5

Injection Characteristics

Performance factors such as torque, speed control, fuel consumption, and exhaust smoke of diesel or other types of fuel injection engines are largely dependent on the injection characteristics. It is, therefore, important to identify these characteristics and to know how they can be controlled.

FUEL DELIVERY CHARACTERISTICS

The speed-delivery characteristic is important because of its influence on the engine torque curve and ease of governing. As shown in Fig. 65, the volumetric efficiency of an engine generally decreases with an increase in speed, because of increasing resistance to air flow through the intake manifold and valves and inertia of the air in the intake system. Thus, it would appear that if the fuel delivery could be made to match the shape of the volumetric efficiency curve a desirable torque curve should result. The torque, however, also depends on the fuel consumption of the engine so that the fuel delivery must be altered accordingly, as shown in Fig. 66. According to this graph the fuel delivery on the bench should be set as much as 10 percent higher than that calculated, to account for the effect of injecting against compression pressure in the engine. Normally, a flat or slightly declining fuel delivery curve will give a satisfactory torque curve, but for turbo-charged engines an inclining delivery characteristic is required because of reduced air flow at low speeds. With torque control governors it is possible to alter the delivery from the pump to obtain a desired torque curve. A declining fuel delivery with increase in speed is desirable since it helps to make the engine inherently self governing, so that simpler and less expensive governor is required.

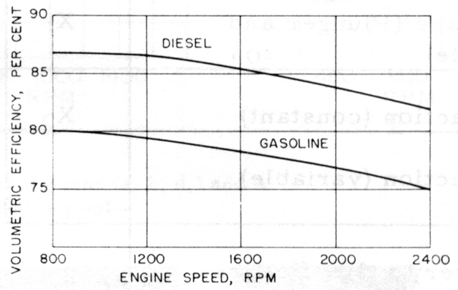

Fig. 65. Volumetric efficiency of typical diesel and gasoline engines.

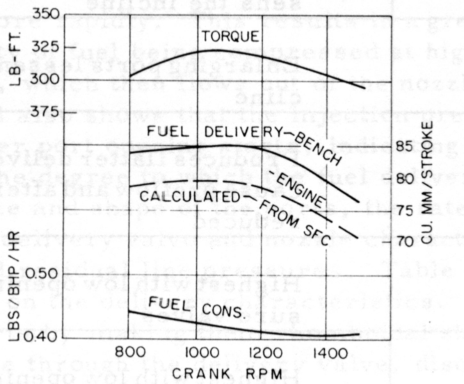

Fig. 66. Fuel delivery characteristics of injection equipment for a diesel tractor engine.

TABLE 4

Factors affecting speed- delivery characteristic of ported pumps

Factor	Influence on delivery			Remarks
	Slight	Moderate	Heavy	
Incomplete filling			X	Not tolerable because of tendency to irregular injection
Fuel compressibility	X			Volume and pressure differential too small
Leakage (Plunger and Nozzle)	X			Leakage only a small fraction of total delivery
Retraction (constant)	X			
Retraction (variable)			X	The larger the clearance the flatter the delivery
Delivery valve-flow restriction and opening pressure		X		Increasing these gives less decline with variable relief valves and less incline with standard valves
Nozzle orifice area		X		Increasing the orifice area gives more incline
Nozzle-opening pressure		X		Lowering the pressure lessens the incline
Ports - Round		X		Enlarging ports lessens incline
Ports - Rhomboidal		X		Produces flatter delivery because preflow and afterflow reduced
Backflow		X		Highest with low opening pressure valves
Afterflow		X		Highest with low opening pressure valves and nozzles and large orifice.
Preflow	X			Highest with low opening pressure valves and nozzle

Ported Pumps

Most pumps of this type, depending on the delivery valve and nozzles used, give an inclining delivery characteristic, as shown in Fig. 67a. One reason for thisis that the throttling of the back flow of fuel through the ports occurs. and delivery of fuel commences before the ports are fully closed. The pressure build up in a ported pump is clearly shown in the oscillograms of Fig. 68 for the instantaneous pressures above the plunger. At higher speeds the throttling effect increases and fuel delivery begins even earlier.

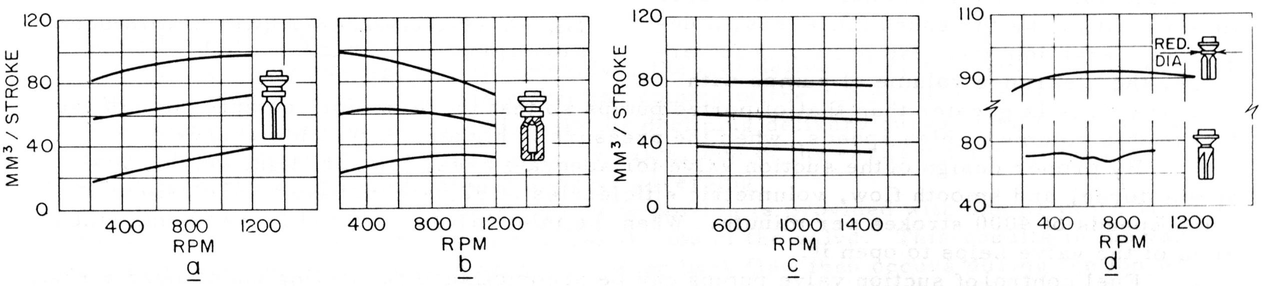

Fig. 67. Fuel delivery vs. speed characteristic of ported pumps (a, b, d,) and a suction valve pump (c) with zero retraction delivery valve.

The quantity of fuel delivered before port closing is termed preflow[1]. A second reason is that fuel delivery continues for a few degrees as after flow after port opening, because of the inertia of the fuel and gradual relief of pressure as the cut-off port is opened. Moreover, as the speed increases the injection pressure also increases so that the delivery valve closes more rapidly. This results in a greater quantity of fuel being compressed at high pressure, which then flows out of the nozzle. Figure 68 also shows that the injection pressure above the plunger continues for several degrees after port opening starts, indicating that afterflow is more pronounced than preflow.

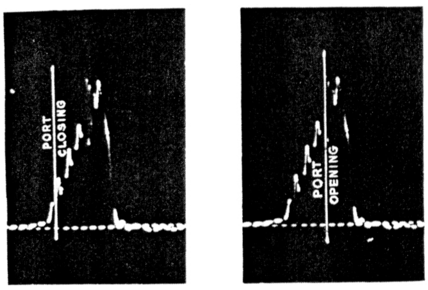

Fig. 68. Pressure oscillograms of a ported pump.

The degree to which the fuel delivery is affected by the preflow and afterflow depends on the size and shape of the ports, the rate of plunger displacement at port closing and port opening. delivery valve and nozzle characteristics such as opening pressure and effective flow area, and residual line pressures. Table 4 gives a tabulation of the various factors and their influence on the delivery characteristics. In general, the more abrupt the closing and opening of the ports, by making them rhomboidal shaped instead of circular, and the greater the flow resistance through the delivery valve, discharge tubing, and nozzle, the less inclining the delivery curve will be in the full load range. At reduced fuel quantities the resistance effect is less pronounced, so that at part load conditions it is more difficult to flatten out the delivery curve.

The type of nozzle has a pronounced effect on the delivery curve as shown in Fig. 69. The fuel delivery, especially at full load, is more inclining with the pintle nozzle because its valve lift controlled orifice area increases with the speed, whereas the orifice area of hole type nozzle is constant.

Valved Pumps

When the incoming fuel is controlled by a spring-loaded suction valve instead of a port, a moderately declining fuel delivery curve is obtained as shown in Fig. 67c (see preceding page). This declining characteristic is largely the result of inertia of the valve which causes an increasing lag in its opening and closing as the speed increases. Thus, some of the fuel drawn into the plunger barrel on the suction stroke is by-passed into the sump on the discharge stroke by late closing of the valve. With the shorter interval of time for filling as the speed increases, a degree of starvation may also occur. Furthermore, the clearance volume of pumps with suction valves is greater than that of ported pumps so that there is more compression of the fuel in this volume at high speeds, when the pressure is higher, than at low speeds.

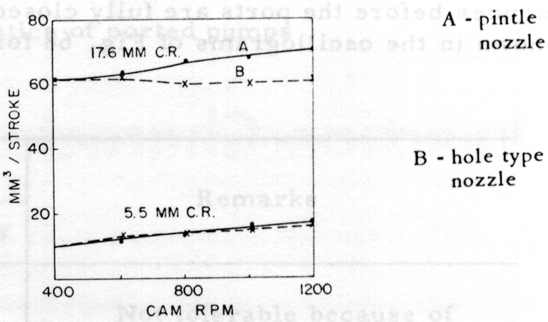

Fig. 69. Effect of nozzle on delivery characteristics of a ported pump.

By proper design of the suction valve for adequate area, low lift, light weight, low spring force, and smooth flow, volumetric efficiencies of 92 percent have been obtained at pump speeds of 4000 strokes per minute. When the inlet valve is located in the plunger the inertia of the valve helps to open it.

Fuel control of suction valve pumps can be accomplished by varying the plunger stroke, by use of a separate spill valve. or with a needle by-pass valve. The first two methods have little effect on the delivery characteristic so that it is still slightly declining. However. with the by-pass needle valve control a steeply inclining speed-delivery characteristic results, because the quantity of fuel by-passed increases as the speed is lowered.

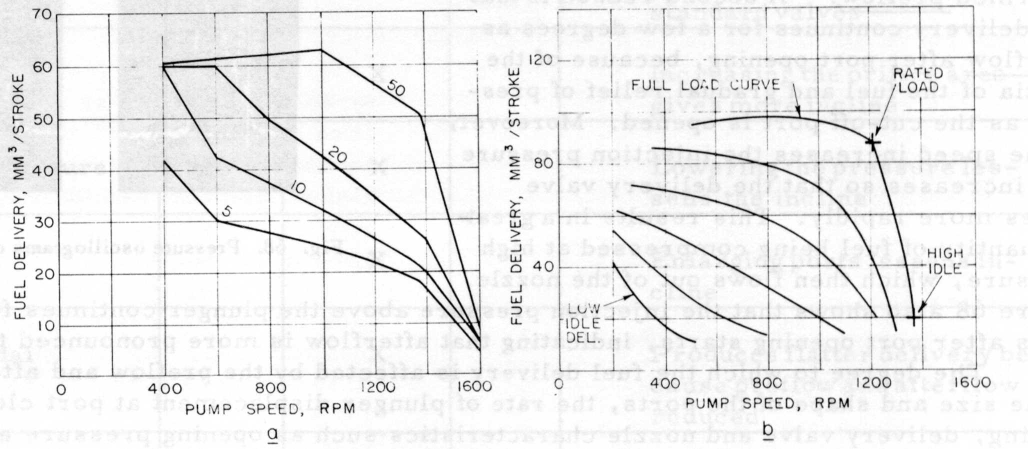

Fig. 70. Fuel delivery vs. speed characteristics.
(a) Throttle control pump at various supply pressures.
(b) Requirements for governorless operation.

Throttled Fuel Inlet

The simplest method of fuel control, and one which gives a declining speed delivery characteristic, is throttling of the fuel entering the plunger barrel. Typical speed-delivery curves of a pump of this type are shown in Fig. 70a.
A problem with this type of fuel control is limiting the maximum fuel delivery to prevent overloading as the speed is reduced. In the Roosa-Master pump this is simply accomplished by limiting the displacement of the pumping plungers.

The declining delivery characteristic inherent with throttling the inlet fuel makes speed governing easier; and where a throttle needle is used, the governor forces are low so that a small governor can be used. While some consideration has been given to eliminating governors with throttled inlet, acceptable speed control and response still necessitates a simple governor. Fig. 70b shows typical speed-delivery characteristics required for governor less engine operation in vehicle applications. The degree of speed regulation is determined by the slope of the delivery curves in the part load and full load over speed range.

IMPROVING DELIVERY CHARACTERISTICS WITH DELIVERY VALVES

As seen from Table4,the possible means for correcting the normally rising speed delivery curve of a ported pump to a flat or declining on are limited. Considerable improvement, however, has been made by various modifications of delivery valves with each engine application requiring a separate development.

Variable Retraction Volume

With a retracting type of delivery valve, the fuel delivered by the pump is a function of the effective plunger stroke less the retraction volume of the valve. If the retraction volume could be made to increase with speed, a flat or declining delivery curve should result. This is the basis for the adjustable relief delivery valve, one of the earliest types of valves introduced by Robert Bosch of Germany to improve governing.

In this valve the lapped cylindrical guide portion is provided with one or more tapered fuel passage lots decreasing in size towards the top of the valve. This results in the valve lifting higher as the pumps peed increases. More back flow then occurs during the longer time required for seating at higher lifts, and a throttling effect is produced by the restriction of the narrowing flutes so that more

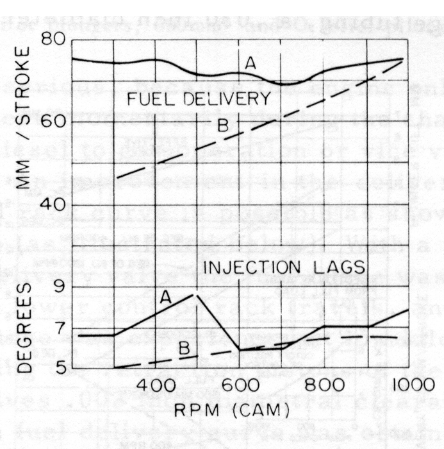

Fig. 71. Injection characteristics with adjustable (A) and standard (B) delivery valves and pintle nozzle.

fuel is retracted from the line. Thus, a nearly flat delivery is obtain ed as shown in the lower curve of Fig. 67d (see page 55) . Some variation of the delivery characteristic is also possible by changing the rate of the delivery valve spring or its initial compression.

A disadvantage of this type of valve is that the injection lag usually increases more with speed than for conventional valves, or it is not consistent (Fig. 71). This is because of its retraction being greater and occurring later in the injection cycle.

Variable Retraction Action

In a more recent development, variable retraction is obtained by providing measured clearance between the retraction piston and the bore in which it operates. This clearance may vary from .001to .003inches for small pumps to .007 inch for larger pumps, depending on the engine torque requirements. As shown in the upper curve of Fig. 67d a fairly flat delivery curve is obtainable with this simple modification of a standard retracting type delivery valve, and normal injection lags result.

The valve lift is practically the same over the entire speed range. Full retraction is obtained at the high speeds, but as the speed decreases the valve seats slower and less retraction takes place a s flow from the pump through the valve clearance and into the line continues until the valve is seated.

The effect of varying the diametral clearance of the retraction piston is shown by the lower family of curves in Fig. 72 (see following page). These show that considerable variation is possible in the delivery-speed characteristics . The distance between the corresponding upper and lower curves represent the effective retraction volumes at the different speeds for various valve clearances.

This valve modification has been developed primarily to obtain desired engine torque characteristics, especially where the governor does not incorporate a torque control. It is used, for example, in the American Bosch single plunger PSB pump for tractor applications where considerable torque increase with decreasing speed is required.

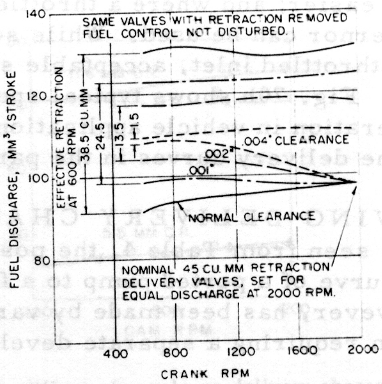

Fig. 72. Effective retraction action of delivery valves with various piston diametral clearances.

Valve Restrictions

Another method of obtaining a declining delivery is placing a restricting orifice in the delivery valve so that the throttling effect is not dependent on the valve lift. The restricting effect and pressure built up in the pump then increases rapidly with the speed, with a resultant decrease in fuel discharged through the nozzle.

In the German Henschel M. E. P. Control a restricted orifice was inserted in the pump end of the discharge tubing, and a spring loaded valve in the line would by-pass fuel whenever the pressure exceeded a predetermined value. The British Simms pump had the restricting orifice in the delivery valve, and it is claimed that this provided some drop in the residual line pressure before the valve seated.

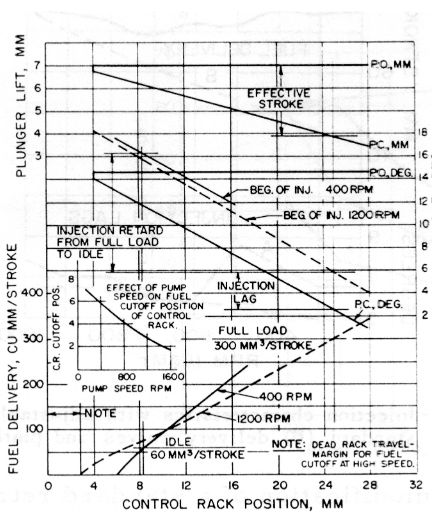

Fig. 73. Typical fuel delivery and timing characteristics of a ported pump. (13mm plunger; 6 hole 0.35mm nozzle.)

In an effort to eliminate the governor and provide satisfactory speed control by the speed-delivery characteristics of the injection system, an extensive program was conducted some years ago at the Pennsylvania State College. Their pump modifications of a small size, conventional multi-plunger pump consisted of a .030 inch restriction at the pump end of the discharge tubing, a .030 inch diameter hole in the delivery valve, and a shallow leakage groove .030 inch wide by .002 inch deep on the bottom lapped surface of the delivery valve body where it seals against the plunger barrel.

Typical speed-delivery characteristics of the effect of the delivery valve restriction alone are shown in Fig. 67b (see page 55). The delivery valve restriction was the primary factor for controlling the declin-ing characteristics. The restriction at the discharge tubing inlet, being in series with that in the delivery valve, was added to smooth out the delivery curves. The purpose of the leakage groove was to reduce the high pressures developed at the pump, but in so doing it tended to give an inclining rather than declining delivery because of greater leakage at low speeds. Considering the requirements illustrated in Fig. 70, it is evident that these modifications would not eliminate the need for a governor.

There are several disadvantages to controlling delivery characteristics by restrictions in or near the delivery valve:. First, a larger than normal plunger diameter is required for a specific engine application because of reduced capacity and increased duration of injection over the full load speed range; Second, higher than normal injection pressures act on the plunger with resultant increased loading of tappets, cams, and camshaft bearings; Third, injection lag increases with speed more rapidly than with a standard pump, so that for most applications an automatic timing advance mechanism would be required. For example, in one application the injection lag increased 3. 5 to 5 camshaft degrees over that obtained with a standard pump.

FUEL DELIVERY VS. CONTROL RACK TRAVEL

Coordination of the governor and injection pump for satisfactory governing requires that the relationship of fuel discharge vs. pump control rack or lever position be known. The fuel delivery curve at a fixed speed should be smooth and free from any aberrations, as shown by the lower curves for 400 to 1200 rpm of Fig. 73. The control rack travel from idle to full load is shown, as well as the dead rack travel from beginning of delivery to zero. The latte r is required to assure fuel shut-off. From the inset left of the center of Fig. 73, it is shown that with ported pumps fuel shut-off occurs at lower control rack positions as the pump speed is increased.

In addition to the fuel delivery curves, Fig. 73 shows the geometric relationships of port closing and port opening on both plunger lift and cam angle bases as well as the beginning of injection at engine idle and full load speeds.

Comparable data from a compound pump for a dual fuel engine is shown in Fig. 74. Fuel delivery by the pilot plunger for ignition of gaseous fuel is constant, but its timing can be varied 5.5 degrees. When operating on the main plunger, the beginning of injection is constant and the fuel quantity is varied according to the engine load. Also plotted are the residual pressures, which reach a maximum of 2800 psi in the pilot range since the pilot delivery valve has no retraction. Varying residual pressures in the transition range between operation on the pilot and main plungers is not serious, because the engine only operates there momentarily during the change from diesel to gas operation or vice versa.

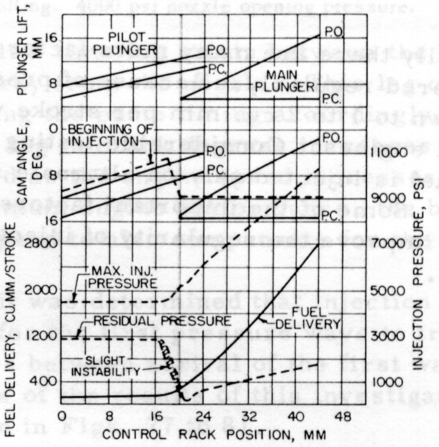

Fig. 74. Characteristics of compound pump. 22mm main and 10mm pilot plungers, 600mm³ and O relief pilot del. valves.

An improvement in the delivery vs. control rack curve is possible as shown by Fig. 75 (as illustrated below). With a standard delivery valve the fuel curve was irregular at lower control rack travels, and engine surge was experienced at low idle speeds. By giving the retraction pistons of the delivery valves .002 inch diametral clearance. a smooth fuel delivery curve was obtained and steady idling resulted.

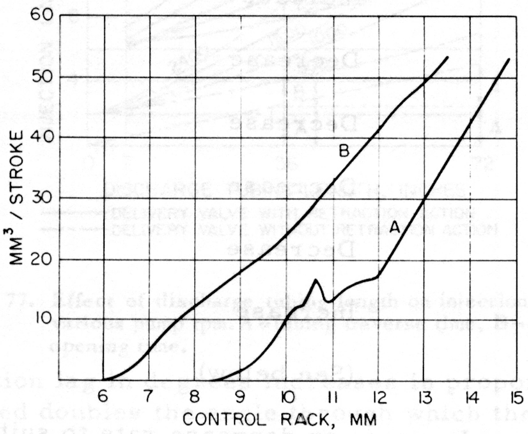

Fig. 75. Effect of delivery valve on slope of fuel curve at idling.

Minimum Regular Injection

With hydraulically operated nozzle valves there is a minimum quantity of fuel that can be delivered regularly on successive strokes by an injection pump. Theoretically this minimum regular delivery depends upon the nozzle opening and closing pressures (P_o and P_c), the trapped fuel volume (V) from the pump delivery valve to the nozzle valve seat, and the modulus of elasticity (K) of the fuel. This can be expressed as:

$$\text{Minimum regular delivery} = Q_m = \frac{V}{K}(P_o - P_c) \qquad (34)$$

For an inwardly opening hole type nozzle the relation between the opening and closing pressures is determined by the nozzle valve diameter (D) and the valve seat diameter (d) as follows:

$$P_c = P_o \frac{(D^2 - d^2)}{D^2} \qquad (35)$$

Then by substitution

$$Q_m = \frac{V}{K}\frac{d^2}{D} P_o \qquad (36)$$

Actually there are many more factors that influence the minimum quantity of fuel that can be delivered regularly. Because of pressure wave phenomena, it is possible to deliver regularly down to 1 to 2 cu. mm per stroke with certain combinations of injection equipment for small engines. Considerable testing is sometimes required to eliminate two stroking, where the fuel is injected only on alternate strokes.

Some of the important factors influencing the regularity of injection which can be alter-ed to improve the regularity of injection are tabulated below in their probable order of importance.

TABLE 5

Means for Reducing Minimum Regular Discharge

Item	Change Required
Cam velocity at part closing	Increase
Discharge tubing bore	Decrease
Trapped fuel volume	Decrease
Nozzle opening pressure	Decrease
Nozzle valve lift	Decrease
Plunger diameter	Increase
Delivery valve retraction volume	(See below)
Nozzle pressure adjusting spring	Increase or decrease rate to suit

The retraction volume of the delivery valve is generally selected to give a spray free of secondary injections at full load. At light loads or idling, the retraction may then be too great so that cyclic irregularity of injection occurs. This can often be corrected with variable retraction action valves giving only partial retraction at idle and light loads to permit a higher residual pressure, which results in regular injections at these conditions.

Irregular injections are, of course, objectionable since they cause engine roughness. They can be detected by feeling uneven pulsations in the discharge tubing, observing the beginning of the spray with a stroboscope, or by recording the instantaneous injection pressures in the discharge tubing. Fig. 76 illustrates a typical pressure diagram of injection irregularity. On the first cycle the injection pressure builds up from zero residual pressure to almost the nozzle opening pressure. No injection occurs, but the residual pressure becomes 1725 psi. On the next stroke injection occurs as the pressure builds up to 6000 psi and the residual pressure then drops to zero.

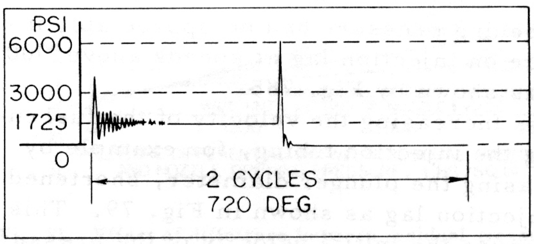

Fig. 76. Pressure oscillograms of cyclic injection irregularity at idling. 4000 psi nozzle opening pressure.

INJECTION LAG

In the chapter on pressure waves it was shown that a time lag exists between the beginning of fuel delivery from the pump and the start of spray from the nozzle. This lag can be readily accounted for in constant speed engines by adjusting the pump timing accordingly, but in variable speed engines it becomes a problem since the lag in crank degrees increases with speed. This results in the injection timing being retarded as the engine speed increases , unless corrected by a timing advance mechanism. For optimum engine performance, the beginning of injection should be advanced with an increase in speed to also compensate for the ignition lag.

In a comprehensive investigation[3] on this subject it was determined that injection lag is made up of two distinct periods: (1) The time required for the first pressure wave to travel from the pump to the nozzle; and (2) the time that elapses between arrival of the first wave at the nozzle and the lifting of its valve off its seat. Some of the results of this investigation with a conventional injection system (R. Bosch) are shown in Figs. 77 to 81.

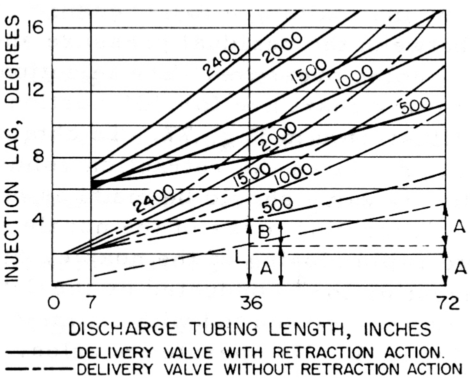

Fig. 77. Effect of discharge tubing length on injection lag at various pump rpm. A = tubing traverse time, B = nozzle opening time.

Fig. 77 shows the effects of tubing length, pump speed, and delivery valve retraction on injection lag. Since the pressure wave travels through the tubing with the velocity of sound in the fuel, these curves show that the injection lag increases with the tubing length. The time calculated for the first pressure wave to reach the nozzle, as indicated by the lower, inclined, dashed line for the conditions of 500 rpm and zero retraction valve, increases directly with the length of curve and the measured injection lag represents the time required for the pressure build up at the valve to open it. For a given speed, lag "B" is practically constant regardless of the tubing length.

As the pump speed increases , the injection lag in degrees increases in proportion, since for equal time intervals doubling the speed doubles the angle through which the pump camshaft rotates. The distances between curves are not equal, as the residual pressures in the discharge tubings vary somewhat at different speeds. With the short 7 inch tubing and zero retraction valve it is significant that the injection lag increased only 1 degree

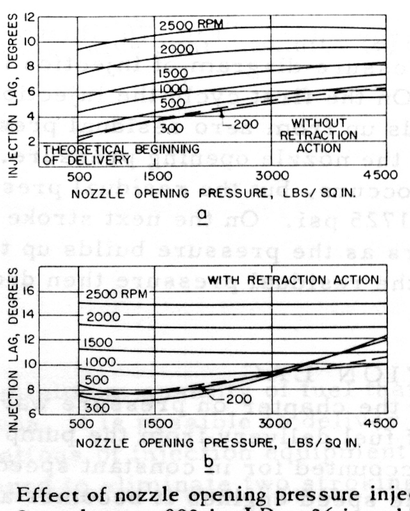

Fig. 78. Effect of nozzle opening pressure injection lag. 8mm plunger, .093 in. I.D. × 36 in. tubing.

over the speed range from 500 to 2400 rpm, whereas with a 36 inch tubing the increase in lag was 6 degrees. This emphasizes one of the advantages of unit injectors.

The injection lag increases with retracting type delivery valves because of lower residual pressures, which also lower the accoustic velocity of the pressure waves, particularly if the residual pressure drops below atmospheric.

Fig. 78a (see following page) shows that the injection lag increases as the nozzle opening pressure is raised, but the effect is less pronounced as the pump speed is increased.
With a retracting type delivery valve the nozzle opening pressure had no appreciable influence on injection lag at speeds above 1000 rpm as shown by Fig. 78b.

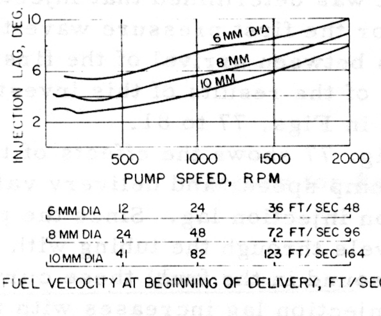

Fig. 79. Effect of fuel velocity on injection lag. O retraction valve, .093 in. I.D. × 36 in. tubing.

Increasing the velocity of the fuel entering the injection tubing, for example by increasing the plunger diameter, shortened the injection lag as shown in Fig. 79. This is because the initial velocity of the pressure wave is the sum of the accoustic velocity and the initial fuel velocity. The residual pressure also generally increases as the initial fuel velocity is increased with resultant higher injection pressure.

Fig. 80 shows that the tubing bore had relatively little effect on the injection lag for bores of 3/32 inch and above. The longer injection lag with 1/16 inch bore tubing at pump speeds above 500 rpm can be attributed to increased flow resistance and residual pressures of 15 to 30 psi, as compared with 450 to 600 psi for the 1/8 inch bore tubing. It was also discovered that the injection lag was not measurably affected by roughness of the tubing bore, nor by sharp bends or corners in the tubing or fuel passages.

It is significant from Fig. 81 that the injection lag can be minimized by maintaining a low pressure differential between the residual pressure and the nozzle opening pressure. The injection lags shown in this chart are for "B" portion of the lag as illustrated in Fig. 77. To obtain high residual pressures the retraction volume of the delivery valve should be made as small as possible without getting secondary injections, and the reciprocating mass of the nozzle and its holder should be reduced as much as possible to promote rapid closing of the nozzle. Similar results to those already reported were obtained in an investigation[4] conducted by the National Advisory Committee for Aeronautics, although their tests were conducted with a timing valve controlled constant-pressure injection system utilizing hydraulically operated nozzles. One of their principal findings, as shown in Fig. 82 (see following page), was that the injection lag decreased as the injection pressure was increased. They also pointed out that discharge tubing for a multi-cylinder engine should all be of equal length to assure uniform injection timing to each cylinder.

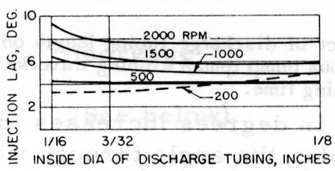

Fig. 80. Effect of discharge tubing bore on injection lag. O retraction valve, 36 in. tubing.

DURATION OF INJECTION

The most important injection characteristic is the spray duration, particularly at full load, since it directly affects the engine power, fuel consumption, and exhaust smoke. Injection pressures and spray characteristics, such as penetration and dispersion, can be compensated for to some extent by combustion chamber design and air turbulence, but for efficient combustion the duration of injection must be closely controlled. Each type of combustion chamber and engine has, in fact, an optimum duration of injection.
In applying an injection system to an engine, considerable attention is given to obtain a suitable spray duration free from secondary injections. If the duration is made too short good fuel economy and smoke are obtained, but the engine runs rough. When the duration is too long, power, fuel consumption, and smoke are poor although the engine may run smoothly. In Table 6 are listed various factors giving their effect on the injection duration, and Table 7 gives a sample set of test data obtained in selecting injection equipment for a locomotive diesel engine.

TABLE 6

Factors affecting duration of injection

Test No.	Plunger Diam. mm.	Del. Valve Retract. cu. mm.	Nozzle Hole Size mm.	Tubing I.D. in.	Peak Inj. Press.-psi Pump	Holder	Duration of Injection Crank Degrees Total	Main	Secondary
1.	18	300	9-0.30	.095	15300	14250	63	41 1/2	2
4.	"	"	9-0.325	.095	13100	11250	37 1/2	37 1/2	2 1/2
8.	"	200	9-0.325	.095	13100	11000	36 1/2	36 1/2	0
9.	"	"	"	.107	12250	10900	36	36	0
13.	19	"	"	"	12900	12000	33	33	0
17.	"	300	"	"	12700	11700	35	35	0
19.	"	"	10-0.325	"	11750	11400	35	35	0

Fig. 83 (see following page) shows the increase with speed of the spray durations and injection pressures with the injection equipment used on an automotive diesel engine. The secondary injection shown was of light intensity, and it could have been eliminated by increasing the retraction volume of the delivery valve.

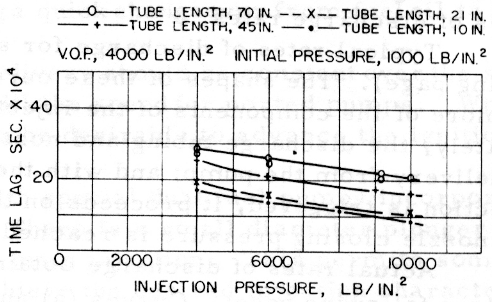

Fig. 82. Effect of injection pressure of common-rail system on injection lag.

RATE OF DISCHARGE

The rate at which the fuel is injected into the combustion chamber is important, since it controls the rate of pressure rise, particularly for open chamber engines, after burning has started. As discussed in the Chapter on Combustion, the fuel has an ignition lag of several crank degrees before it starts to burn, and so the rate of injection during the period should be limited to prevent "diesel knock. " The high temperature developed in the combustion chamber after the ignition delay period enables the balance of the metered fuel to burn more uniformly at a controlled rate.

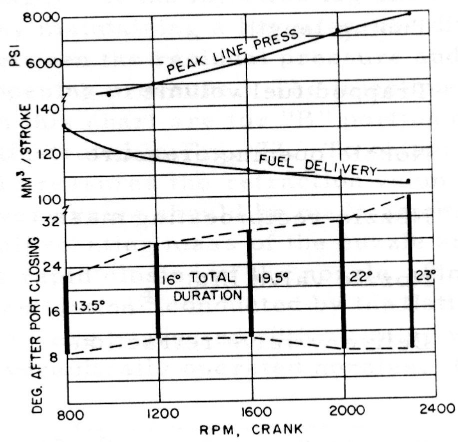

Fig. 83. Typical spray durations and pressures for an open-chamber automotive diesel engine.

Several investigators have found that by maintaining a low initial rate of injection, the quantity of fuel available for combustion when ignition occurs is so small that knock cannot be created. Others have suggested that if the fuel is injected rapidly, it will only burn as fast as air is made available. Combustion can be controlled by air turbulence and combustion chamber design, so that successful engines have been built with various types of discharge rates. Typical rates of discharge for several injection systems are shown in Fig. 84 (see following page). The shapes of these curves can be modified to some extent by a change in one or more of the components of the injection system as shown in Table 8. Unfortunately, the discharge tubing and nozzle restriction with a jerk pump system alters the rate of delivery from the pump; and with the common-rail and accumulator nozzle systems once the injection is triggered, it proceeds on the basis of fuel expansion from the accumulator until the nozzle closing pressure is reached. Actual rates of discharge obtained with various injection equipment are shown in Fig.85 (see following page). Curves (a) and (b) show the injection rates for a locomotive open chamber engine with jerk pumps of the ported and spill valve types respectively. The cut-off appears faster with the latter type pump because of the larger by-pass area of the spill valve.

TABLE 7

Injection equipment data and injection characteristics

Nozzle opening press. 3500 psi Disch. tubing 27" long
Fuel quantity: 1140 cu, mm/ stroke at 525 pump rpm

Test No.	Plunger Diam. mm.	Del. Valve Retract. cu. mm.	Nozzle Hole Size mm.	Tubing I.D. in.	Peak Inj. Press.-psi Pump	Holder	Duration of Injection Crank Degrees Total	Main	Secondary
1.	18	300	9-0.30	.095	15300	14250	63	41 1/2	2
4.	"	"	9-0.325	.095	13100	11250	37 1/2	37 1/2	2 1/2
8.	"	200	9-0.325	.095	13100	11000	36 1/2	36 1/2	0
9.	"	"	"	.107	12250	10900	36	36	0
13.	19	"	"	"	12900	12000	33	33	0
17.	"	300	"	"	12700	11700	35	35	0
19.	"	"	10-0.325	"	11750	11400	35	35	0

Curves (c) and (d) for an engine with a Lanova combustion chamber show the rates of discharge obtained with jerk pumps of the variable stroke and ported type, respectively. In the former the fuel quantity was varied by lifting the tappet off the cam and injection continued to the end of the cam lift. The beginning of injection was quite rapid but the ending was slow, and it extended beyond the end of lift for a prolonged injection with considerable dribble. The shorter injection of the ported pump gave better engine performance.

Rates of discharge with a unit injector for a two-cycle open-chamber automotive engine are shown by curves (e) 5. By eliminating the discharge tubing the fluctuations in rates are eliminated at high speeds, and the rates of injection closely follow the rate of plunger displacement as determined by the plunger diameter and cam.

The rates of injection with a ported pump for a four cycle, open-chamber automotive engine are shown by curves (f). The beginning and ending of the injections are fairly steep to assure a well atomized spray free from secondary injections and pronounced dribble. These curves also illustrate the increase in rate of discharge per cam degree and the shortening of duration which occurs as the pump speed is decreased.

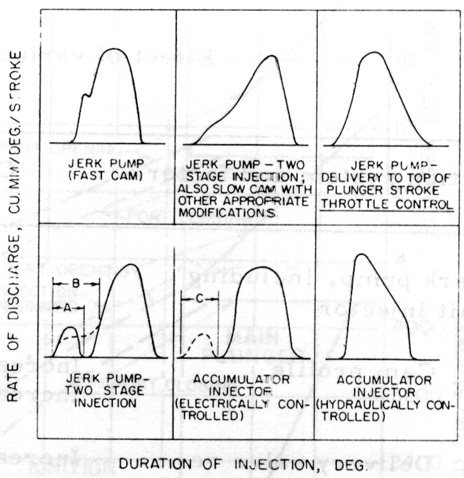

Fig. 84. Typical rates of discharge.

TABLE 8

Effect of various factors on rate of discharge

Injection system and factor	Change	Effect on rate
A. Jerk pump, including unit injector		
1. Cam profile	Increase velocity at port closing Increase velocity at port opening	Higher initial rate Sharper termination
2. Delivery valve retraction	Increase volume	Little, but less control
3. Residual pressure	Increase uniformity	Little, but more reproducible
4. Tubing bore and length	Decrease	Rate approaches that of pump
5. Nozzle opening pressure	Lower	Beginning and ending slower
6. Port opening (throttle inlet)	Inject to end of stroke	Slow termination
B. Jerk pump, two stage (mechanical)		
1. Cam profile	Slowly increasing velocity	Low initial rate
2. Residual pressure	Maintain constant over speed range	Required for low initial rate
C. Accumulator System (hydraulic)		
1. Nozzle opening pressure	Decrease	Little, if any
2. Nozzle closing pressure	Decrease	Slow termination
3. Accumulator volume	Decrease	High initial rate, shorter duration
4. Rate of drop of triggering pressure	Decrease	Can produce pre-injection, but speed sensitive
5. Engine speed	Decrease	Rate and discharge increase
D. Accumulator injector (electrical)	Various	Wide range possible by changing pressure level and electrical timing of duration

INJECTION CHARACTERISTS OF DUAL FUEL SYSTEM

The injection characteristics required for dual fuel operation are very demanding. First, suitable spray durations and injection pressures are required throughout the speed and load range for satisfactory engine operation on fuel oil alone. Second, when operating on gas the ignition fuel quantity is only 5 to 10 percent of the full load quantity, and this small fuel quantity must be regularly metered and atomized through the same nozzle as used for normal diesel operation. The ignition fuel quantity should be less than the regular diesel idle quantity to prevent engine over-speed when making a quick change over from fuel oil to gas operation, and to reduce the cost of the ignition fuel.

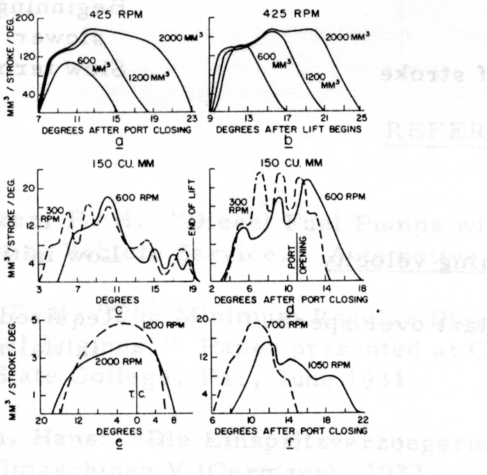

Fig. 85. Measured rates of discharge with various injection equipment.

For variable speed operation the ignition fuel delivery should be constant over the speed range, whereas normally the full delivery increases with the speed for ported pumps. When operating with gas at various engine loads, it may also be desirable to advance the ignition fuel timing as the load decreases.

These exacting requirements are obtainable with special dual fuel pumps incorporating two plungers with separate delivery valves for each. There is a small diameter plunger for the ignition fuel and a larger diameter plunger for the main injection. This permits some selection of plunger diameters and delivery valves to achieve the desired injection characteristics for both full load and ignition fuel quantities. This is complicated somewhat by using the same nozzle and discharge tubing for both conditions. This results in the ignition fuel spray duration being very short, perhaps 4 to 6 crank degrees.

Fig. 86 shows typical fuel delivery characteristic, injection timing, and geometric timing of a telescopic dual fuel pump.

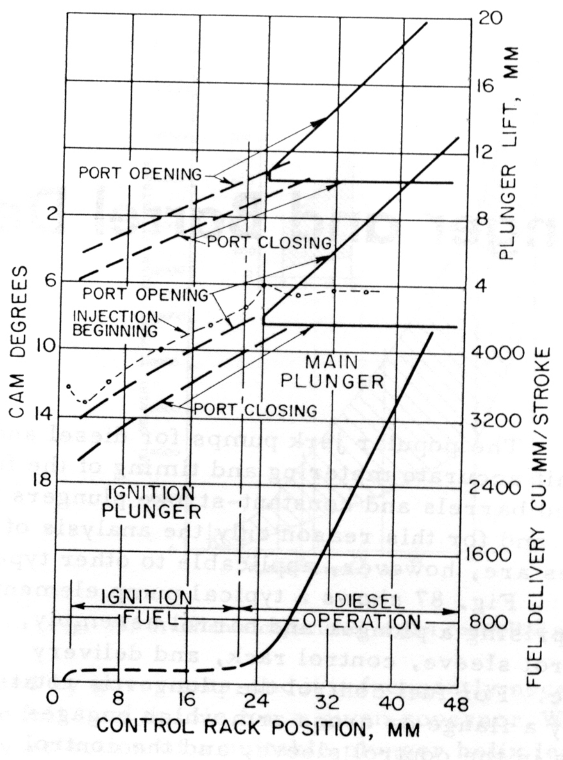

Fig. 86. Geometric and injection characteristic of a telescopic dual fuel pump

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