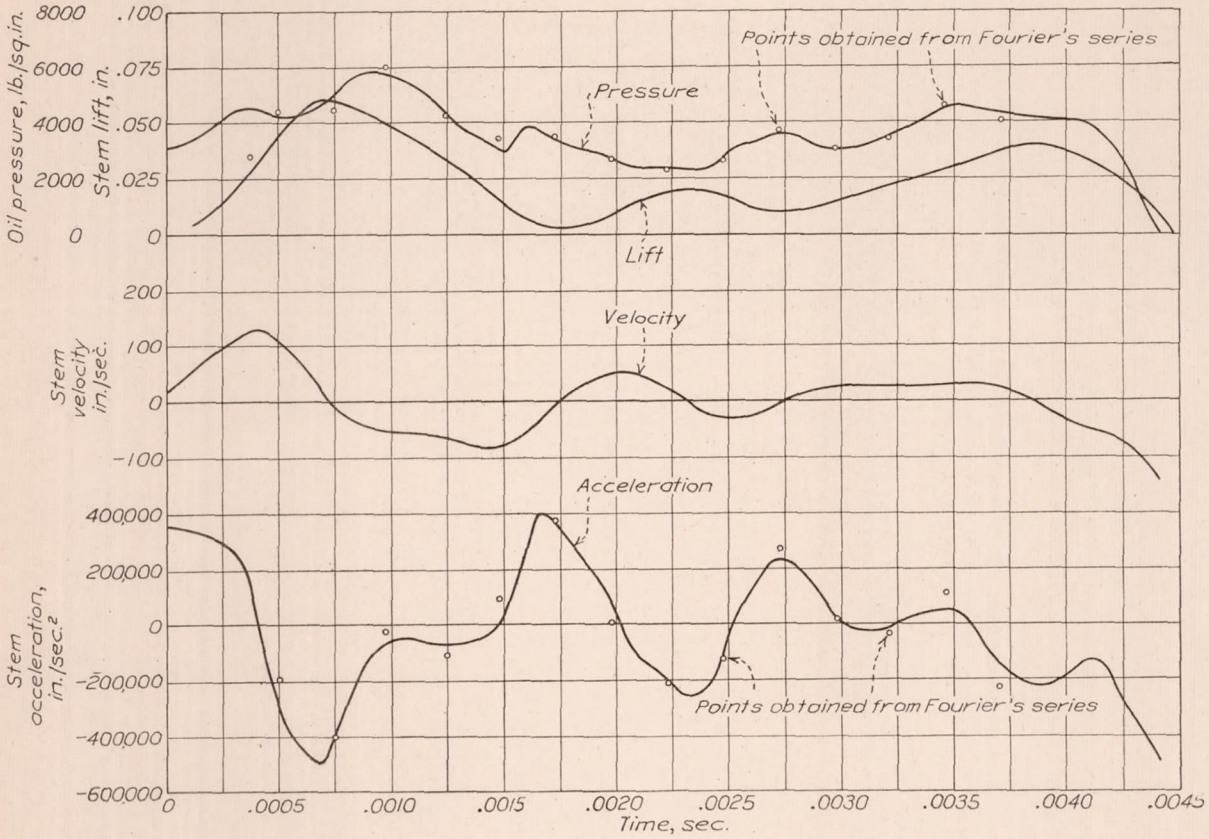


FIGURE 8.—Velocity, acceleration, and pressure curves obtained by tangent method and values of acceleration and pressure obtained from Fourier's series

TEST RESULTS

Effect of the Ratio of Discharge Orifice Area to Area of Smallest Restriction between the High-pressure Reservoir and Discharge Orifice.—The tube connecting the high-pressure reservoir with the timing valve had an internal diameter of 0.063 inch. The diameter of the passage connecting the timing valve to the injection

valve tube was 0.094 inch. The injection valve tube was made of seamless steel tubing with an outside diameter of ¼ inch and an inside diameter of 0.125 inch. The oil passage in the injection valve between the injection valve tube connection and the passage around the valve stem had a diameter of 0.094 inch.

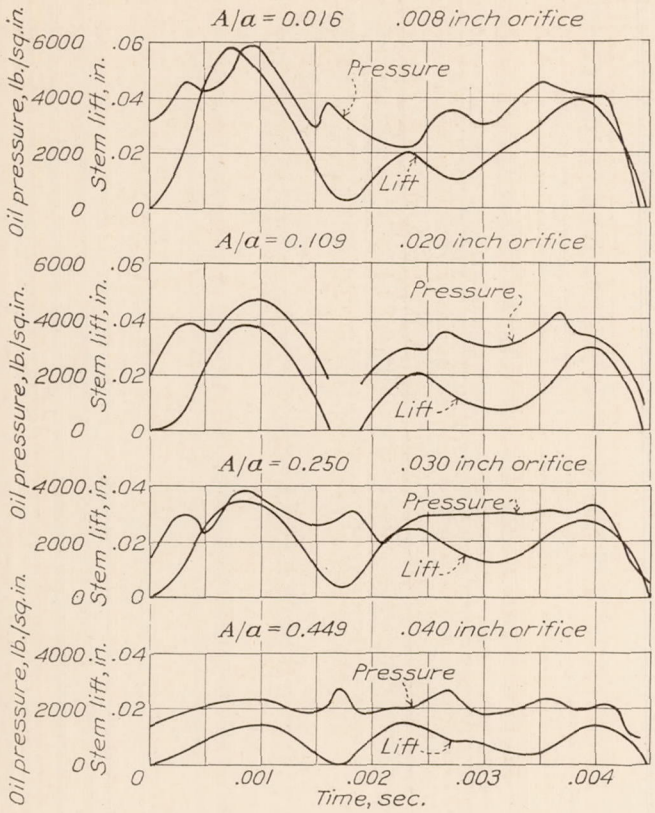


FIGURE 9.—Effect of orifice diameter on stem lift and oil pressure

Injection pressure	3,500 lb./sq. in.
Valve opening pressure	2,500 lb./sq. in.
Initial pressure	300 lb./sq. in.
Tube length	13 in.

Consequently, the smallest restriction was the 0.063-inch tube between the high-pressure reservoir and the timing valve. Two series of tests were conducted to determine the effect of the ratio of the discharge orifice area to the area of the smallest restriction in the line. In the first series the stem lift was determined for different diameter discharge orifices. In the second series the stem lift was determined with a constant discharge orifice diameter, but with various restriction orifices inserted in the injection line at the entrance to the automatic injection valve.

The effect of varying the discharge orifice diameter is shown in Figure 9. The ratio of the dicharge orifice area to the restriction area is symbolized by A/a. As the ratio was increased, more of the energy of the initial pressure wave was expended in discharging the fuel oil, and less of it was utilized in a wave reflected back to the high-pressure reservoir. Consequently, the intensity of the pressure waves and the stem lift decreased as the orifice diameter increased. Between the ratio of 0.25 and 0.45 there is a ratio above which the wave energy was apparently entirely transformed into the kinetic energy of the discharging oil and the pressures at the discharge orifice remained virtually constant, although the stem oscillated.

Figure 10 shows the effect of the ratio A/a when the discharge orifice remained constant and the diameter of the restriction orifice was varied. The same phenomenon occurred as is shown in Figure 9. The important fact to notice in these two figures is the similarity of the curves for the same ratio. For the 0.020-inch restriction orifice with a ratio of 1.0 the valve stem oscillated to a greater extent than with the ratio of 0.45. Although the values given for the instantaneous pressures in this particular case are open to question because of the difficulty of analyzing the curve at the line of zero lift, the curve does show that there were violent fluctuations of pressure, because of the rapid opening and closing of the valve.

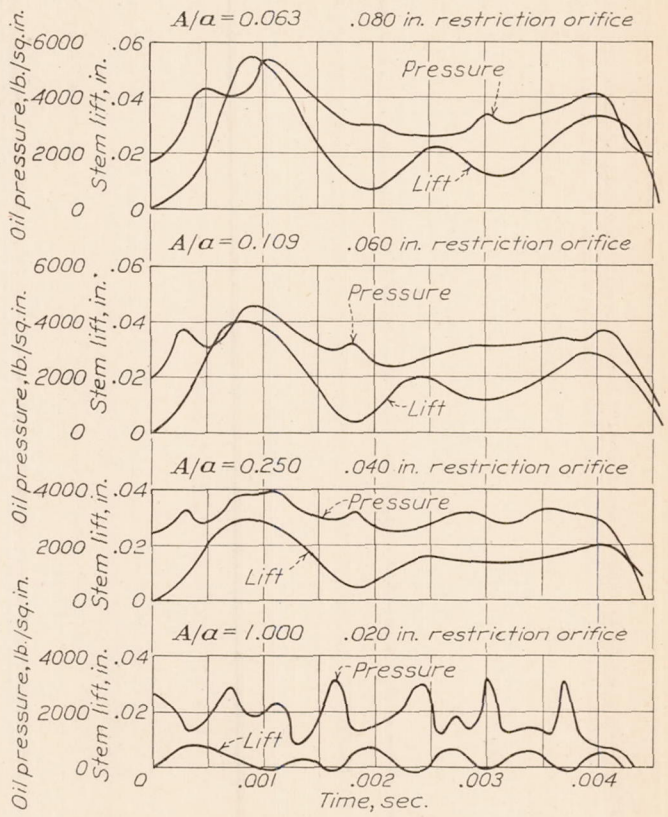


FIGURE 10.—Effect of restriction orifices on stem lift and oil pressures

Injection pressure	3,500 lb./sq. in.
Valve opening pressure	2,500 lb./sq. in.
Initial pressure	300 lb./sq. in.
Tube length	13 in.
Orifice diameter	0.020 in.

The effect of the ratio of discharge orifice area to the restriction orifice area on the effective pressures is shown in Figure 11. As the ratio decreased, the effective pressure approached the static pressure in the high-pressure reservoir. The difference between the static pressure and the effective pressure was directly

proportional to the ratio. Because the effective pressure decreased as the orifice area was increased, the quantity of fuel discharged, Figure 12, did not vary

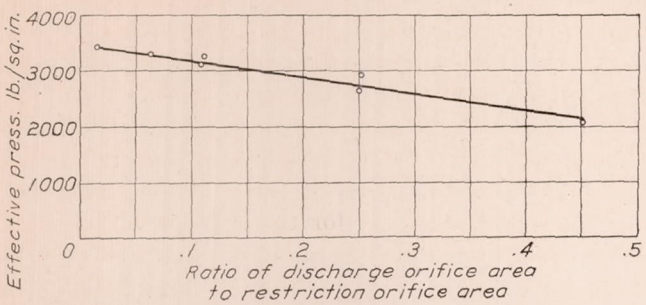


FIGURE 11.—Effect of ratio of discharge orifice area to restriction orifice area on effective pressure

directly with the orifice area. Both the computed and actual discharge curves follow the same general trend. The discrepancy between the two is due to the losses in weighing the fuel, to the restriction to flow when the valve stem approached the seat, and also because the mean pressure across the stem was

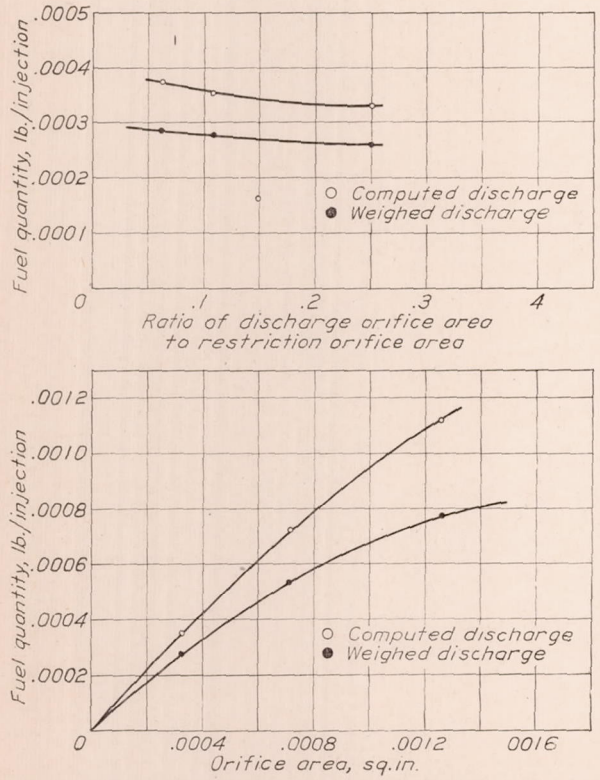


FIGURE 12.—Effect of restriction orifice area and discharge orifice area on fuel quantity discharged

probably greater than the mean pressure across the discharge orifice. In Figure 12 it is also seen that as the ratio A/a decreased, the fuel quantity discharged increased.

It can be concluded from the figures that for an injection system of this type the area of the smallest restriction in the fuel line should not be less than four times the area of the discharge orifice.

Effect of Injection Valve Tube Length.—The effects of the injection valve tube length on the stem lift and pressure variations are shown in Figure 13. As the length of the injection valve tube was increased the period of vibration of the oil column was increased. The computed periods for the tube lengths are 0.0020 second for the 13-inch tube, 0.0028 second for the 24-inch tube, 0.0037 second for the 37-inch tube, and 0.0063 second for the 70-inch tube. With the 13-inch tube the pressure reached a second maximum and had started to decrease when cut-off occurred. With the

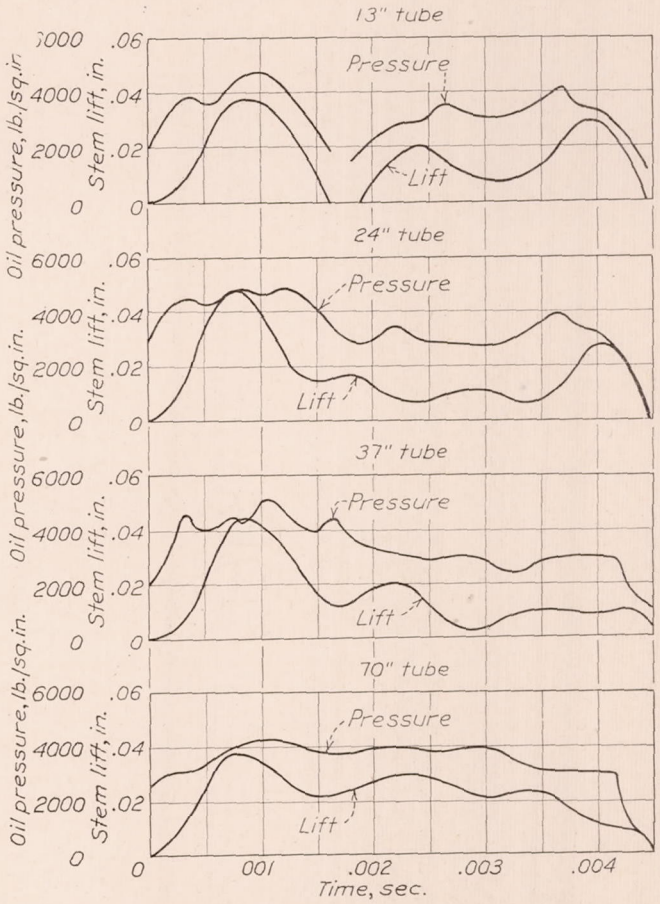


FIGURE 13.—Effect of tube length on stem lift and oil pressure

Orifice diameter	0.020 in.
Injection pressure	3,500 lb./sq. in.
Initial pressure	300 lb./sq. in.
Valve opening pressure	2,500 lb./sq. in.

24-inch tube the pressure reached a minimum and had started to increase when cut-off occurred. With the 37-inch tube cut-off occurred when the pressure had reached a minimum. The 70-inch tube gave virtually a constant pressure during the whole injection period, as the initial wave had just started to decrease at cut-off. It is interesting to note that there was a slight increase in the injection period for the longer tubes, although the setting of the cut-off valve was the same for all the tubes. Consequently, the time lag between the opening of the timing valve and the opening of the injection valve, instead of increasing

directly with the tube length, should show a slight negative curvature, as has already been shown in the experimental work on the time lag (Reference 5).

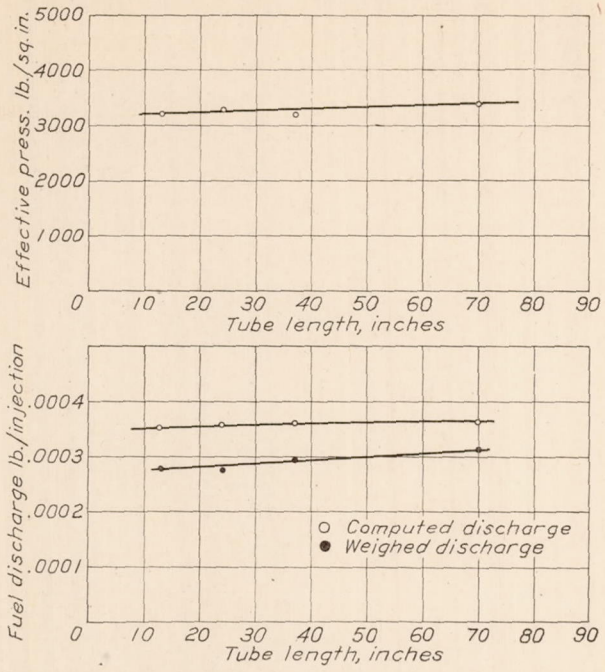


FIGURE 14.—Effect of tube length on effective pressure and fuel discharged

As the tube length was increased the pressure and the lift at the point of cut-off decreased, the bouncing of the stem after cut-off decreased and with it the tendency towards secondary discharges. This is in accordance with the work of Beardsley (Reference 7), in which he found that with a tube length of 7 inches secondary discharges appeared, but with a tube length of 41 inches there were no secondary discharges. It can be concluded that secondary discharges will occur when spray cut-off occurs during the peak of a pressure wave.

These curves also substantiate Beardsley's results (Reference 7) which showed that the spray penetration was slightly increased with an increase in tube length. The penetration of the spray is largely dependent on the maximum pressure of the initial wave and the time required to reach this maximum, but is aided to some extent by the pressures which follow this maximum. An examination of Figure 13 shows that the maximum pressures and the time required to reach these pressures were nearly the same for the 13, 24, and 37 inch tubes. However, as the tube length was increased, the rate of pressure drop after the maximum was reached decreased and the resultant penetration was, therefore, greater. With the 70-inch tube the maximum pressure was less than with the other tubes, but remained at this maximum value for a longer time.

The curves show that to obtain a high constant pressure with an injection system of this type the period of injection should be one-half the period of the pressure waves in the injection valve tube, or the length of the injection tube in inches should be twice the injection period in ten-thousandths of a second.

The effects of the tube length on the effective pressure and fuel quantities discharged are shown in Figure 14. As Figure 13 indicates, the effective pressure increased with tube length until with the 70-inch tube the effective pressure was approximately the same as the static injection pressure in the reservoir. The curves for the weight of oil discharged show that the weight increased with the tube length. It is interesting to note that with the 70-inch tube, in which the pressure remained virtually constant, there is considerably less deviation between the computed and the actual discharges. This indicates that the oil flow in the shorter tubes did not follow the rapid fluctuations in pressure.

By using a smaller discharge orifice, that is, a smaller ratio of discharge orifice area to restriction area, effective pressures greater than the pressure in the reservoir can be obtained. Figure 15 shows the stem lift and pressure curves for a 0.008-inch orifice and a ratio of

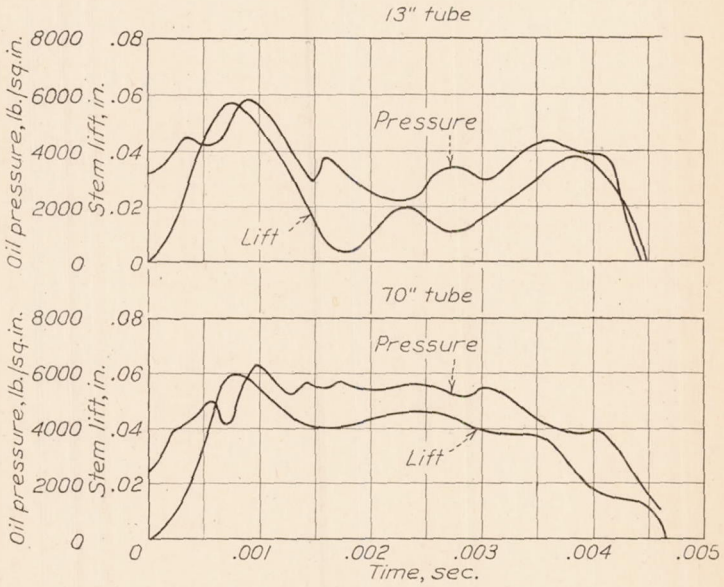


FIGURE 15.—Effect of tube length on stem lift and oil pressure

Injection pressure	3,500 lb./sq. in.
Initial pressure	300 lb./sq. in.
Valve opening pressure	2,000 lb./sq. in.
Orifice diameter	0.008 in.

A/a equal to 0.016 with 13-inch and 70-inch injection tubes. The effective pressure with the 13-inch tube was 3,500 pounds per square inch and for the 70-inch tube was 4,500 pounds per square inch—29 per cent higher than the static injection pressure. Such a small

ratio of discharge orifice diameter to injection valve tube diameter would, however, not be practicable from a construction standpoint.

Effect of Timing-valve Camshaft Speed and Injection Period.—The effect of the timing-valve camshaft speed and the injection period on the pressure fluctuations is shown in Figure 16. As the timing-valve camshaft speed was decreased, the maximum pressure reached on the first and second waves decreased slightly. The pressure wave was damped out before the third oscillation, so that for the lowest r. p. m. the pressure after the first 0.0045 second remained virtually constant at the pressure in the reservoir. Figure 17 shows the effect of the camshaft speed and injection period on the effective pressure and the fuel quantity discharged. The effective pressure dropped from 3,300 pounds per square inch at 1,140 r. p. m. to 3,150 pounds per square inch at 480 r. p. m. As the rate of fuel discharge varies as the square root of the pressure, the mean rate of discharge at 480 r. p. m. is only 2 per cent less than at 1,140 r. p. m. It can be concluded from these curves that for an injection system of this type the fuel spray penetration and distribution, except for the effect of air flow, and the time for delivering a given fuel quantity, are practically independent of engine speed.

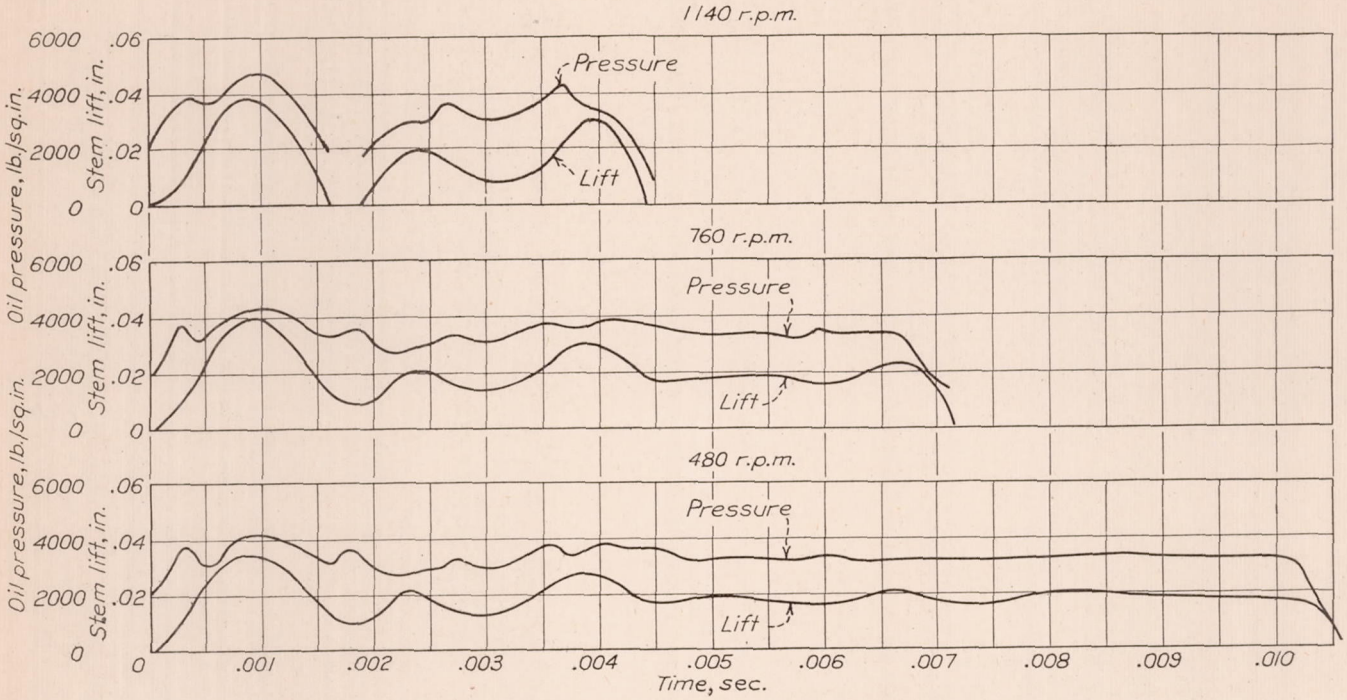


FIGURE 16.—Effect of timing valve camshaft speed and injection period on stem lift and oil pressure

Injection pressure	3,500 lb./sq. in.	Tube length	13 in.
Initial pressure	300 lb./sq. in.	Orifice diameter	0.020 in.
Valve opening pressure	2,500 lb./sq. in.		

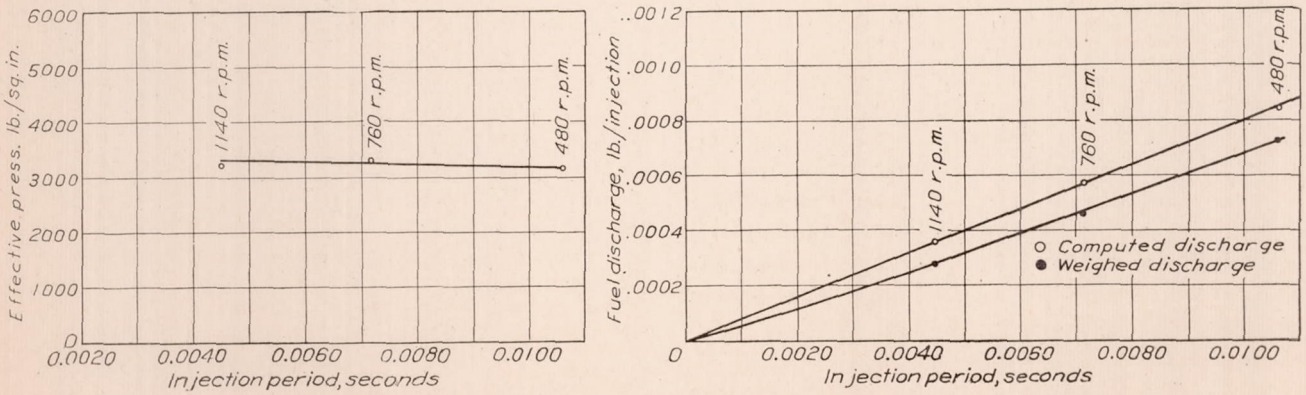


FIGURE 17.—Effect of timing valve camshaft speed and injection period on effective pressure and fuel discharged

Effect of High-pressure Reservoir Volume.—Another factor in the design of a common-rail fuel injection system is the volume of the high-pressure reservoir. The volume should be sufficient so that the pressure drop during injection is small, and at the same time the reservoir should be small enough to be in propor-

tion to the rest of the injection system. For the tests already presented the volume was 20 cubic inches. A test was made with a 10-cubic inch reservoir. The results of this test in comparison to one with the 20-cubic inch reservoir are shown in Figure

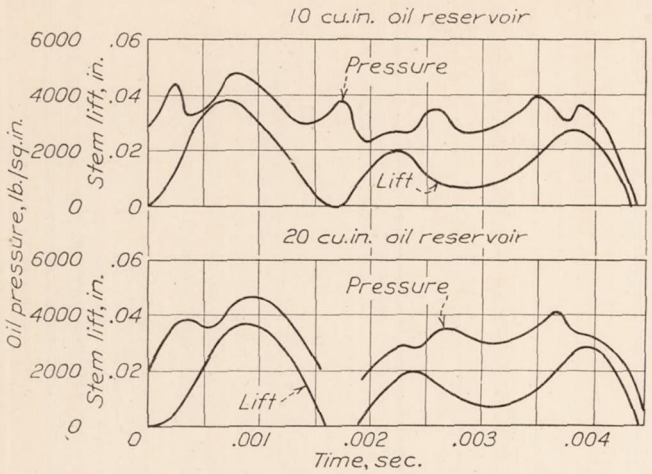


FIGURE 18.—Effect of oil reservoir volume on stem lift and oil pressure

Injection pressure	3,500 lb./sq. in.
Initial pressure	300 lb./sq. in.
Valve opening pressure	2,500 lb./sq. in.
Orifice diameter	0.020 in.
Tube length	13 ins.

18. The stem lift and pressure curves are virtually the same for the two reservoirs. Further research is necessary, however, to determine the minimum high-pressure reservoir volume that can be employed without affecting the pressures at the discharge orifice.

Effect of Initial Tube Pressure.—The effect of the initial pressure in the injection valve tube before the opening of the timing valve on the stem lift and the pressure variations is shown in Figure 19. As the initial pressure was increased, the maximum pressure decreased, and the pressure curve changed from a wave curve to a curve of almost constant pressure. There was, however, little change in the effective pressures—3,100 pounds per square inch for the 300 pounds per square inch initial pressure, 3,300 pounds per square inch for the 1,000 and 2,500 pounds per square inch initial pressures. The total injection period increased with the initial pressure. Therefore, since the setting of the by-pass valve remained the same for all the tests, the time lag of injection decreased as the initial pressure increased. In the report on the time lags of injection (Reference 5) it was stated that the time lag was independent of the initial pressure, providing the injection pressure was considerably in excess of the valve opening pressure. However, the results presented here show that when the valve opening pressure is comparable to the injection pressure the time lag is dependent on the initial pressure.

To maintain an initial pressure in the injection valve tube of an injection system of this type preceding the opening of the timing valve the same pressure must be left in the tube at cut-off of each injection. Consequently, the cut-off of the spray would not be as sharp as shown in Figure 19 where the fuel pressure was released to atmospheric. Beardsley (Reference 7) has shown that the spray penetration increases slightly with the initial pressure. The curves show that this increase as the initial pressure increases is due to the slower rate of pressure drop after the first wave. It can be concluded from the figure that a high initial pressure in the fuel injection line does not present any advantages which can not be obtained by lengthening the injection tube, but on the contrary does cause a slower cut-off of the fuel spray.

Effect of Valve Opening Pressure.—Decreasing the valve opening pressure (fig. 20) had little effect on the stem movement and pressure variations other than to increase the stem lift and so prevent the stem from touching the seat during its oscillations. The injection period increased as the valve opening pressure decreased, as has already been pointed out in the work on the time lags. (Reference 5.) High valve opening pressures cause the fuel spray to start at a higher pressure and consequently some increase in spray penetration should be expected. The effective pressure

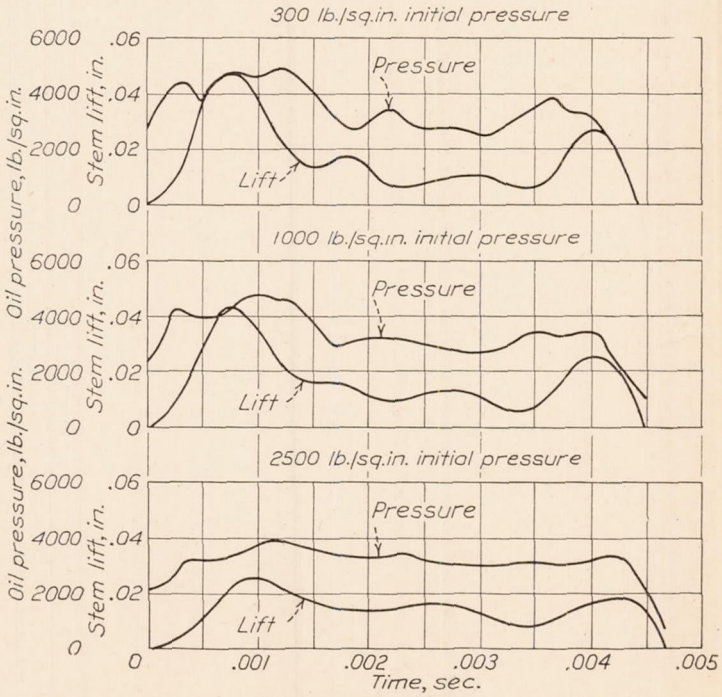


FIGURE 19.—Effect of initial pressure on stem lift and oil pressure

Injection pressure	3,500 lbs./sq. in.
Valve opening pressure	2,500 lbs./sq. in.
Tube length	24 ins.
Orifice diameter	0.020 in.

increased slightly with the valve opening pressure, Figure 21, since the pressures at the start of injection were higher. The increase in the effective pressure

was more than offset by the increase in the time lag and consequent decrease in the injection period. Hence, the fuel quantity discharged decreased slightly with an increase in the valve opening pressure. (Fig. 21.) From these curves it can be concluded that the valve opening pressure has little effect on the effective pressure and the fuel quantity discharged, but does affect the stem lift.

Effect of Injection Pressure.—Figure 22 shows the effects of the injection pressure on the stem lift and pressure variations. In the first example the valve opening pressure was greater than the injection pressure of 1,800 pounds per square inch. However, the first wave from the reservoir built up to a pressure of 2,400 pounds per square inch and consequently lifted the stem slightly. The valve then closed and remained closed until the second wave reached it. The second wave was of less intensity than the first, but was sufficient to again lift the stem. The period of this wave obtained from the figure was 0.0034 second. The computed value is 0.0028 second. With an injection pressure of 2,500 pounds per square inch the valve opened four times during the injection period. With an injection pressure of 3,500 pounds per square inch, as has been shown before, the valve remained open during the whole injection period. When the injection pressure was further increased to 4,000 pounds per square inch the pressure and stem lift curves were of the same form as for the 3,500-pound per square inch injection pressure, but of greater intensity. The effective pressure for the 3,500 and 4,000 pound per square inch injection pressures were 3,300 and 3,900 pounds per square inch, respectively. As the injection pressure increased the injection period increased, due to the shorter time lag of injection. (Reference 5.) The injection pressure to be used in the design of an injection system of this type is based on the time required to inject a given fuel quantity into the engine and the spray penetration, dispersion, and atomization desired. The curves show that unless the total hydraulic force on the stem after it is lifted is sufficiently greater than the spring force on the stem when the valve is closed, the stem will not remain off the seat during the whole injection period.

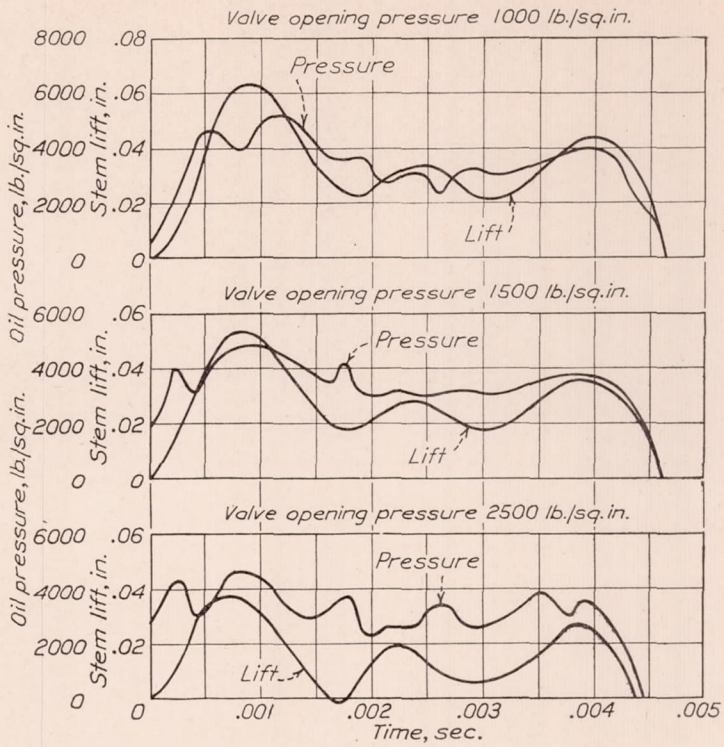


FIGURE 20.—Effect of valve opening pressure on stem lift and oil pressure

Injection pressure	3,500 lb./sq. in.
Initial pressure	300 lb./sq. in.
Tube length	13 in.
Orifice diameter	0.020 in.

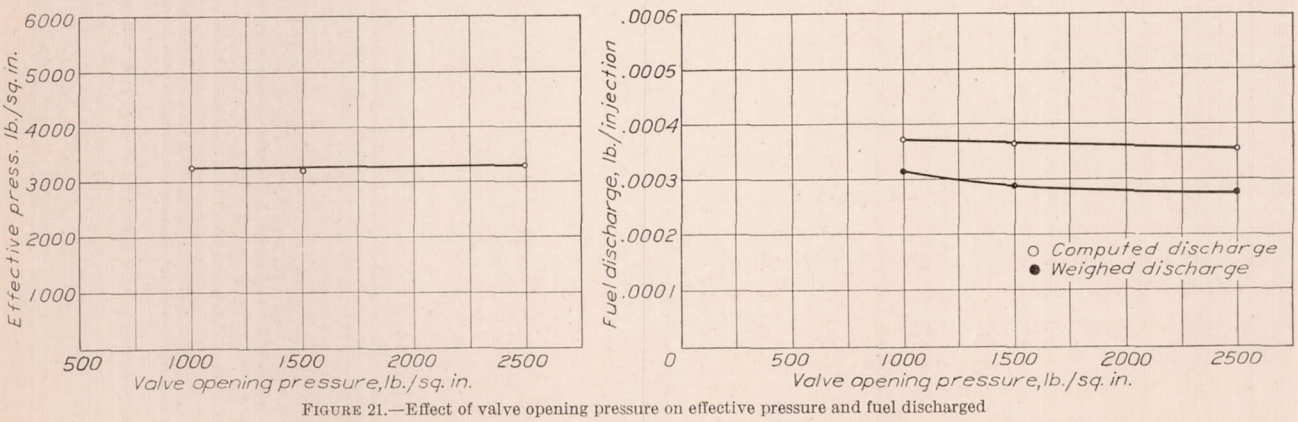


FIGURE 21.—Effect of valve opening pressure on effective pressure and fuel discharged

CONCLUSIONS

There are two main conclusions to be drawn from these tests: First, with a common-rail fuel injection system in which the source of high pressure is at some distance from the discharge orifice, pressure waves will occur between the high-pressure reservoir and the dis-

charge orifice; second, these pressure waves can be controlled in a manner advantageous to the injection system.

Several conclusions can also be drawn as to the design of such an injection system.

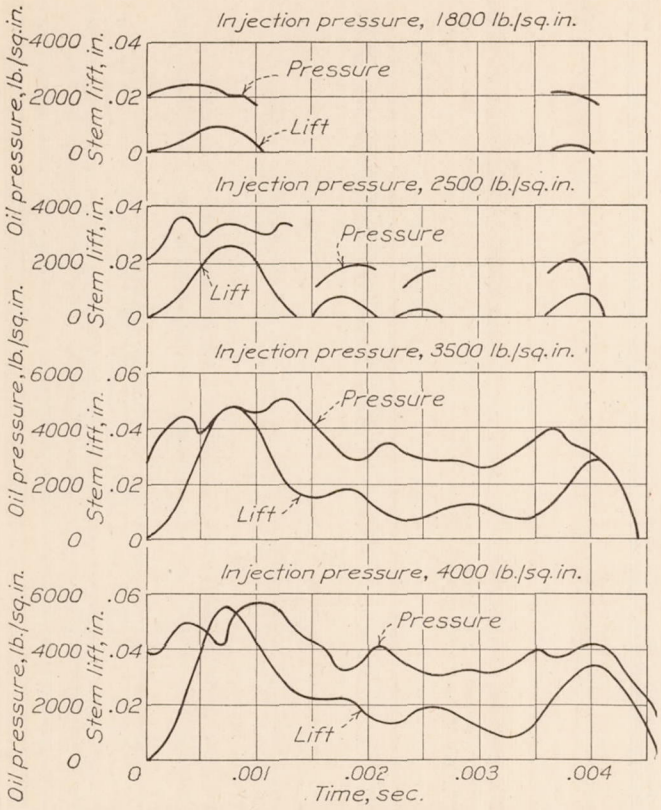


FIGURE 22.—Effect of injection pressure on stem lift and oil pressure

Initial pressure	300 lb./sq. in.
Valve opening pressure	2,500 lb./sq. in.
Tube length	24 in.
Orifice diameter	0.020 in.

1. All oil passages between the high-pressure reservoir and the discharge orifice should have areas at least four times the total area of the discharges orifice.

2. The injection tube connecting the high-pressure reservoir to the discharge orifice should have a length in inches equal to twice the injection period in ten-thousandths of a second.

3. The instantaneous injection pressures are virtually independent of engine speed.

4. The injection period in seconds for a given fuel quantity is independent of engine speed.

5. A high initial or residual pressure in the injection tube has no material advantages to the injection of the fuel.

LANGLEY MEMORIAL AERONAUTICAL LABORATORY,
NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS,
LANGLEY, VA., *May 20, 1930.*

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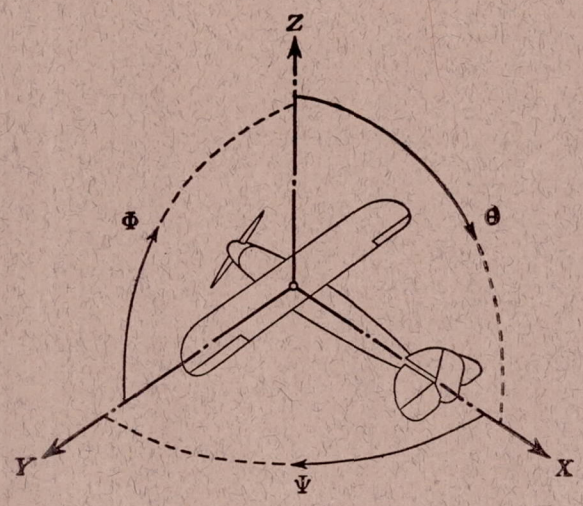

Positive directions of axes and angles (forces and moments) are shown by arrows

Axis		Force (parallel to axis) symbol	Moment about axis			Angle		Velocities	
Designation	Symbol		Designation	Symbol	Positive direction	Designation	Symbol	Linear (component along axis)	Angular
Longitudinal	X	X	rolling	L	$Y \longrightarrow Z$	roll	Φ	u	p
Lateral	Y	Y	pitching	M	$Z \longrightarrow X$	pitch	Θ	v	q
Normal	Z	Z	yawing	N	$X \longrightarrow Y$	yaw	Ψ	w	r

Absolute coefficients of moment

$$C_L = \frac{L}{qbS} \quad C_M = \frac{M}{qcS} \quad C_N = \frac{N}{qfS}$$

Angle of set of control surface (relative to neutral position), δ. (Indicate surface by proper subscript.)

4. PROPELLER SYMBOLS

D, Diameter.
p_e, Effective pitch
p_g, Mean geometric pitch.
p_s, Standard pitch.
p_v, Zero thrust.
p_a, Zero torque.
p/D, Pitch ratio.
V', Inflow velocity.
V_s, Slip stream velocity.

T, Thrust.
Q, Torque.
P, Power.
(If "coefficients" are introduced all units used must be consistent.)
η, Efficiency $= T\,V/P$.
n, Revolutions per sec., r. p. s.
N, Revolutions per minute., R. P. M.
Φ, Effective helix angle $= \tan^{-1}\left(\frac{V}{2\pi rn}\right)$

5. NUMERICAL RELATIONS

1 HP = 76.04 kg/m/sec. = 550 lb./ft./sec.
1 kg/m/sec. = 0.01315 HP.
1 mi./hr. = 0.44704 m/sec.
1 m/sec. = 2.23693 mi./hr.

1 lb. = 0.4535924277 kg.
1 kg = 2.2046224 lb.
1 mi. = 1609.35 m = 5280 ft.
1 m = 3.2808333 ft.