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DISCHARGE CHARACTERISTICS OF A DOUBLE INJECTION-VALVE SINGLE-PUMP INJECTION SYSTEM

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SUMMARY

The discharge characteristics of two similar injection valves operated by a single-cylinder fuel-injection pump were determined with an apparatus that measured the quantity of fuel discharged from each valve during every 0.5° of pump rotation. It was found that similar discharges took place from the two valves at all pump speeds when the valve-opening pressures, the nozzle-orifice diameters, and the injection-tube lengths were the same for both valves. Under these conditions, the effects of changing the pump speed, the pump throttle setting, or the nozzle-orifice diameter were very similar to those occurring with a single-injection valve. By a proper selection of discharge-orifice areas and valve-opening pressures it was possible to obtain a great many combinations of discharge quantities, discharge rates, and injection timings for the two valves. A series of tests using injection tubes of unequal lengths for the two valves showed that under these conditions the injection timing and the fuel quantity discharged from each valve varied widely and erratically with changes in the pump speed.

INTRODUCTION

Probably the most difficult problem encountered in the development of each new design of high-speed compression-ignition engine is the uniform distribution of the injected fuel to all of the air in the combustion chamber. The distribution may be improved in two general ways: by increasing the velocity and changing the direction of the air movement within the chambers; and by changing the characteristics, number, and location of the fuel sprays. A number of engines using more than one injection valve per cylinder have been developed, the Junkers Jumo 205-C with four valves and the Clerget 14F-2 and the Mercedes-Benz OF-2 with two valves per cylinder being outstanding examples. In these engines, two valves are operated by each pump unit (reference 1).

The single-cylinder compression-ignition test engines used by the N.A.C.A. are usually constructed so that injection valves may be located at three or more places in the cylinder head. For several series of tests two injection valves have been used simultaneously, both valves being operated by the same pump. In each case, however, the tests were of a supplementary nature and were not continued long enough to warrant general conclusions on the value of such an arrangement. Preliminary to further engine tests using more than one injection valve, the timing and rate-of-discharge characteristics of a double-valve single-pump injection system were determined, and the results are presented in this report.

APPARATUS

Two injection valves with differential-area lapped stems loaded by helical springs were operated by a single-cylinder Bosch pump having a 10-millimeter diameter plunger. (Cross sections of an injection valve and a pump essentially the same as those used for the present test are shown in figs. 1 and 2 of reference 2.) The injection tubing was composed of three parts joined together by a Y-shaped connecting block, the pump being connected to the stem of the Y and the injection valves connected to the branches. The distance from the pump plunger to the beginning of the tubing was 6 inches, and the length of the fuel passages in each injection valve was 2.5 inches. (See fig. 1(a).)

Single-orifice nozzles were used with each valve. The nominal orifice diameters were 0.015, 0.022, 0.033, and 0.040 inch, and each nozzle had an orifice length of 0.137 inch. The two injection valves and their sets of nozzles were intended to be identical, but the following small variations were found: Valve 1 had a stem diameter of 0.225 inch and a spring constant of 605 pounds per inch of deflection. Valve 2 had a stem diameter of 0.223 inch and a spring constant of 565 pounds per inch of deflection. The nozzle used in valve 1 and designed to have an orifice diameter of 0.015 inch actually had an orifice diameter of 0.017 inch.

The tests were conducted with the apparatus described in reference 3, slightly altered to allow the two valves to be simultaneously tested. A narrow slot in a steel disk rotating between the valve nozzles and the two fuel collecting cups allowed the discharge from the nozzles to collect within the cups for only 0.5° of pump rotation. An electric counter recorded the number of injections, and from this value and the weight of the fuel collected by the cups the amount of fuel discharged per pump degree per cycle was computed for each valve. A Diesel fuel with a viscosity of 0.052 poise at 22° C. and a specific gravity of 0.831 at 15° C. was used for the tests.

The standard test conditions used, unless otherwise stated, were as follows:

Pump speed	750 r.p.m.
Valve-opening pressure	3,000 lb./sq. in.
Maximum stem lift	0.018 in.
Nozzle-orifice diameter	0.022 in.
Pump throttle setting	0.5 full throttle
Fuel pressure at inlet to pump .	50 lb./sq. in.
Inside diameter of injection tubes	1/8 in.
Length of main injection tube . .	30 in.
Length of each branch injection tube	11 in.

TESTS AND RESULTS

Rate-of-Discharge Tests

The test results are plotted (figs. 2-9) as fuel discharge in pounds per degree against degrees of pump rotation. The zero of the abscissa scale represents the closing of the fuel inlet ports by the top of the pump plunger, so that the injection lag may be read directly in pump degrees. The position at which the port was opened by the helical edge on the pump plunger, representing cut-off at the pump, is also indicated. The weight per injection given in each case is the total discharge from both valves, obtained by weighing the fuel supplied to the pump and counting the number of pump cycles. The values under the curves denote the weight of fuel ($\times 10^{-5}$ pound) separately discharged from the two valves determined by integrating the curves. It will be noted that the sum of the values under the curves is less than the weight per injection except at 250 r.p.m. At this speed the fewer number of injections caught might result in a decreased accuracy in measurement.

The first three series of tests were made with similar nozzles in the two injection valves, with the valve-opening pressures set as nearly equal as possible, and with branch tubes of equal lengths. Figures 2, 3, and 4 show the effects of changing the pump speed, the pump throttle setting, and the nozzle-orifice diameters, respectively, on the discharge characteristics of the valves. The results in each case are very similar to those previously obtained with another Bosch pump and a single-injection valve of the same type (reference 3).

Throttling of the fuel flow may occur at two places in an injection valve of the type used for these tests, between the stem and its seat and at the orifice. When the orifice area is small compared with the flow area at the stem seat most of the throttling is done at the orifice, but with large orifices a considerable portion of the throttling occurs at the stem seat. Valve 1 had a stiffer spring than valve 2 and therefore at any given fuel pressure its stem lifted a lesser amount than the stem in valve 2. (The difference in stem diameters only partly counteracted this effect.) Figure 4 shows that valve 1 discharged less fuel than valve 2 when the two larger pairs of identical orifices were used, but more fuel when the 0.017- and 0.015-inch orifices were used in valves 1 and 2, respectively. The results indicate that throttling at the stem seats was important when pairs of nozzles, each having an orifice diameter of 0.022 or 0.040 inch, were used, but when the smallest pair of orifices was used, the discharges were controlled almost entirely at the orifices.

In figure 5 the data on the effect of orifice diameter have been replotted, this time adding the separate discharges from the two valves. These total discharge rates for the system are then compared with the rates obtained when using only valve 2 with a single orifice, the area of which was in each case very nearly equal to the sum of the orifice areas in the two valves. For these comparative tests the Y connecting block and branch tubes were replaced by a plain connection and a tube of the same length as either branch tube and having a flow area slightly greater than the total flow area of both. (See fig. 1(b).) The figure shows that the timing of the sprays and the quantity of fuel discharged from the double-valve system were about the same as with a single-valve system under similar conditions and with the same total discharge-orifice area. The rates of fuel discharge, however, were somewhat different.

The next series of tests was made with dissimilar conditions for the two injection valves. Figure 6 shows the effect of lowering the opening pressure of valve 2 to 2,500 and 2,000 pounds per square inch while maintaining that of valve 1 at 3,000, all other conditions remaining the same. Lowering the opening pressure of valve 2 advanced the injection timing for that valve, retarded that for valve 1, increased the amount of fuel discharged from valve 2, and reduced that from valve 1. The time of spray stop remained practically unchanged. In several cases a preliminary discharge from valve 1 occurred ahead of the main discharge. Apparently when valve 1 was opened by a pressure wave from the pump, the much greater rate of flow already taking place through valve 2 postponed the building up of the static pressure. An increase in the intensity of the pressure wave caused by increasing the pump speed or a lowering of the valve-opening pressure should result in valve 1 remaining open during the entire discharge period. The summations of the discharge curves for both valves (at the right in fig. 6) show maximum rates occurring close to the ends of the discharge periods.

Figure 7 shows the effect of using unequal orifice diameters in a double-valve system. In each test a 0.015-inch diameter orifice was used in valve 2. For the first three tests the valve-opening pressure was kept the same for both valves and orifice diameters of 0.022, 0.033, and 0.040 inch were used in valve 1.

The results indicate that, when the 0.022-inch orifice was used in valve 1, the amount of fuel discharged from each orifice was very nearly proportional to its area but that, when larger orifices were used in valve 1, it did not discharge a proportionately greater quantity of fuel. The explanation of this behavior is the same as previously mentioned; with the larger orifice sizes the discharge was reduced by the throttling area between the valve stem and the seat. The use of unequal orifice areas in the two injection valves had practically no effect on the timing of the start and stop of fuel discharge.

For the last test of this series, a 0.033-inch orifice was used in valve 1 and the opening pressure of valve 2 was reduced to 2,000 pounds per square inch. As shown in the lower part of figure 7, the results were similar to those of the previous tests for the effect of changing the valve-opening pressure but were less pronounced because of the unequal orifice areas. By a proper selection of orifice areas and valve-opening pressures, it would be possible to keep the discharge weights from the two valves equal but to advance the timing of the spray start from one orifice with respect to the other.

Another generally accepted means of changing the injection lag is to use injection tubes of different lengths. A series of tests was made with the branch tube leading to valve 2 increased in length from 11 to 36-1/4 inches and with all other conditions the same for both valves. (See fig. 1(c).) As figure 8 shows, the valve at the end of the shorter tube did not always begin injecting first. At a pump speed of 750 r.p.m. and a valve-opening pressure of 1,000 pounds per square inch, valve 1 began injecting 2° ahead of valve 2. At the same speed but at valve-opening pressures of either 2,000 or 3,000 pounds per square inch, valve 2 began injecting about 2.5° ahead of valve 1. However, when the pump speed was raised to 1,330 r.p.m. and the valve-opening pressure of both valves kept at 3,000 pounds per square inch (fig. 9), valve 1 began injecting about 3.5° ahead of valve 2. At pump speeds below 750 r.p.m., the injections were very uneven and, at 250 r.p.m. and a valve-opening pressure of 3,000 pounds per square inch, valve 2 stopped operating and the entire discharge came from valve 1. This erratic behavior was caused by complex reflections and divisions of the pressure waves that are always present in high-speed injection systems as discussed in reference 4 and in the next section of this paper.

Spray-Timing Tests

After the rate-of-discharge tests were completed, the two valves were so mounted that the time lags of the start and stop of the fuel sprays could be measured by means of a Stroborama. Branch tubes of different lengths were used over a pump-speed range from 200 to 1,000 r.p.m.

Figure 10 shows the results of spray-timing tests using the same injection-valve conditions and tube lengths as were used in obtaining the rate-of-discharge data shown in figure 8. In the analysis of these curves it is important to remember that a complicated system of pressure waves existed in the injection tubing because of the Y arrangement of the three segments a, b, and c as shown in figure 1(c). The minimum injection lag that could be obtained was the time required for the initial pressure-wave front, originating at the pump plunger at the time the pump port closed, to traverse the injection tube to the injection valves. For injection valve 1 this time is designated as $t_a + t_b$, and for injection valve 2 as $t_a + t_c$. The first value, assuming a wave velocity of 52,000 inches per second (reference 5), is 0.95×10^{-3} second and the second value 1.44×10^{-3} second. An examination of the curves in figure 10 shows that, with the lowest injection valve-opening pressure used and at pump speeds of 1,000 r.p.m. and over, the injection lags corresponded to those minimum values. For all other conditions the injection lags were greater than the minimum values, and showed a general increase with decreasing pump speed. The curves are not smooth and they cross and recross each other; one valve injecting first and then the other.

In an analysis of these curves the reflection of the wave front from both the injection valves as well as from the pump plunger must be considered. For instance, in those cases in which the time lag was greater than the minimum already given, the initial wave is reflected from the injection valves in full intensity. Consider a specific case. The initial wave starts from the pump at the closing of the pump port. It reaches the Y and since tubes b and c are of the same internal diameter as a, the wave is divided into two equal parts, one part traveling along b and the other along c. When the wave traveling along b reaches the injection valve and does not open it, the wave is reflected in full intensity back along b. When this reflected wave reaches the Y, it is again divided into two equal parts, one traveling along a and the other along c. Considering the maximum travel of the initial wave front with a single reflection at each tube end, the time is represented for valve 1 as $t_a + t_b + t_b + t_c + t_c + t_a + t_a + t_b$ or $3t_a + 3t_b + 2t_c$, which is equal to 4.35×10^{-3} second. By similar analogy the maximum time for valve 2 is $3t_a + 2t_b + 3t_c$ or 4.84×10^{-3} second. There are, for each valve, six combinations of travel that are taken by the fractions of the initial wave front before all its energy again reaches either valve. It is the combination of these fractions that causes the irregularities in the values of the injection lag.

These combinations are also responsible for the fact that the injection may start first from injection valve 2, although its injection tube is longer than that for valve 1. Assume that injection will start from either valve provided that the initial wave front originating at the injection pump is reinforced by a single reflected front. This injection lag for valve 1 is $t_a + 2t_c + t_b$ or 2.44×10^{-3} second. For injection valve 2 this lag is $t_a + 2t_b + t_c$ or 1.96×10^{-3} second. For this case the injection valve on the longer tube would have the shorter lag. The data presented in figure 10 show that the injection did start first for injection valve 2 for a considerable range of engine speed and that, as the lag approached a value between 1.5 and 2.0×10^{-3} second, the lags approached a common value and then the injection started first from the shorter tube. Any values of lag less then 1.96×10^{-3} second indicate that the injection valves are being opened before any reflected wave fronts reach them and, provided that the injection valve-opening pressures are equal, the injection must start first from the shorter injection tube.

Spray-timing tests were also made with a special fitting in the Bosch pump, which reduced the distance from the pump plunger to the meeting point of the branch tubes to 2-1/4 inches. The changes in the injection lags with pump speed were no less erratic with this tubing arrangement, shown in figure 1(d), than with the previous arrangement. (See fig. 11.) Reducing the length of the injection tube to valve 2 from 59 to 49 inches resulted in smaller differences in the injection lags for the two valves but valve 2 still injected first at some speeds. When the branch tubes were of equal lengths, the injection lags were practically the same for both valves at all pump speeds.

The foregoing discussion has been confined to the time lag for spray start. The time lag for spray stop is the interval between the opening of the bypass port in the pump and the end of fuel discharge from the valve. It was usually greater than the time required for a wave to travel from the port to the valve because fuel under pressure was trapped in the injection tube by a check valve at the pump exit, and injection continued until this pressure dropped to the closing pressure of the injection valve.

CONCLUSIONS

The following conclusions are obtained from the results of tests made with the double-valve injection system described in this report:

1. Similar fuel discharges took place from two injection valves operated by a single pump unit at all pump speeds provided that the length of the tubing to each valve was the same and the valves were practically identical in all respects.

2. The timing of the fuel sprays and the total quantity of fuel discharged from both valves of the balanced double-valve injection system were about the same as those obtained with a single-valve system having an orifice area equal to the total area of both valves of the double-valve system.

3. The quantities of fuel discharged by the two valves when using nozzles having different orifice areas were proportional to the orifice areas only when all other conditions were the same for both valves and when the orifice areas were small compared to the flow area between the valve stem and seat,

4. By a proper selection of orifice areas and valve-opening pressures it was possible to obtain a great many combinations of discharge quantities, discharge rates, and injection timings for the two valves.

5. The timing of the sprays and the fuel quantity discharged varied widely and erratically with changes in

the pump speed whenever tubes of unequal length were used between the valves and the pump.

Langley Memorial Aeronautical Laboratory,
National Advisory Committee for Aeronautics,
Langley Field, Va., April 22, 1937.

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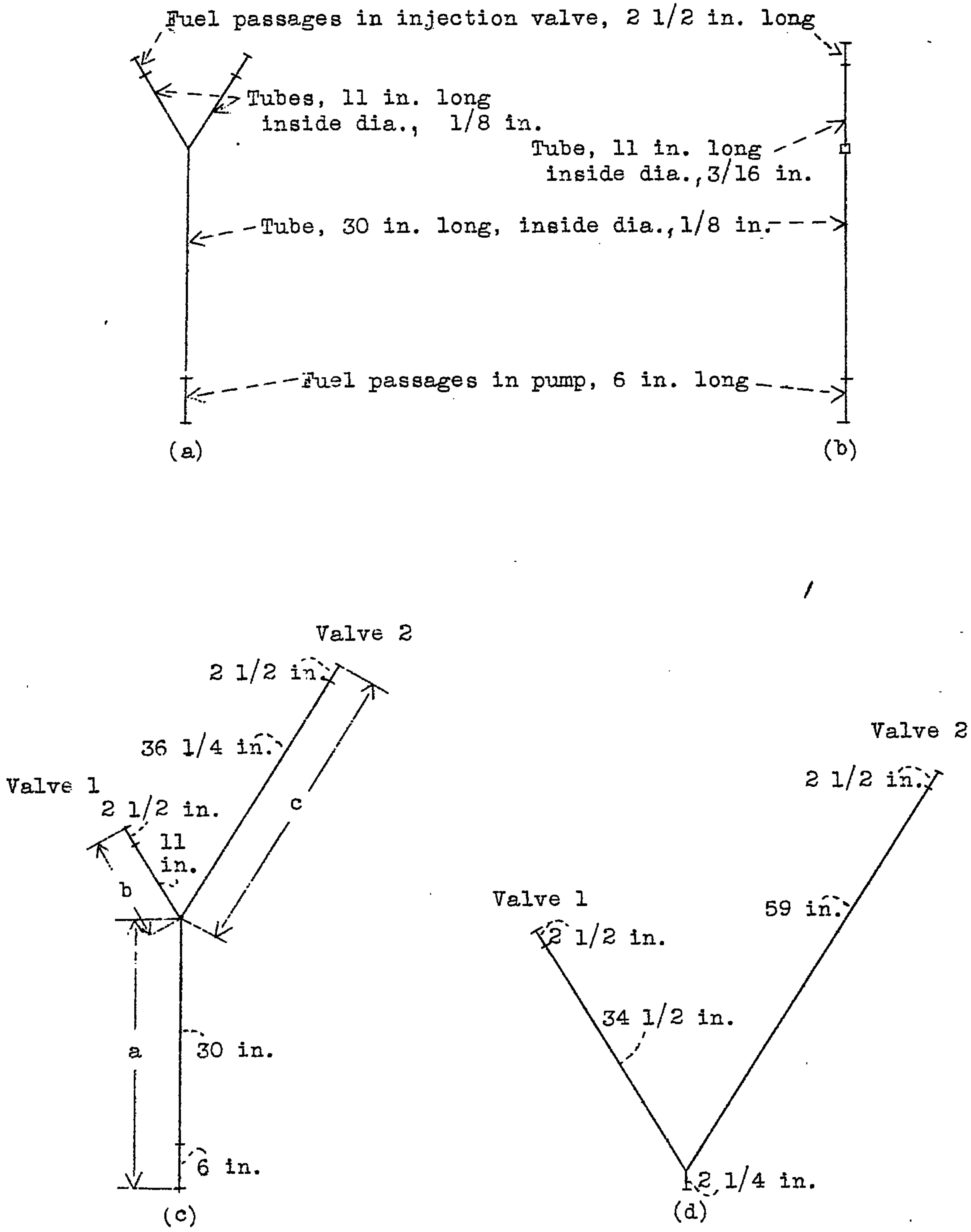

Figure 1.- Injection-tubing arrangements used during the tests.

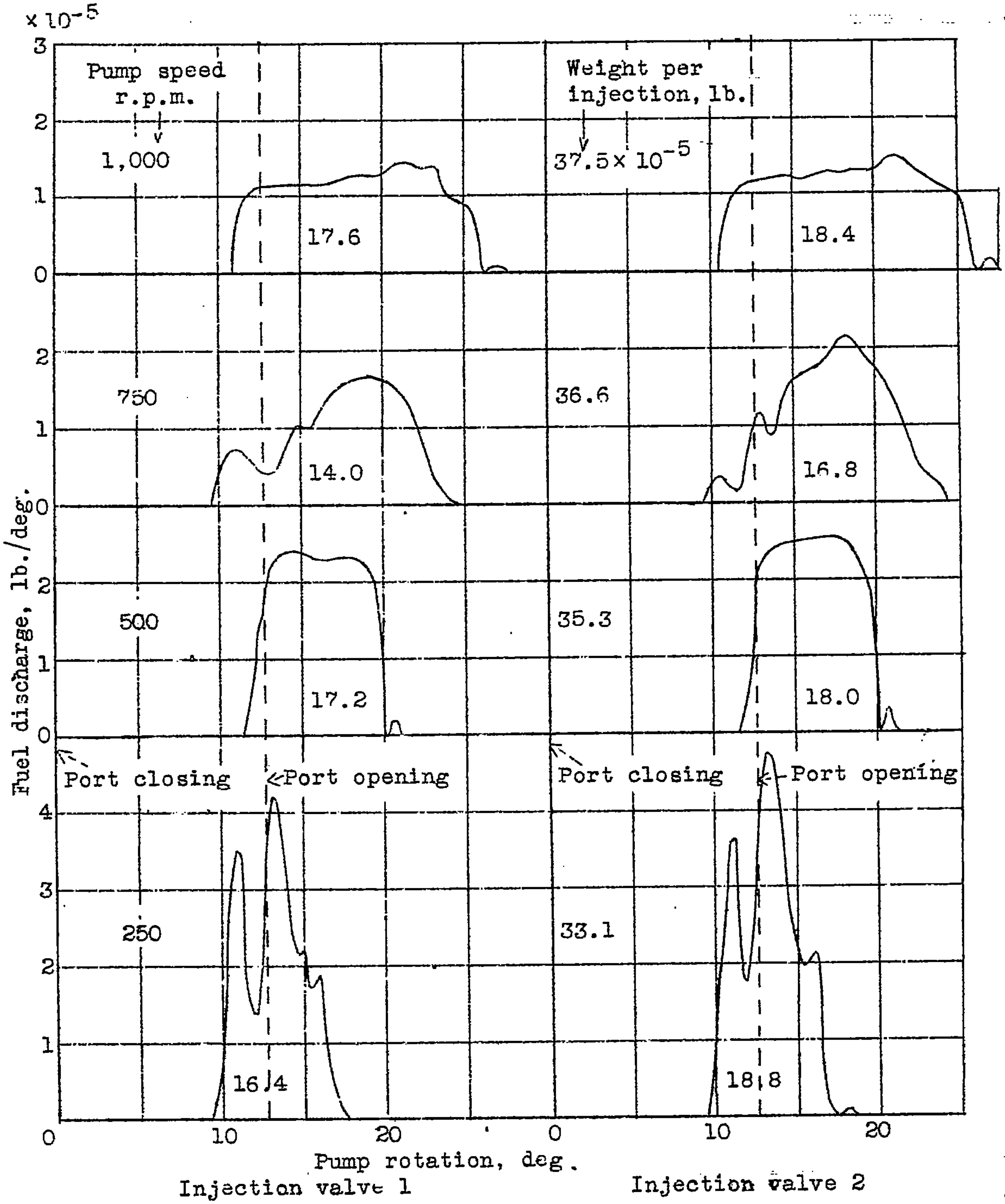

Figure 2.- Effect of pump speed on the discharge characteristics of a double-valve injection system. Valve-opening pressure, 3,000 lb./sq.in.; orifice diameter, 0.022 in.; throttle setting, 0.5; branch tubes of equal lengths.

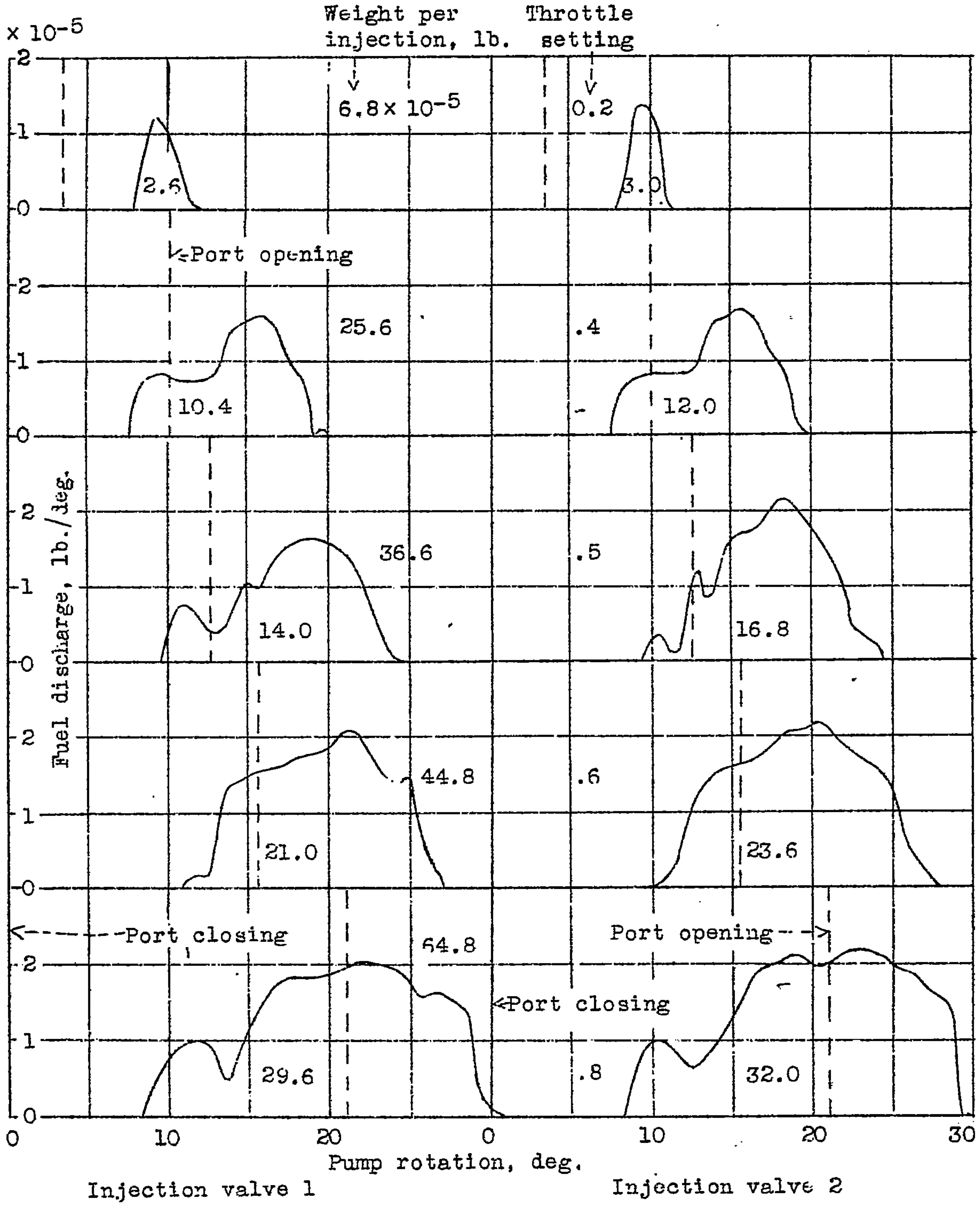

Figure 3.- Effect of throttle setting on the discharge characteristics of a double-valve injection system. Pump speed, 750 r.p.m.; valve-opening pressure, 3,000 lb./sq.in.; orifice diameter, 0.022 in.; branch tubes of equal lengths.

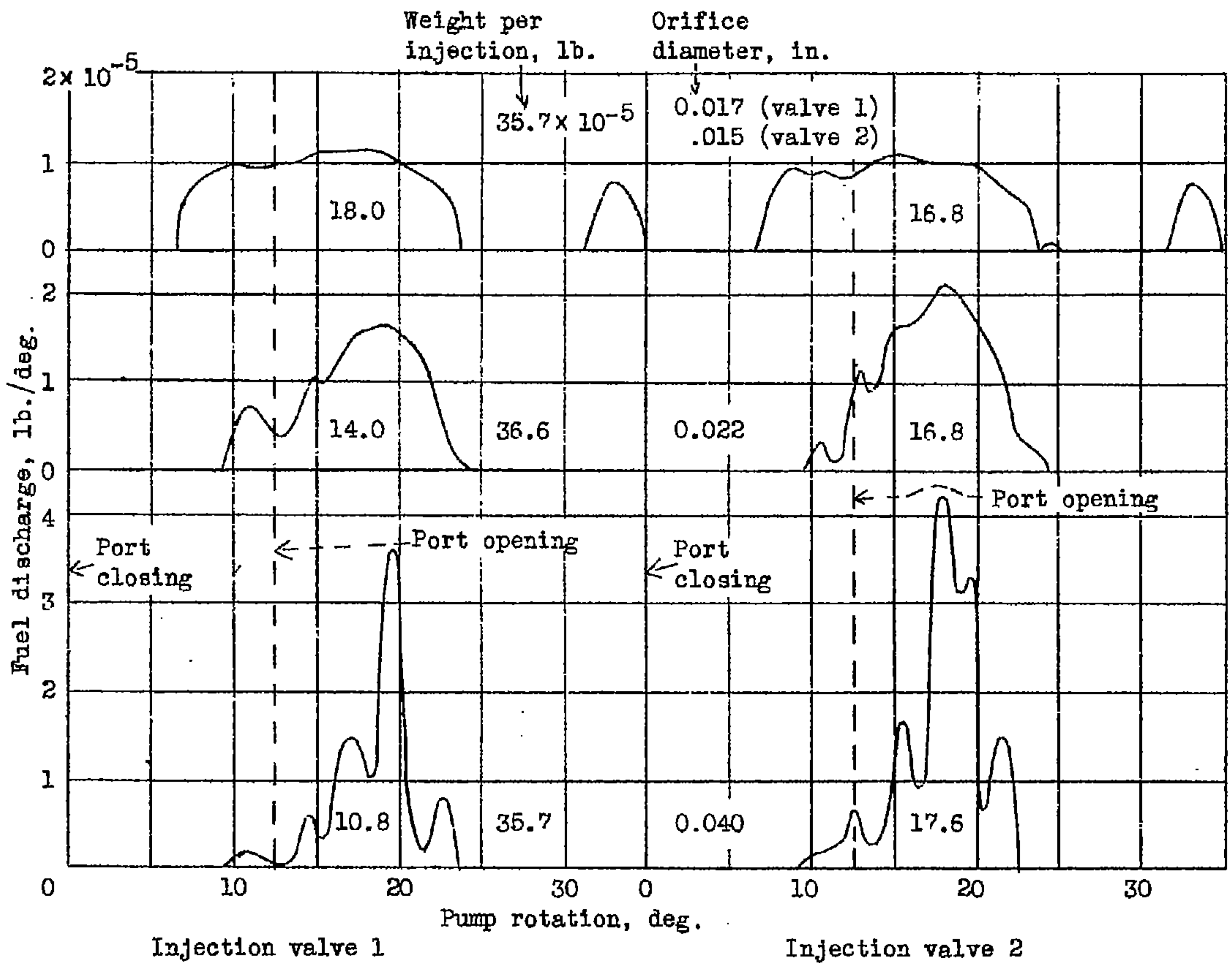

Figure 4.- Effect of orifice diameter on the discharge characteristics of a double-valve injection system. Pump speed, 750 r.p.m.; valve-opening pressure, 3,000 lb./sq.in. throttle setting, 0.5; branch tubes or equal lengths.

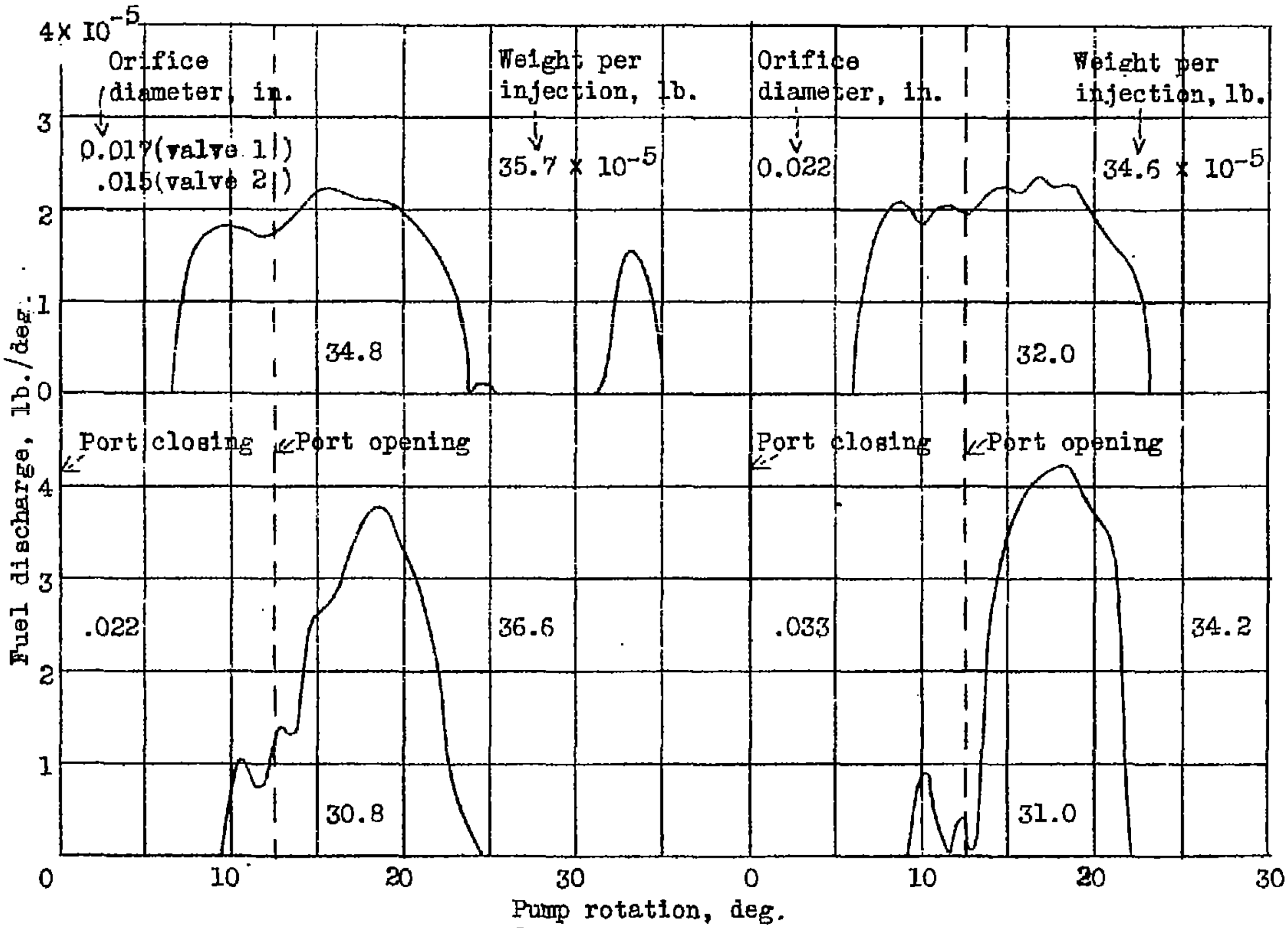

Figure 5.- Total discharge from two valves with equal orifice diameters compared with that from one valve with equivalent orifice area. Pump speed, 750 r.p.m.; valve-opening pressure, 3,000 lb./sq.in.; throttle setting, 0.5

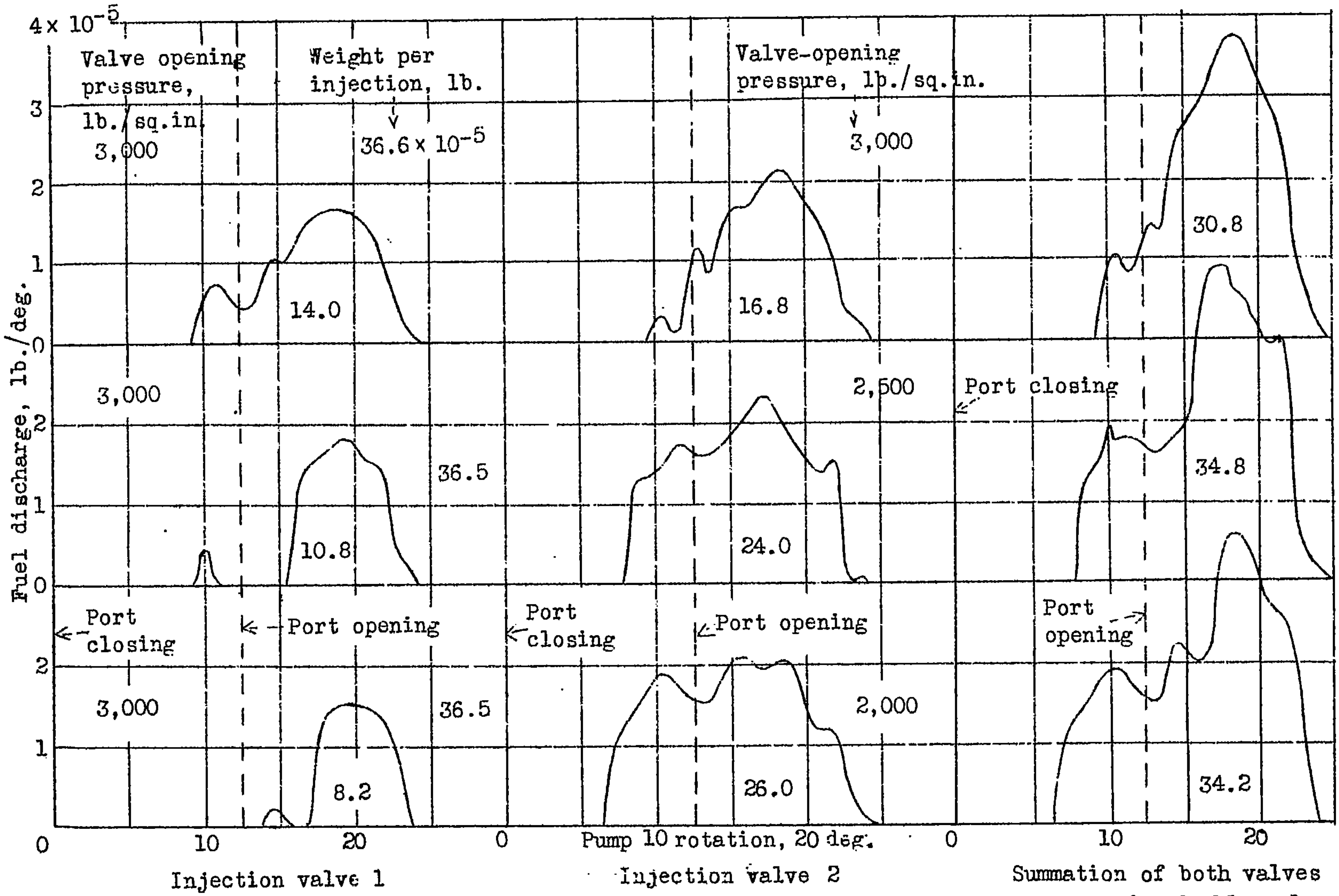

Figure 6.- Effect of unequal valve-opening pressures on the discharge characteristics of a double-valve injection system. Pump speed, 750 r.p.m.; orifice diameter, 0.022 in.; throttle setting, 0.5; branch tubes of equal lengths.

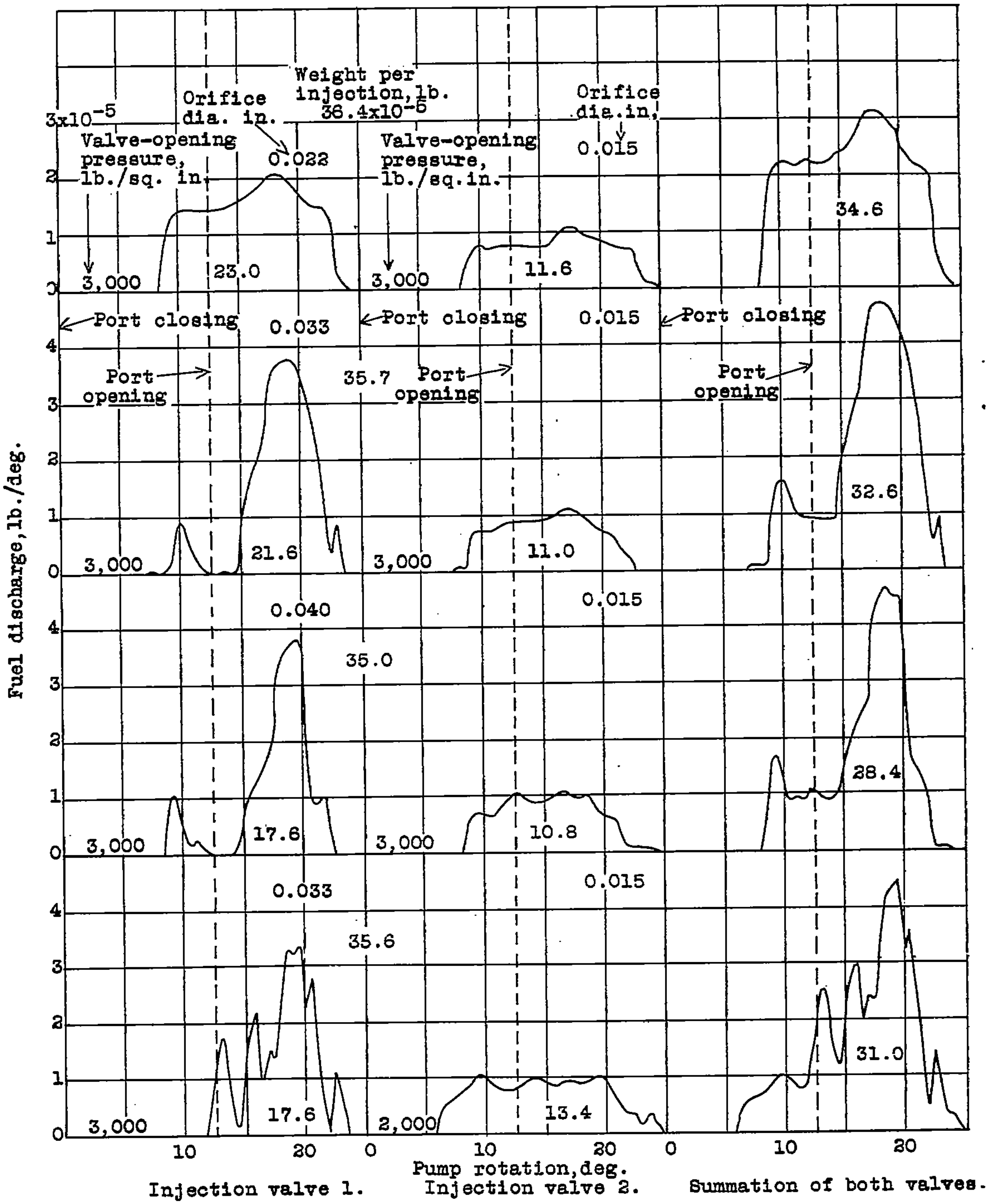

Figure 7.- Effect of unequal orifice diameters on the discharge characteristics of a double-valve injection system. Pump speed, 750 r.p.m.; throttle setting, 0.5; branch tubes of equal lengths.

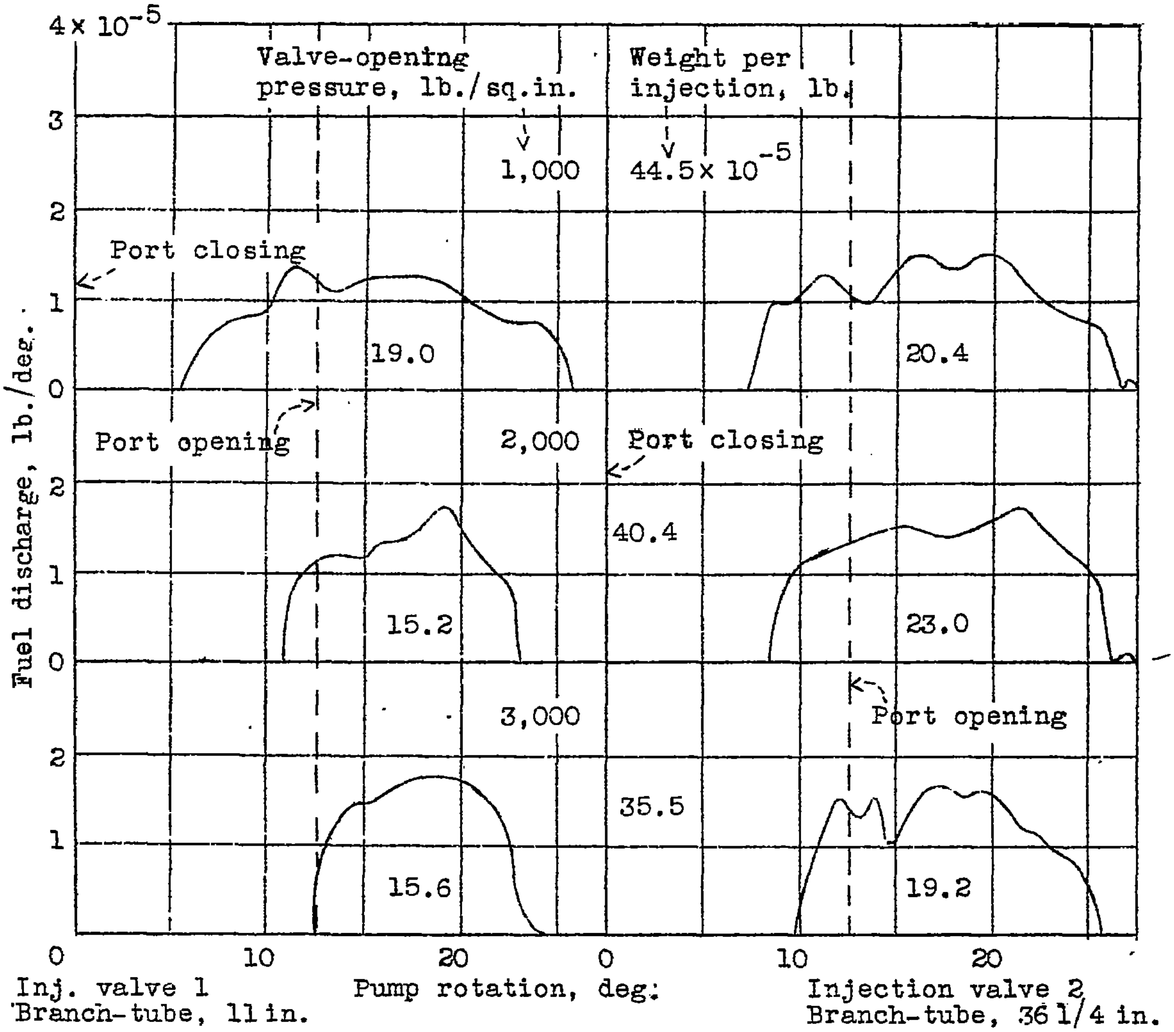

Figure 8.- Effect of valve-opening pressure on the discharge charcteristics of a double-valve injection system using branch tubes of unequal lengths. Pump speed, 750 r.p.m.; orifice diameter, 0.022 in.; throttle setting, 0.5

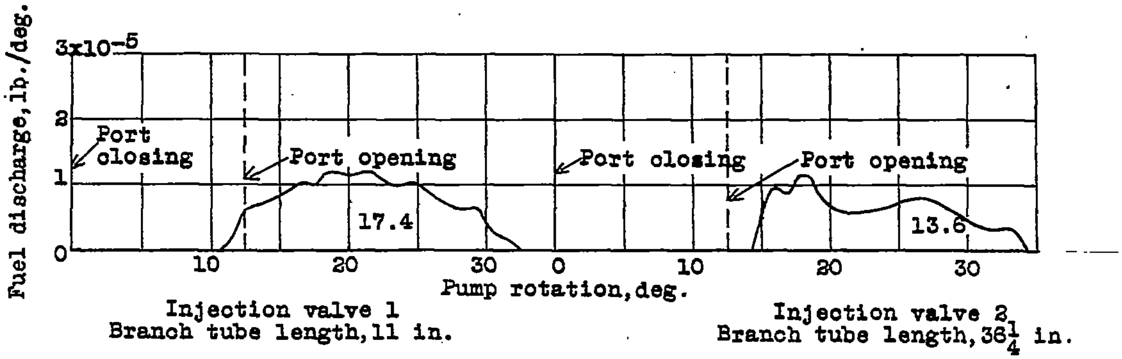

Figure 9.- Discharge characteristics of a double-valve injection system using branch tubes of unequal lengths. Pump speed, 1,330 r.p.m.; valve-opening pressure, 3,000 lb./sq.in.; orifice diameter, 0.022 in.; throttle setting, 0.5; weight per injection, 39.0×10^{-5} lb.

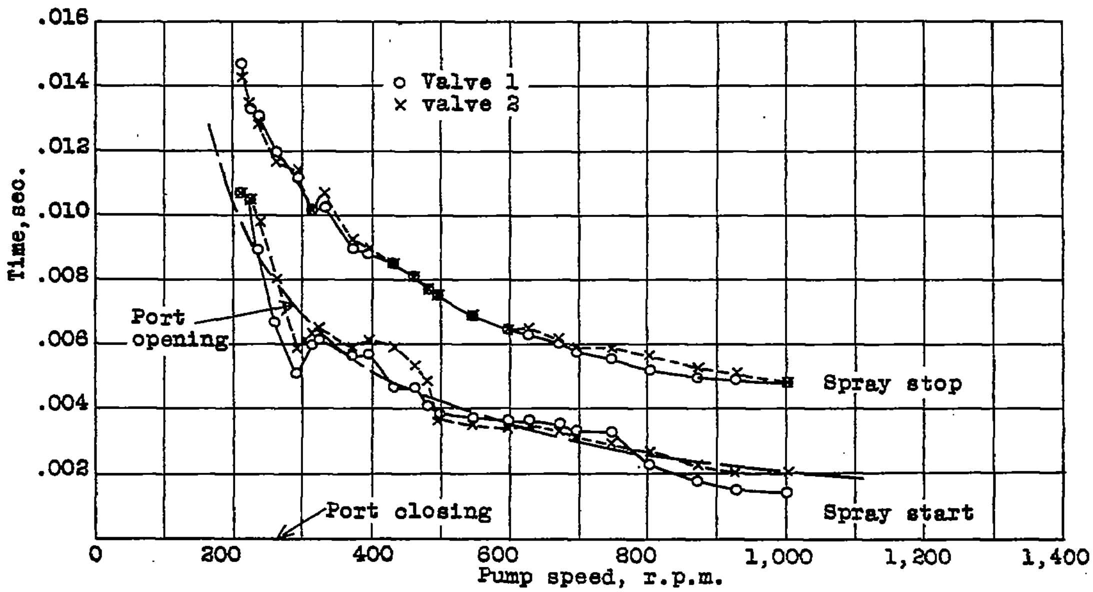

Figure 11.- Effect of pump speed on the time lag for spray start and stop for a double-valve injection system using branch tubes of unequal lengths. Valve-opening pressure, 3,000 lb./sq.in.; orifice diameter, 0.022 in.; throttle setting, 0.5; length of branch tube to valve 1, 34-1/2 in.; length of branch tube to valve 2, 59 in.

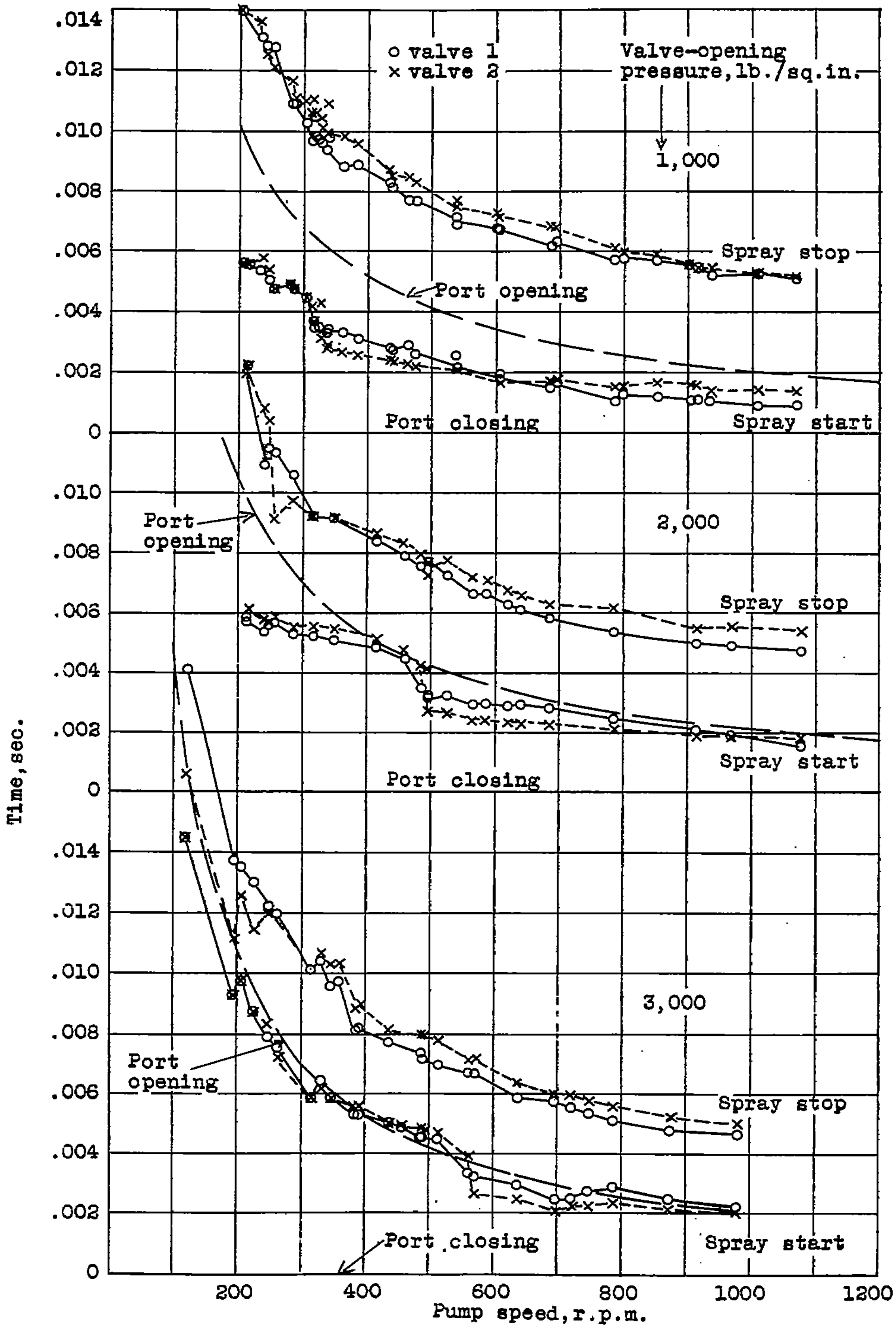

Figure 10.- Effect of pump speed and valve-opening pressure on the time lag for spray start and stop for a double-valve injection system using branch tubes of unequal lengths. Orifice diameter, 0.022 in.; throttle setting, 0.5; length of branch tube to valve 1, 11 in.; length of branch tube to valve 2, 36-1/4 in.