960

~~3205~~
~~189~~
~~Copy~~ 1

TECHNICAL MEMORANDUMS

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

No. 993

DIRECT INJECTION IN INTERNAL-COMBUSTION ENGINES

By Jean E. Tuscher

Publications Scientifiques et Techniques
du Ministère de l'Air, No. 89, 1939

Washington
November 1941

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

TECHNICAL MEMORANDUM NO. 993

RESEARCHES ON DIRECT INJECTION IN INTERNAL COMBUSTION ENGINES*

By Jean E. Tuscher

Experiments carried out on two different types of engines show that the laws relating injection and combustion and, finally, the operating qualities of the engine itself, are general, and are influenced neither by the shape of the combustion chamber nor by a more or less strong turbulence maintained in this chamber.

The study of retarded cycles, during which combustion takes place entirely during the expansion stroke, has provided a means for reducing the combustion speed and for increasing the concentration of the air-fuel ratio up to saturation while maintaining a high thermal efficiency.

The combination of a short injection period and retarded cycle will produce the greatest specific power from a Diesel engine, while reducing, at the same time, the fatigue of its parts to within acceptable limits.

To sum up, these researches have adduced a solution for reducing the fatigue of the Diesel engine by permitting the preservation of its components and, at the same time, raising its specific horsepower to a par with that of carburetor engines, while maintaining for the Diesel engine its prerogative of burning heavy fuel under optimum economical conditions.

The feeding of Diesel engines by injection pumps actuated by engine compression, achieves the required high speeds of injection readily and permits rigorous control of the combustible charge introduced into each cylinder and of the peak pressure in the resultant cycle.

The suppression of the mechanical control of the pumps and the pressure lines simplifies the construction of direct-injection engines and improves their dependability in service.

*"Recherches sur l'injection directe dans les moteurs a combustion." Publications Scientifiques et Techniques du Ministère de l'Air, No. 89, 1939.

Further industrial improvements in the injector-pump units used in these tests should raise no insuperable difficulties, although the 50-hour endurance test may not be sufficient to give adequate assurance of perfect behavior of the mechanism under a 20- or 50-times longer duration, as is habitual in industrial practice.

Nevertheless, the stability in operation, the total absence of wear during these laboratory tests, together with the simplicity of the device permit us to look for no insurmountable difficulties.

PART I

Chapter 1

FEED SYSTEM CONTROLLED BY ENGINE COMPRESSION

The Winterthur Engine

1. Test stand.- The experiments recounted below were made on a test stand of the Research Laboratories of the Air Ministry, involving a single-cylinder, two-stroke-cycle Diesel engine (bore, 125 mm; stroke, 170 mm) with direct injection, developing 8 horsepower at 800 rpm, and 10 horsepower when supercharged.

This engine, built by the "Société Suisse pour la construction de Locomotives et de Machines à Winterthur," is normally operated by a Bosch pump and injector.

The exhaust ports are overlapped by the piston, the piston displacement being 1500 liters, the volumetric ratio of compression, 12. The compression of the scavenging air is effected in the engine crankcase.

The engine is coupled to a dynamometer whose load is regulated by a table of resistances. The arm of the balance is 716 millimeters long. Revolution counters and totalizers, a 100-cubic-centimeter sample for measuring the consumption, and a device for recording the diagrams with Lohmann and Michels' indicator, complete the equipment.

2. Experimental procedure.- The Bosch equipment was replaced by a device in which pump and injector form one

mechanical unit actuated by the compression of the engine.

A feed line leads the fuel without compression in the device, which is mounted on the cylinder head in the central seat usually reserved for the injector.

This solution obviates the mechanical-pump control and the fuel-pressure lines, and is applicable to two-stroke-cycle and four-stroke-cycle engines alike. For the combustion cycle following the injection, it offers the advantage of a duration of injection independent of the engine speed.

The experiments comprised the recording of the power and consumption curves of the engine for different pressure and duration of injection.

The results and conclusions discussed hereinafter were arrived at after experimentation with more than 100 feed systems, all actuated by engine compression. The principal characteristics of these various devices are described in the patents filed by the author.

3. Description of device.- Figure 1 is an external view of the injector-pump assembly tried out in the Air Ministry; it is shown attached to the cylinder head of the Winterthur engine.

The mechanism is shown in figure 2. A support 1, projecting from the gas cylinder 2, is clamped to the cylinder head in place of the conventional injector. A sleeve 7, sliding over a fixed plunger 8 of section S_3, forms the pump chamber 9. This chamber 9, empties into that of the working chamber of the engine through the atomization nozzles 6, of total section S_7 procured in an injector 10, carried on a shoulder of pump chamber 9.

The atomization orifices 6, are controlled by a differential needle 11 of sections S_4 and S_5, with limited positive opening. A spring R returns needle 11 to its seat. (See the enlarged sketch, fig. 3.)

The admission of the combustible in the pump chamber is insured by automatic valve 12, guided in the hollow 13 of plunger 8, through which the combustible arrives.

The sleeve 7, is externally engaged in the piston 14, of section S_2 which, through a conical port, obdurates

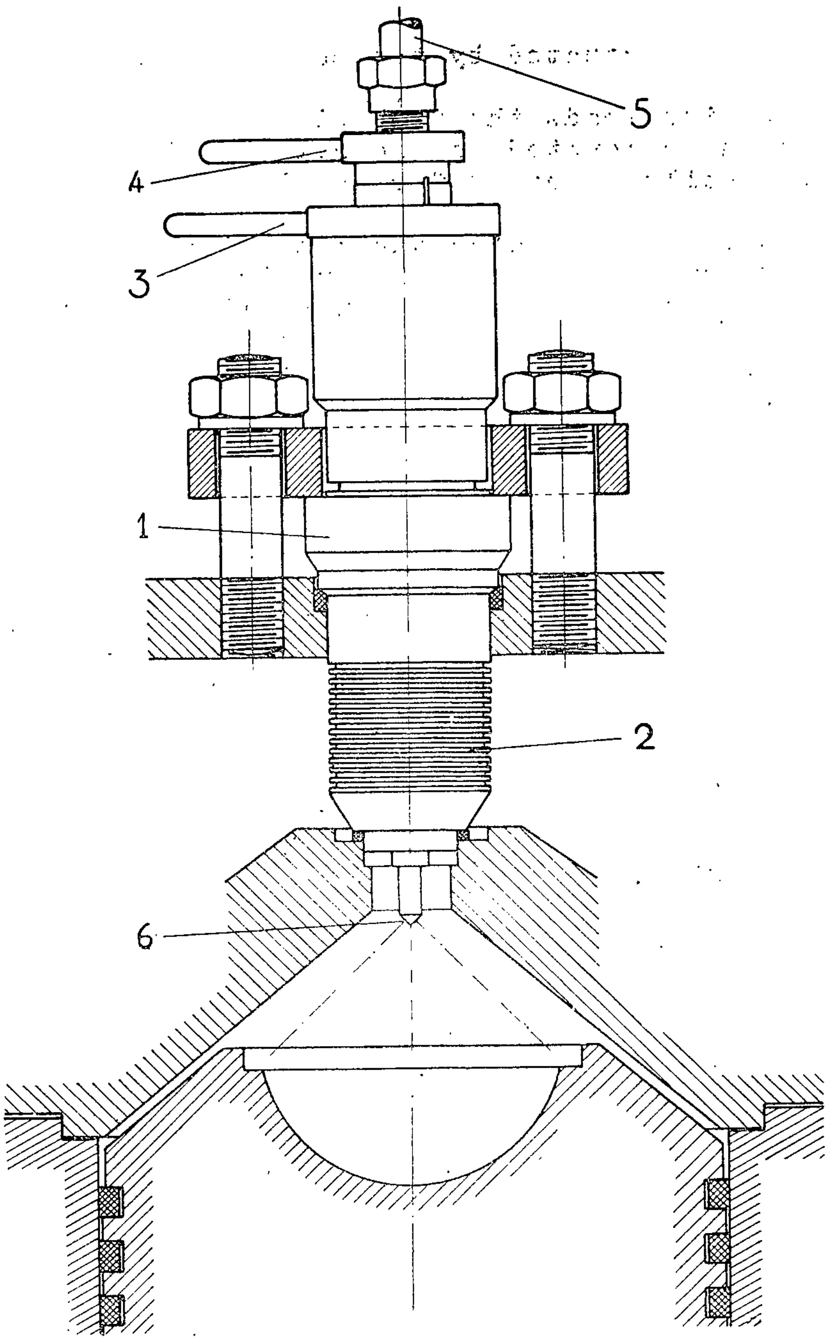


Figure 1.- Tuscher pump mounted on cylinder head of Winterthur engine.
1 support, 2 gas cylinder, 3 feed control, 4 injection point control, 5 feed line, 6 nozzle

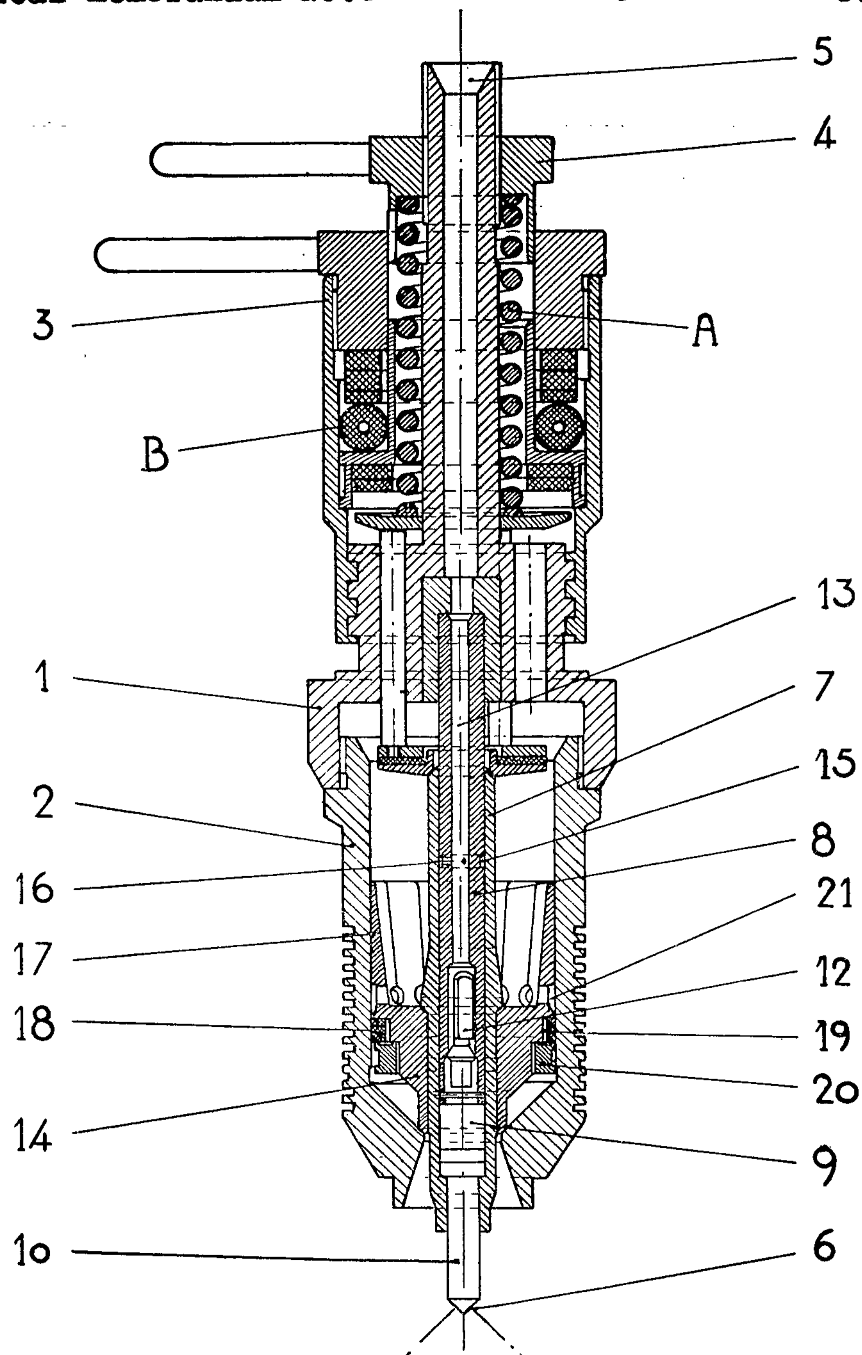


Figure 2.- Tuscher pump-injector unit.
1 body, 2 gas cylinder, 3 feed control, 4 injection point control, 5 feed line, 6 nozzles, 7 pump sleeve, 8 plunger, 9 pump chamber, 10 injector, 12 inlet valve, 13 feed, 14 gas piston, 15 and 16 labyrinth and holes for returning leakages toward feed, 17 elastic slippers, 18 rings, 19 spring, 20 labyrinth screw, 21 oil groove, A, return spring for injection point, B, shock absorber.

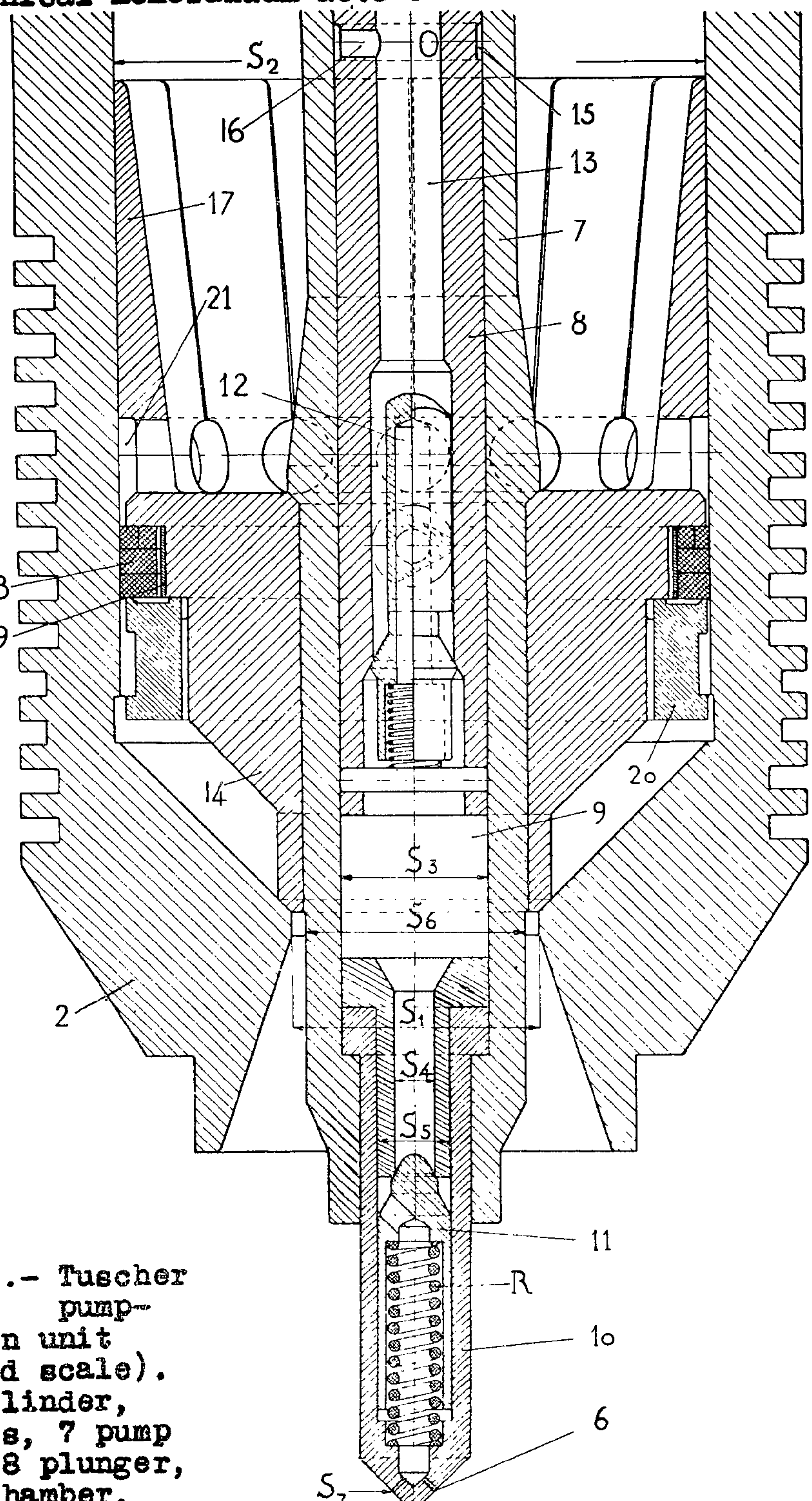


Figure 3.- Tuscher pump-injection unit (enlarged scale). 2 gas cylinder, 6 nozzles, 7 pump sleeve, 8 plunger, 9 pump chamber, 10 injector sleeve, 11 needle, 12 inlet valve, 13 feed, 14 gas piston, 15 and 16 labyrinth and holes for returning leakages toward feed, 17 elastic slippers, 18 rings, 19 spring, 20 labyrinth screw, 21 oil groove, R, needle spring.

the exit of section S_1 of gas cylinder 2, toward the working chamber of the engine.

The stroke of pump 7 and, consequently, of piston 14, is elastically limited in a damper B by an adjustable stop 3, screwing on support 1. The return of piston 14 to its seat S_1 in gas cylinder 2, is assued by spring A, whose initial force is regulated by screw 4 on support 1.

The dimensions of the device are those of the ordinary injector. Its total weight is 700 grams, and its moving part weighs 50 grams.

4. <u>Calculation of the operation</u>.- When the compression p_c on section S_1 becomes greater than the return spring A

$$p_c S_1 \geq A$$

piston 14 is raised from its seat S_1 and the pressure of the compression becomes:

$$p_c S_2 = P$$

The combustible admitted into the chamber of pump 9 is then compressed at a pressure p_i:

$$p_i = \frac{p_c (S_2 - S_1)}{S_3} = \frac{P - A}{S_3} \qquad (1)$$

To this pressure corresponds a well-defined rate of flow v_i through the section S_7 of the atomization nozzles 6. This rate is given by Torricelli's law:

$$v_i = \mu \sqrt{2gh}$$

wherein

v_i is speed of flow in m/s, through section S_7

μ coefficient of flow of the nozzles; 0.8 for the pipes, the diameter of which is one-fourth of the length

g acceleration of gravity, 9.81 m/s²

h flow pressure in meters, of height of the water

With δ as the density of gas oil, $\delta = 0.85$:

$$p_i = \frac{h\ \delta}{10} \quad \text{hence} \quad h = \frac{10\ p_i}{\delta}$$

we have:

$$v_i^2 = \frac{\mu^2\ 2g\ 10\ p_i}{\delta} = \frac{0.64 \times 19.62 \times 10\ p_i}{0.85} = 148\ p_t$$

$$V_i = 12.2 \sqrt{p_i} \simeq 12 \sqrt{p_i} \qquad (2)$$

and, as the injection takes place in air compressed at pressure p_c:

$$v_i = 12 \sqrt{p_i - p_c}$$

To this speed of flow v_i at pressure p_i, there also corresponds a well-defined speed v_p of the sleeve pump 7, and of piston 14, which actuates it.

The law of continuity enables us to write:

$$v_p = \frac{S_7}{S_3}\ v_i$$

In reality, p_i, v_i, and v_p have no constant value during the injection, since the pump leaves from a position of rest to return, after a stroke c, at speed equal to zero. On the other hand, as will be seen later on, the compression p_c may be considered as constant when the injection is achieved at either side of top center and during a period not exceeding that of the ignition lag.

The equation of motion is, in reality, if m is the mass of the moving part of the device :

$$m\ \frac{d^2x}{dt^2} = p_c\ (S_2 - S_1) - p_i\ S_3$$

wherein x is the path traversed.

The speed of the pump is, as seen:

$$v_p = \frac{dx}{dt} = \frac{S_7}{S_3} v_i = \frac{S_7}{S_3} 12 \sqrt{p_i}$$

By replacing this value of p_i thus expressed, in equation (4) gives:

$$m \frac{d^2x}{dt^2} = p_c (S_2 - S_1) - \left(\frac{dx}{dt}\right)^2 \frac{S_3}{\left(12 \frac{S_7}{S_3}\right)^2}$$

or, differently expressed:

$$m \frac{dv_p}{dt} = p_c (S_2 - S_1) - v_p{}^2 \frac{S_3}{\left(12 \frac{S_7}{S_3}\right)^2} \qquad (5)$$

Without integrating, it is seen that, if the acceleration $\frac{dv_p}{dt}$ becomes zero, the speed limit $v_{p\infty}$ of the pump, actuated under constant compression p_c, becomes:

$$v_{p\infty} = \left(12 \frac{S_7}{S_3}\right) \sqrt{\frac{p_c (S_2 - S_1)}{S_3}}$$

This speed limit is fairly conformable to equations (1), (2), and (3).

The differential equation (5) may be written

$$\frac{m\, dv_p}{p_c(S_2-S_1) - v_p{}^2 \frac{S_3}{\left(12 \frac{S_7}{S_3}\right)^2}} = dt \quad \text{or} \quad \frac{dv_p \left[\frac{m}{S_3} \left(12 \frac{S_7}{S_3}\right)^2\right]}{\frac{p_c(S_2-S_1)}{S_3} \left(12 \frac{S_7}{S_3}\right)^2 - v_p{}^2} = dt \qquad (6)$$

By posing:

$$\frac{m}{S_3} \left(12 \frac{S_7}{S_3}\right)^2 = \frac{1}{C_2} \quad \text{and} \quad \left(12 \frac{S_7}{S_3}\right) \sqrt{\frac{p_c (S_2 - S_1)}{S_3}} = \alpha$$

equation (6) becomes:

$$\frac{dv_p}{\alpha^2 - v_p{}^2} = C_2\, dt \quad \text{or} \quad \frac{dv_p}{v_p{}^2 - \alpha^2} = - C_2\, dt \qquad (7)$$

but

$$\frac{1}{v_p{}^2 - \alpha^2} = \frac{A}{v_p - \alpha} + \frac{B}{v_p + \alpha}$$

if $Av_p + A_a + Bv_p - B_a = 1$; that is, to say, $A = - B$ and $A = \frac{1}{2\alpha}$ (7) becomes:

$$\frac{dv_p}{v_p{}^2 - \alpha^2} = \frac{\frac{1}{2\alpha}\, dv_p}{v_p - \alpha} - \frac{\frac{1}{2\alpha}\, dv_p}{v_p + \alpha} = - C_2 \; dt$$

by integrating:

$$\log (v_p - \alpha) - \log (v_p + \alpha) = - C_2 \; 2 \; \alpha t + \log Cte$$

Thus, the general solution of the differential equation of motion is:

$$\frac{v_p - \alpha}{v_p + \alpha} = C_1 \; e^{-2\alpha C_2 t} \qquad (8)$$

where the constant C_1 has a value defined by the limit conditions for

$$t = 0 \; \ldots\ldots \; v_p = 0, \text{ hence } C_1 = - 1$$

and, as seen for

$$t = \infty \quad v_p = \alpha = \left(12 \; \frac{S_7}{S_3}\right) \sqrt{\frac{p_c \; (S_2 - S_1)}{S_3}}$$

and

$$C_2 = \frac{S_3}{m \left(12 \; \frac{S_7}{S_3}\right)^2}$$

The equation of motion of the pump is:

$$\frac{v_p - \alpha}{v_p + \alpha} = - \; e^{-C_2 2\alpha t}$$

This curve of the speed is of the following form:

It is important to know if, for a time interval $t = t'$

corresponding to the duration of injection, we are in region A or B. If in region B, the difference between v_p real and $v_{p\infty}$ is very small and the motion of the pump can, without appreciable error, be considered as being produced at uniform limit speed $v_{p\infty}$.

Now, the exponential e^{-x} decreases very rapidly with x, and the calculation of a particular case shows that the pump speed attains a value near $v_{p\infty}$ in the very small fraction of a millisecond.

Without major error, the motion may be conceded as being produced at speed $v_{p\infty}$.

The exponent C_2 shows that the limit is reached so much quicker as the moving mass m is smaller, as the section S_3 of the plunger is larger, and the section of nozzles S_7 is smaller. In fact, C_2 varies as $S_3{}^3/S_7{}^2$.

5. Needle valve spring.- The initial spring force R of the differential needle 11 plays no part in the solution of the speed and pressure of injection. To check experimentally that this spring R has no effect on the injection, use of the thin-walled sleeve pumps 7, is sufficient. In fact, they burst as easily with a spring R, permitting the opening of the needle at an initial pressure of 50 atmospheres, as with a spring determining a pressure p_i, starting at 500 atmospheres. It is essentially the same for the pumps with mechanical control feeding injectors with differential needle.

In these arrangements, the profile of the cams actuating the pumps define the speed of the plungers in relation to that of the engine. Thus, the speed and pressure of injection vary according to the engine speed, with the result that section S_7 of the atomizing nozzles is adapted for only one well-defined speed of rotation. And, it was experimentally established that there is one initial force R of the spring which is best suited to the running of the engine at a specified speed and load.

This force R, is accurately determined by the injection pressure p_i, defined above by equations (2) and (3):

$$p_i = \left(\frac{S_3 \ v_p}{12 \times S_7}\right)^2 \quad \text{and} \quad R = p_i \ (S_4 - S_5)$$

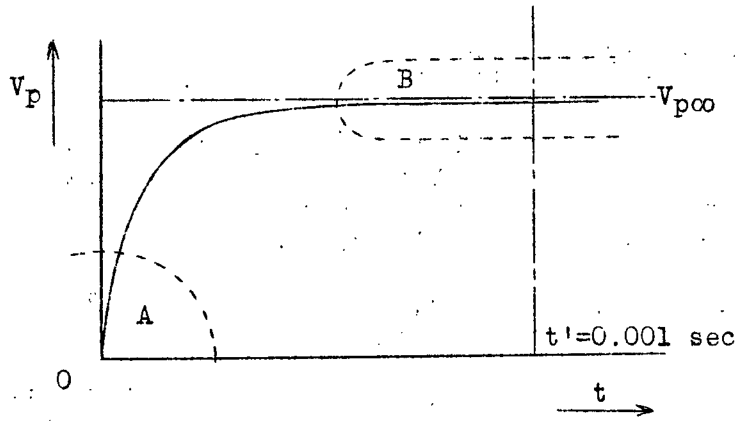


Figure 4.- Speed of pump.

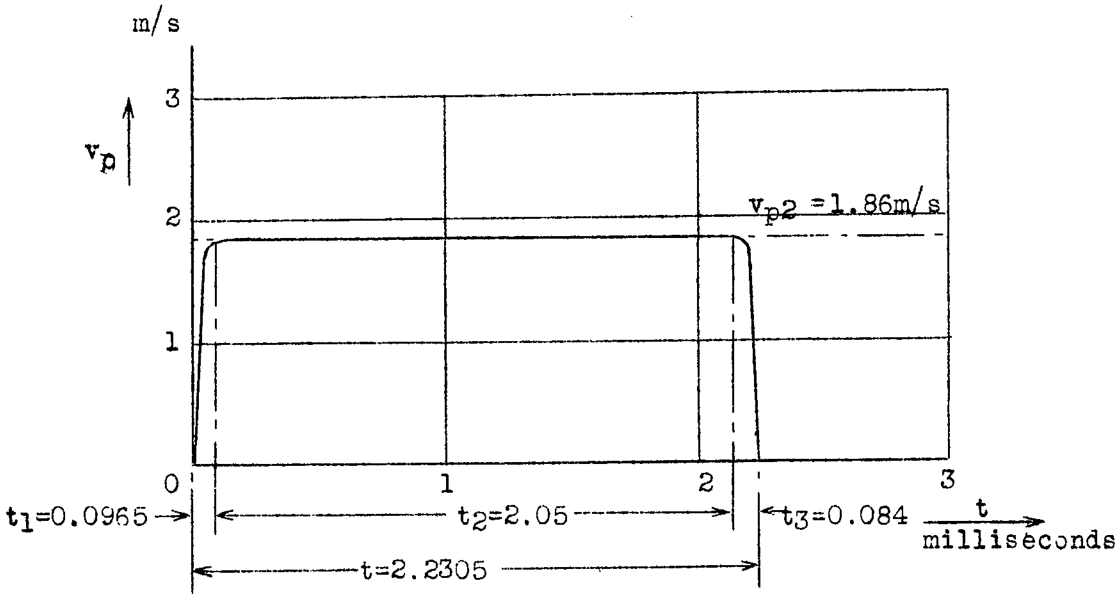


Figure 5.- Diagram of injection.

The cam profile is supposed to give a constant speed v_p to the plunger during the injection. v_p and S_3 are the speed and the section of the pump plunger, S_7 the section of the spray nozzles, and S_4-S_5 the differential section of the needle.

For all other engine speeds, section S_7 of the nozzles is poorly adapted. For example, if the engine speed is slower, the speed of the plunger will be likewise, and the pressure p_i of the flow will be lower also. Since the spring R permits the needle to open only for a higher pressure p_i, which cannot be maintained by the law of continuity, the result is an injection by jerks and vibrations of the differential needle which are attributed, rightly or wrongly, to phenomena of pressure waves and resonance.

When an injector operates under those conditions, the fuel flow is no longer controlled by the section S_7, which should, obligatorily, be the weakest section of passage between the pump and the working chamber of the engine, but by the height of lift of the vibrating needle, which flattens the fuel before entering the nozzles and breaks the penetration of the fuel jets in the working chamber.

Returning to the feed systems actuated by engine compression, it is clear that the pump speed v_p is here independent of that of the engine.

The duration of injection has for index, not a crankshaft angle turning at a variable speed n, but a time interval t, which depends solely upon compression ratio, the different sections of the device, and the masses in motion.

The initial force of spring R amounts to very little and admits of only an upper limit

$$R = p_{i\infty} S_4$$

for which the pump remains in rest position.

In practice it suffices to so select the return of spring R that the pressure of the needle on its seat assures a tight seal of the pump chamber. This pressure should be of the order of 5 to 10 kg/mm^2 of seat-bearing surface. The flow of the pump chamber at the outlet of the tubes conforms to the law of continuity and the needle does not vibrate during injection.

In practice, the pump motion may, without appreciable error, be considered as being effected at speed $v_{p\infty}$ or, as we shall term it, limit speed.

To get an approximate idea of the injection without computing the points of the exponential curve, the duration t of an injection is divided into three periods:

t_1, period of opening of the needle.

t_2, period of injection at limit speed.

t_3, period of closure of needle after percussion of the moving part on the feed stop 3.

6. Period of opening of needle.- When piston 10 leaves its seat S_1, the force of acceleration F of the pump is:

$$F = P - (A + H)$$

where

P is the pressure of the gas on piston 14 . . . . $P = p_c\ S_2$

A force of the spring regulating the point of injection . . . . $A = p_c\ S_1$

H hydraulic brake of the pump . . . . $H = p_i\ S_3$

As may be seen in figure 3, the inlet valve 12, with triangular guidance, has a cavity forming a compensating chamber. Thus, the hydraulic brake H starts by being zero at the beginning of motion, piston 14 rises quickly from its seat S_1, and the compression gases exert their force p_c on the large section S_2 of the piston, and the brake H continues up to a value

$$\frac{R}{S_4}\ S_3$$

when the needle quits its seat. It subsequently decreases to $R\ S_3/S_5$, where the needle meets its lifting stop, then increases again to the limit $p_i\ S_3$. These changes are so rapid that one can take a mean value of the hydraulic brake during the first period:

$$H = \frac{R\ S_3}{S_4}$$

which reverts to assuming the uniform acceleration a for computing the time of opening of the needle:

$$a = \frac{F}{m}$$

Then the duration of opening is

$$t_1 = \frac{v_p\ m}{F}$$

Thus, it is seen that, the greater the initial force of spring R in a given device, the longer the period of opening.

The path traveled during time interval t_1 is:

$$c_1 = \frac{F\ t^2}{2m}$$

and the feed injected:

$$Q_1 = S_3\ c_1$$

7. Needle-closing time.- With c_3 denoting the course of the moving part of the device after striking the shock absorber B of the feed screw 3, and knowing the speed $v_{p\infty}$ of the moving system at the moment of striking, we can write:

$$m\ \frac{v_{p\infty}^2}{2} = (p_{i_2}\ S_3 + A_3 + B - p_c\ S_2)\ c_3$$

where A_3 and B record the forces of return spring and shock absorber at the moment of striking. We have:

$$c_3 = \frac{m\ v_{p\infty}^2}{2(p_{i_2}\ S_3 + A_3 + B - p_c\ S_2)}$$

The amount of fuel injected during this interval of closing will be:

$$Q_3 = c_3 \; S_3$$

The course of the closure is traveled at a speed which decreases from $v_{p\infty}$ to 0. This course measures only several hundredths of millimeters and, the duration of the closing period can, without appreciable error, be computed for a speed equal to $\frac{v_{p\infty}}{2}$,

$$t_3 = \frac{2c_3}{v_{p\infty}}$$

The closing of the needle takes place under the simultaneous action of its spring R and its kinetic energy when the needle opens positively, i.e., in the sense of flow of fuel.

The best injection will be that taking place at limit pressure $p_{l\infty}$. The time of opening t_1 and closing t_3 of the needle valve, should therefore be reduced as much as possible.

The time t_3 is so much shorter as the shock absorber is stronger, but in no case should it be suppressed, because its intervention at the end of the stroke assures the elastic balancing of the gas pressure on the piston, and thus avoids the exchange of speed of the needle and of the sleeve pump at the moment of striking. Absence of the shock absorber instantaneously induces, by bouncing of the needle on its seat, the unpriming of the apparatus by an entry of compressed gas into the pump chamber.

8. Duration of injection under limiting conditions.- Knowing the quantity Q of one injection, the quantity Q_2 injected under limiting conditions $v_{p\infty}$ and $p_{i\infty}$ or, simply, v_{p_2} and p_{i_2}, is:

$$Q_2 = Q - (Q_1 + Q_3)$$

The duration t of an injection Q under limiting conditions being

$$t = \frac{c}{v_{p_2}} \quad \text{where} \quad c = \frac{Q}{S_3}$$

is the entire course of the pump, the duration t_2 of the injection at pressure p_{i_2} is deduced at:

$$t_2 = \frac{t\ Q_2}{Q}$$

and

$$T = t_1 + t_2 + t_3$$

is the actual time of an injection. It is very little greater than the injection t under limiting conditions.

9. <u>Numerical calculation of the device in the Winterthur engine</u>.- The given data are:

$D_1 = 10$ mm $D_2 = 24$ $D_3 = 5$ $D_4 = 1.6$ $D_5 = 3$ $D_7 = 4 \times 0.20$ mm

$S_1 = 78.54$ mm² $S_2 = 452.4$ $S_3 = 19.63$ mm²

$S_4 = 2.01$ $S_5 = 7.07$ $S_7 = 0.1256$ mm²

Engine compression is 32 atm at 800 rpm $p_c = 32$ atm

Maximum lift of pump . . . . $c = 4$ mm

Maximum feed $Q = c\ S_3$ . . . . $Q = 78.54$ mm³

Weight of moving parts, 50 grams

Mass m . . . . $m \cong 0.0050$

Find: time, stroke, feed, and pressure, during the three periods of an injection of 78.54 mm³.

Under limiting conditions:

Limiting pressure of injection . . $p_{i_2} = \dfrac{p_c(S_2 - S_1)}{S_3} = 610$ atm

Limiting speed of injection . . $v_{i_2} = 12\sqrt{p_{i_2} - p_c} = 290$ m/s

Limiting speed of pump. . . . . $v_{p_2} = \dfrac{S_7}{S_3}\ v_{i_2} = 1.86$ m/s

Duration of injection . . . . $t = \dfrac{c}{v_{p_2}} = 0.00215$ s

Needle Lift

Initial force of spring R	$R = 2{,}400$ kg
Pressure at lift of needle $p_{i_1} = R/S_4$.	$p_{i_1} = 120$ atm
Pressure of compression $P = p_c\ S_2$ . . .	$P = 145$ kg
Return spring $A = p_c\ S_1$	$A = 25$ kg
Hydraulic brake $H = p_{i_1}\ S_3$	$H = 23{,}500$ kg
Force of acceleration $F = P - (A + H)$.	$F = 96{,}500$ kg

$$\text{Time} \quad t_1 = \frac{v_{p_2}\ m}{F} = 0.0000965 \text{ s}$$

$$\text{Lift} \quad c_1 = \frac{F\ t^2}{2m} = 0.0895 \text{ mm}$$

Feed quantity $Q_1 = c_1\ S_3 = 1.75\ \text{mm}^3$

By raising the pressure p_{i_1} at rise of needle, by increasing the tare of spring R, or reducing the section S_4 while maintaining the same spring R as, for example,

$$D_4 = 1 \text{ mm} \ldots S_4 = 0.7854 \text{ mm}^2 \qquad R = 2400 \text{ kg}$$

the pressure at needle rise becomes $p_{i_1} = 305$ atmospheres, the hydraulic brake $H = 60$ kg, and the acceleration, $F = 60$ kg.

Then, we find:

Duration ..... $t_1 = 0.000155$ s

Lift .... $c_1 = 0.144$ mm

Feed quantity. $Q_1 = 2.83\ \text{mm}^3$

Thus, it is seen that the duration of opening of the needle is merely a few hundredths of a millisecond and that it slightly increases in relation to the pressure p_{i_1} at needle lift, in consequence, in relation to the tare of spring R.

Closing of Needle

For a shock absorber of B = 100 kg, and knowing that the specific force of spring R increases 3 kg/mm of deflection, the expression of the kinetic energy gives the value of the travel of the moving part on the damper B of the stop:

$$\text{Stroke} \quad c_3 = \frac{m\, v_{p_2}^2}{2(p_{i_2}\, S_3 + A_3 + B - p_c\, S_2)} = 0.078 \text{ mm}$$

$$\text{Duration} \quad t_3 = \frac{2c_3}{v_{p_2}} = \ldots\ldots\ldots\ldots\ldots \quad 0.000084 \text{ s}$$

$$\text{Feed volume} \quad Q_3 = c_3\, S_3 = \ldots\ldots\ldots\ldots\ldots \quad 1.53 \text{ mm}^3$$

Injection under Limiting Conditions

$$\text{Feed volume} \quad Q_2 = Q - (Q_1 + Q_3) = \ldots\ldots \quad 75.26 \text{ mm}^3$$

$$\text{Duration} \quad t_2 = \frac{t\, Q_2}{Q} = \ldots\ldots\ldots\ldots\ldots \quad 0.00205 \text{ s}$$

$$\text{Travel} \quad c_2 = \frac{Q_2}{S_3} = \ldots\ldots\ldots\ldots\ldots \quad 3.8325 \text{ mm}$$

Hence:

Injection pressure .... p_{i_2} = 610 atm

Speed of injection of nozzles .... v_{i_2} = 290 m/s

Speed of pump .... v_{p_2} = 1.86 m/s

Duration sec	Feed volume mm³	Stroke mm	Crankshaft angle at 800 rpm
t_1 = 0.0000965	Q_1 = 1.75	c_1 = 0.0895	φ : 27' 48"
t_2 = 0.0020500	Q_2 = 75.26	c_2 = 3.8325	φ : 9° 50'
t_3 = 0.0000840	Q_3 = 1.53	c_3 = 0.0780	φ : 24' 12"
T = 0.0022305	Q = 78.54	c = 4	φ : 10° 42'
as against			
T = 0.00215	Q = 78.54	c = 4	φ : 10° 18"

for the injection entirely under limiting conditions.

Hence, as indicated by equation (8), the time of opening and closing of the needle is negligible, and the total injection may be computed as under limiting conditions.

The injection card is reproduced in figure 5.

Chapter 2

TEST STAND RECORDING

10. Changes of pressure and of rate of injection.- The experiments included the recording of consumption and output at 800 rpm on the Winterthur engine, with different feed systems (figs. 1 and 2), pressures, and rates of injection.

The maximum horsepowers were measured for the dynamometer load, which permits a speed of 800 rpm.

Three sets of curves corresponding to:

610 ..... 425 ... and ..... 310 atm

injection pressures were recorded for different durations of injection.

The 610, 425, and 310 atm injection pressures were obtained by utilizing successively, plungers and sleeve pumps of 5-, 6-, and 7-millimeter diameter, in the same device.

The duration of injection is modified by changing the section S_7 of the spray nozzles. The same injection sleeves, 10, served in the three test series. The length of the nozzles was uniformly chosen at 7 to 8/10 millimeter.

The results therefore are strictly comparable, since the passive resistance, the moving mass, the flattening of the gas at entry of cylinder 2, and the flattening of the fuel under pressure are exactly the same for all plotted curves.

By computing at limiting conditions, the injection

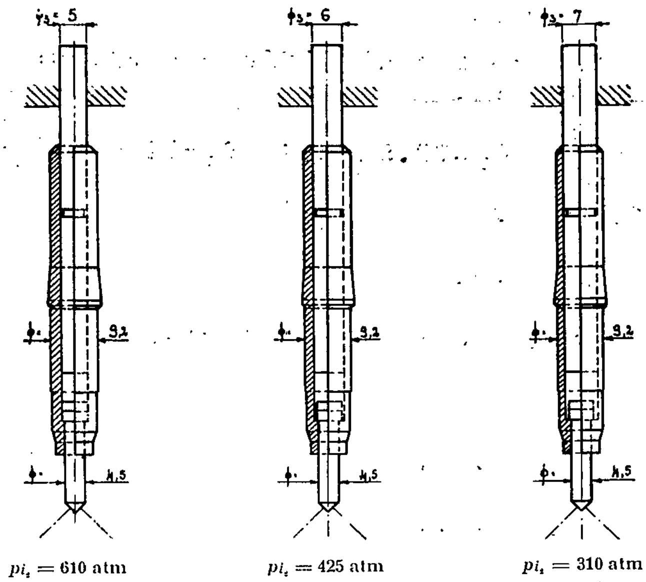


Figure 6.- Sleeves and plungers used for varying injection pressures.

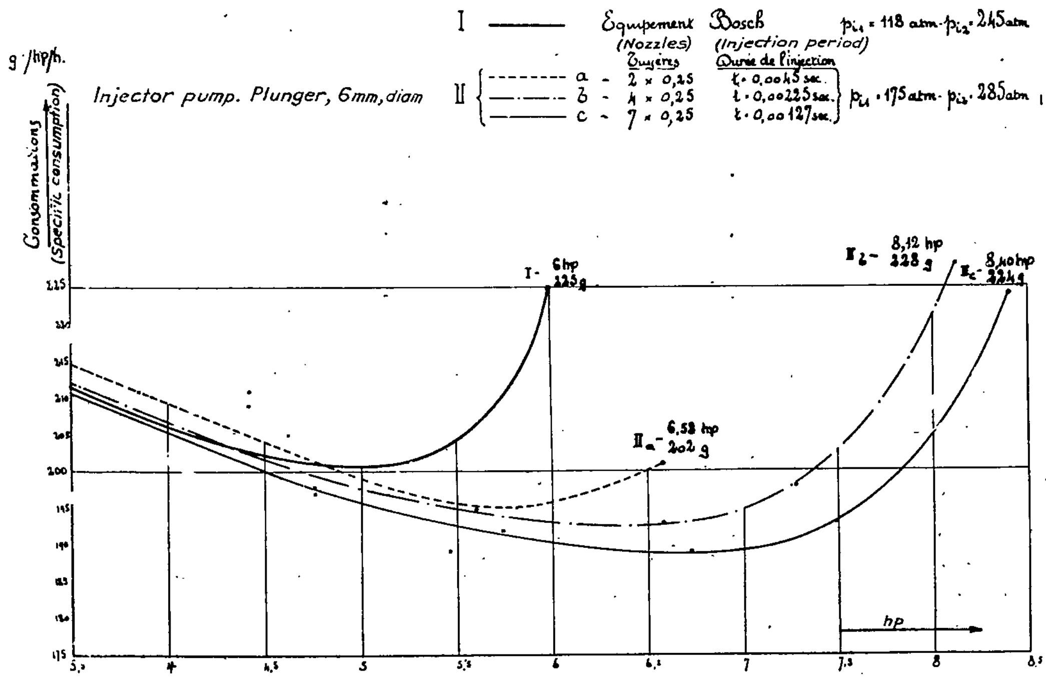


Figure 28.- Comparative power and consumption curves, plunger: 6mm dia.

produced with the device at different speeds and pressures, the constant data are:

$D_1 = 10$ mm $S_1 = 78.54$ mm^2 $D_2 = 24$ mm $S_2 = 452.4$ mm^2

$R = 2400$ kg $Q = 80.00$ mm^3

The variable data are:

For modifying the injection pressure:

		I	II	III	
Injection pressures		610	425	310	atm
Diameter of plunger	D_3	5	6	7	mm
Section of plunger	S_3	19.63	28.27	38.48	mm^2
Maximum lift	c	4	3	2	mm

To modify the duration of injection:

No.	Number of nozzles	Diameter mm	Section S_7 mm^2	Diameter D_4 mm	Section S_4 mm^2	Pressure p_{i_1} atm
a	4	0.20	0.1256	1.6	2.0106	120
b	4	.20	.1256	1.0	.7854	305
c	4	.30	.2827	1.6	2.0106	120
d	6	.25	.2945	1.6	2.0106	120
e	10	.20	.3142	1.6	2.0106	120
f	6	.30	.4240	1.6	2.0106	120

The injector nozzles a and b also have four 0.20-millimeter openings, but the needle of the first opens at the pressure p_{i_1} of 120 atmospheres; that of the second, at p_{i_1} of 305 atmospheres.

The calculation, at limit conditions, of the injection produced by the experimental device at different speeds and pressures, gave the figures appended in the following table:

Table A

Test No.	Plunger diameter	Number of nozzles	Diameter	Inject'n press.	Pump speed	Duration of injection	Duration of injection
	mm		mm	atm	m s	s	degres
I-a	5	4	0,20	610	1,86	0,00215	10° 18′
I-b	5	4	0,20	610	1,86	0,00215	10° 18′
I-c	5	4	0,30	610	4,18	0,00096	4° 36′
I-d	5	6	0,25	610	4,35	0,00092	4° 25′
I-e	5	10	0,20	610	4,65	0,00086	4° 7′
I-f	5	6	0,30	610	6,25	0,00064	3° 5′
II-a	6	4	0,20	425	1,06	0,00283	13° 35′
II-b	6	4	0,20	425	1,06	0,00283	13° 35′
II-c	6	4	0,30	425	2,38	0,00126	6° 3′
II-d	6	6	0,25	425	2,48	0,00121	5° 48′
II-e	6	10	0,20	425	2,65	0,00113	5° 26′
II-f	6	6	0,30	425	3,56	0,00084	4° 2′
III-a	7	4	0,20	310	0,65	0,00308	14° 48′
III-b	7	4	0,20	310	0,65	0,00308	14° 48′
III-c	7	4	0,30	310	1,47	0,00136	6° 32′
III-d	7	6	0,25	310	1,53	0,00131	6° 18′
III-e	7	10	0,20	310	1,63	0,00123	5° 54′
III-f	7	6	0,30	310	2,20	0,00091	4° 22′

II. INJECTION OF GAS OIL.

Using gas oil of 0.836 density the following measurements were made; at every change of load of the engine a diagram was plotted with the Lehman and Michels indicator, the point of injection then being so adjusted that the pressure at a point in the combustion cycle does not exceed 60 atm. the cooling water temperature was always kept at 50° C at the outlet.

TABLEAU Ia $p_{i_2} = 610$ atm $S_7 = 0{,}1256$ mm² — 4 × 0,20 — $p_{i_1} = 120$ atm.

Engine speed	Tare of balance	Horse-power	Time to burn 100 cm³	Specific consumption	Return of spring A	Inject'n advance angle	Maximum pres. of cycle
rpm	kg	hp	s	g/hp/h	kg	degrés	atm
800	0,500	0,40	7′03″	1.770	17	10	40
800	3,100	2,48	5′22″	373	17	10	44
800	5,900	4,72	4′10″	254	18,5	9	46
800	8,500	6,80	3′20″	220	18,5	9	47
800	9,500	7,60	3′ 2″	216	20,5	7	45
800	10,700	8,56	2′45″	212	20,5	7	47
800	13,000	10,40	2′18″	208	22,5	5	48
800	14,200	11,36	2′04″	212	22,5	5	50
800	15,100	12,08	1′56″	214	22,5	5	52
800	16,200	12,96	1′43″	224	22,5	5	54
800	17,100	13,68	1′35″	230	22,5	5	54

Normal running, idling, 200 rpm, minimum consumption 208 g/hp/h at 10.4 hp, 54.5mm³/ injection; maximum injection: 79 mm³, maximum stroke 4.02 mm.

TABLEAU I b

$p_{i_2} = 610$ atm $S_7 = 0{,}1256$ mm² — 4 × 0,20 — $p_{i_1} = 305$ atm.

Engine speed	Tare of balance	Horse-power	Time to burn 100 cm³	Specific consumption	Return of spring A	Inject'n advance angle	Maximum pres. of cycle
rpm	kg	hp	s	g hp/h	kg	degrés	atm
800	0,500	0,40	6′52″	1.820	17	10	40
800	3,300	2,64	5′ 9″	378	17	10	44
800	6,200	4,96	4′ 1″	252	17	10	45
800	8,700	6,96	3′16″	220	18,5	9	50
800	9,800	7,84	3′	213	20,5	7	50
800	11,400	9,12	2′37″	210	22,5	5	48
800	13,800	11,04	2′14″	203	22,5	5	48
800	14,700	11,76	2′ 5″	204	22,5	5	49
800	15,600	12,48	1′54″	211	22,5	5	49
800	16,700	13,36	1′42″	221	22,5	5	50

Normal run, idling 200 rpm.
Minimum consumption: 203 g/hp/h at 11.04 hp, 57.8 mm³ injection.
Maximum injection: 76.5 mm³/ injection, maximum stroke: 3.9 mm.
The maximum power developed is a little less, while the specific fuel consumptions are slightly improved.
The choice of needle closing pressure has no capital influence.

TABLEAU I c

$p_{i_2} = 610$ atm $S_7 = 0{,}2827$ mm² — 4 × 0,30 — $p_{i_1} = 120$ atm.

Engine speed	Tare of balance	Horse-power	Time to burn 100 cm³	Specific consumption	Return of spring A	Inject'n advance angle	Maximum pres.of cycle
rpm	kg	hp	s	g/hp/h	kg	degrés	atm
800	0,500	0,40	6′34″	1.900	17	10	40
800	3,400	2,72	4′50″	380	17	10	48
800	6,200	4,96	3′57″	256	18,5	9	52
800	8,900	7,32	3′10″	216	20,5	7	50
800	10,100	8,08	2′55″	212	22	6	50
800	11,400	9,12	2′39″	208	22	6	52
800	13,800	11,04	2′14″	202	23,5	4	50
800	14,900	12,32	2′ 2″	200	23,5	4	51
800	15,900	12,72	1′54″	207	23,5	4	52
800	16,900	13,52	1′42″	219	23,5	4	56
800	17,700	14,16	1′28″	241	23,5	4	60

Faster run, idling 200 rpm.
Minimum consumption: 200 g/hp/h at 12.32 hp, 61.5 mm³ injected.
Maximum injection: 85 mm³, maximum stroke: 4.35 mm.

TABLEAU I *d*

$p_{i_2} = 610$ atm $S_7 = 0,2945$ mm² — $6 \times 0,25$ — $p_{i_1} = 120$ atm.

Engine speed	Tare of balance	Horse-power	Time to burn 100 cm³	Specific consump tion	Return of spring A	Inject'n advance angle	Maximum pres.of cycle
rpm	kg	hp	s	g/hp/h	kg	degres	atm
800	0,500	0,40	7'16"	1.720	17	10	42
800	3,300	2,64	5'14"	362	17	10	47
800	6,100	4,88	4' 2"	253	18,5	9	48
800	8,600	6,88	3'17"	221	20,5	7	51
800	9,900	7,92	2'59"	211	22,5	5	52
800	11,100	8,88	2'45"	204	22,5	5	53
800	13,500	10,80	2'23"	194	22,5	5	54
800	14,800	11,84	2'13"	190	22,5	5	63
800	16,000	12,80	2' 3"	189	24,5	1	52
800	17,100	13,68	1'51"	197	24,5	1	52
800	18,300	14,64	1'40"	205	24,5	1	54
800	19,200	15,36	1'29"	219	24,5	1	60

Fast run, no knock. Maximum delay in the last four points. Idling 200 rpm.
Minimum consumption: 189 g/hp/h at 12.80 hp, 61 mm³ injection.
Maximum injection: 84.5 mm³, maximum stroke: 4.3 mm.

TABLEAU I *e*

$p_{i_2} = 610$ atm $S_7 = 0,3142$ mm² — $10 \times 0,20$ — $p_{i_1} = 120$ atm.

Engine speed	Tare of balance	Horse-power	Time to burn 100 cm³	Specific consump tion	Return of spring A	Inject'n advance angle	Maximum pres.of cycle
rpm	kg	hp	s	g/hp/h	kg	degrés	atm
800	0,700	0,56	6'30"	1.380	17	10	42
800	3,700	2,96	4'50"	349	17	10	48
800	6,600	5,28	3'46"	252	18,5	9	50
800	9,000	7,20	3'01"	230	18,5	9	52
800	10,000	8,00	2'49"	222	22,5	5	48
800	11,300	9,04	2'36"	212	22,5	5	48
800	13,900	11,12	2' 4"	217	22,5	5	48
800	14,800	11,84	1'56"	218	22,5	5	51
800	15,900	12,72	1'47"	219	24,5	1	60
800	16,800	13,44	1'40"	223	24,5	1	62
800	17,700	14,16	1'22"	250	24,5	1	65

Run violent; crests of explosions at almost constant volume.
Minimum consumption: 212 g/hp/h at 9.04 hp, 48 mm³ injection volume.
Maximum injection: 81.5 mm³, maximum stroke: 4.15 mm.

TABLEAU I *f*

$p_{i_2} = 610$ atm $S_7 = 0,4240$ mm² — $6 \times 0,30$ — $p_{i_1} = 120$ atm.

Engine speed	Tare of balance	Horse-power	Time to burn 100 cm³	Specific consump tion	Return of spring A	Inject'n advance angle	Maximum pres. of cycle
rpm	kg	hp	s	g/hp/h	kg	degrés	atm
800	0,700	0,56	6'36"	1.350	17	10	35
800	3,400	2,72	4'58"	370	17	10	40
800	6,400	5,12	3'53"	252	18,5	9	45
800	8,900	7,12	3'10"	222	18,5	9	53
800	10,100	8,08	2'53"	214	20,5	7	53
800	11,400	9,12	2'40"	205	20,5	7	56
800	13,400	10,72	2'17"	203	22,5	5	49
800	14,700	11,76	2' 7"	201	23,5	4	43
800	16,100	12,88	1'55"	202	23,5	4	52
800	17,200	13,76	1'45"	207	24,5	1	56
800	18,400	14,72	1'36"	213	24,5	1	60
800	19,500	15,60	1'25"	219	24,5	1	62

Run very brisk without knocking; idling 200 rpm, pick-up very vigorous.
Specific consumption almost constant between 8 and 15.6 hp.
Minimum consumption: 202 g/hp/h at 12.88 hp, 65 mm³ injection.
Maximum injection: 88 mm³, maximum stroke: 4.48 mm.

Injection pressure: 425 atm

TABLEAU II *a*

$p_{i_2} = 425$ atm $S_7 = 0,1256$ mm² — $4 \times 0,20$ — $p_{i_1} = 120$ atm.

Engine speed	Tare of balance	Horse-power	Time to burn 100 cm³	Specific consump tion	Return of spring A	Inject'n advance angle	Maximum pres.of cycle
rpm	kg	hp	s	g/hp/h	kg	degres	atm
800	0,700	0,56	6'35"	1.350	17	10	37
800	3,500	2,80	4'50"	370	17	10	43
800	6,200	4,96	3'52"	261	17	10	49
800	8,800	7,04	3'10"	224	18,5	9	48
800	10,000	8,00	2'53"	217	20,5	7	45
800	11,400	9,12	2'35"	212	20,5	7	45
800	13,700	10,96	2'10"	209	22,5	5	40
800	14,800	11,84	2'	210	22,5	5	50
800	15,900	12,72	1'48"	218	22,5	5	50
800	17,000	13,60	1'31"	243	22,5	5	52
800	17,700	14,16	1'22"	260	22,5	5	54

Run normal, the peak pressure of the cycle increases a little with the charge; idling at 200 rpm.
Minimum consumption: 209 g/hp/h at 10.96 hp, 57.7 mm³ injection.
Maximum injection: 89 mm³, maximum stroke: 3.15 mm.

TABLEAU II *b*

$p_{i_2} = 425$ atm $S_7 = 0{,}1256$ mm² — $4 \times 0{,}20$ — $p_{i_1} = 305$ atm.

Engine speed	Tare of balance	Horse-power	Time to burn 100 cm³	Specific consumption	Return of spring A	Inject'n advance angle	Maximum pres. of cycle
rpm	kg	hp	s	g/hp/h	kg	degrés	atm
800	0,700	0,56	6'50"	1.310	17	10	38
800	3,300	2,64	5' 4"	375	17	10	41
800	5,900	4,72	4' 5"	260	20,5	7	40
800	8,600	6,88	3'17"	222	20,5	7	44
800	9,700	7,76	3'	215	20,5	7	45
800	11,100	8,88	2'42"	208	20,5	7	45
800	13,000	10,40	2'19"	207	20,5	7	45
800	14,200	11,36	2' 4"	204	20,5	7	46
800	15,500	12,40	1'52"	216	20,5	7	47
800	16,500	13,20	1'41"	225	20,5	7	47
800	17,500	14,00	1'25"	252	20,5	7	52

Without changing the injection advance the pressure peak of the cycle increases very little with the charge.
Normal run - idling: 200 rpm.
Minimum consumption: 204 g/hp/h at 11.36 hp, 60.6 mm³.
Maximum injection: 88 mm³, maximum stroke: 3.10 mm.

TABLEAU II *c*

$p_{i_2} = 425$ atm $S_7 = 0{,}2827$ mm² — $4 \times 0{,}30$ — $p_{i_1} = 120$ atm.

Engine speed	Tare of balance	Horse-power	Time to burn 100 cm³	Specific consumption	Return of spring A	Inject'n advance angle	Maximum pres. of cycle
rpm	kg	hp	s	g/hp/h	kg	degres	atm
800	0,500	0,40	6'40"	1.870	17	10	38
800	3,300	2,64	5' 6"	370	17	10	45
800	5,700	4,56	4' 7"	261	17	10	49
800	8,200	6,56	3'22"	226	18,5	9	52
800	9,400	7,52	3' 3"	218	20,5	7	53
800	10,900	8,72	2'45"	208	22,5	5	52
800	12,900	10,32	2'24"	201	22,5	5	56
800	13,900	11,32	2'12"	199	23,5	4	52
800	15,400	12,32	1'58"	205	24,5	1	50
800	16,400	13,12	1'47"	206	24,5	1	53
800	17,700	14,16	1'36"	221	24,5	1	55
800	18,500	14,80	1'20"	252	24,5	1	58

Run faster; injection at dead center; idling: 200 rpm.
Minimum consumption: 199 g/hp/h at 11.32 hp, 57 mm³ injection.
Maximum injection: 94 mm³, maximum stroke: 3.3 mm.

TABLEAU II *d*

$p_{i_2} = 425$ atm $S_7 = 0{,}2945$ mm² — $6 \times 0{,}25$ — $p_{i_1} = 120$ atm.

Engine speed	Tare of balance	Horse-power	Time to burn 100 cm³	Specific consumption	Return of spring A	Inject'n advance angle	Maximum pres. of cycle
rpm	kg	hp	s	g/hp/h	kg	degrés	atm
800	0,700	0,56	6'30"	1.375	17	10	32
800	3,300	2,64	4'53"	388	18,5	9	37
800	6,100	4,88	3'53"	263	18,5	9	39
800	8,600	6,88	3'15"	224	18,5	9	48
800	9,800	7,84	2'59"	213	18,5	9	50
800	11,300	9,04	2'40"	207	20,5	7	53
800	13,300	10,64	2'25"	195	20,5	7	58
800	14,500	11,60	2'15"	192	23,5	4	53
800	15,500	12,40	2' 4"	196	23,5	4	56
800	16,700	13,36	1'51"	202	24,5	1	53
800	17,900	14,32	1'39"	211	24,5	1	56
800	18,900	15,12	1'27"	229	24,5	1	60

Run fast - quick pickup - idling: 200 rpm.
Minimum consumption: 192 g/hp/h at 11.60 hp, 55.6 mm³ injection.
Maximum injection: 86 mm³, maximum stroke: 3.04 mm.

TABLEAU II *e*

$p_{i_2} = 425$ atm $S_7 = 0{,}31416$ mm² — $10 \times 0{,}20$ — $p_{i_1} = 120$ atm.

Engine speed	Tare of balance	Horse-power	Time to burn 100 cm³	Specific consumption	Return of spring A	Inject'n advance angle	Maximum pres. of cycle
rpm	kg	hp	s	g/hp/h	kg	degrés	atm.
800	0,500	0,40	6'54"	1.800	17	10	39
800	3,100	2,48	5'32"	367	17	10	45
800	5,700	4,56	4'	264	17	10	.47
800	8,200	6,56	3'17"	233	20,5	7	48
800	9,600	7,68	2'58"	219	20,5	7	48
800	10,800	8,64	2'39"	218	20,5	7	50
800	13,500	10,80	2' 3"	225	22,5	5	49
800	14,100	11,28	1'55"	231	24,5	1	46
800	15,500	12,40	1'38"	246	24,5	1	52
800	16,500	13,20	1'22"	276	24,5	1	52
800	17,400	13,92	1'	359	24,5	1	60

Run fast - pickup mediocre, idling: 200 rpm.
Minimum consumption: 218 g/hp/h at 8.64 hp, 47 mm³ injection.
Maximum injection 125 mm³, maximum stroke: 4.40 mm, 359 g/hp/h at 13.92 hp. Injector with 10 orifices of 0.20 poorly suited.

TABLEAU II *f*

$p_{i_2} = 425$ atm $S_7 = 0{,}4240$ mm² — $6 \times 0{,}30$ — $p_{i_1} = 120$ atm.

Engine speed	Tare of balance	Horse power	Time to burn 100 cm³	Specific consump tion	Return of spring A	Inject'n advance angle	Maximum pres. of cycle
rpm	kg	hp	s	g/hp/h	kg	degrés	atm
800	0,500	0,40	6'34"	1.810	17	10	32
800	3,300	2,64	4'50"	392	17	10	40
800	6,100	4,88	3'50"	267	17	10	47
800	8,600	6,88	3'11"	228	18,5	9	50
800	9,600	7,68	3'	217	20,5	7	47
800	11,300	9,04	2'41"	205	22,5	5	47
800	13,500	10,80	2'20"	198	22,5	5	50
800	14,500	11,60	2'11"	197	24	3	50
800	15,700	12,56	1'58"	203	24	3	52
800	16,700	13,36	1'48"	206	24,5	1	57
800	17,900	14.32	1'38"	215	24,5	1	60
800	19,100	15,28	1'28"	224	24,5	1	60

Run fast, pickup excellent; idling 200 rpm.
Minimum consumption: 197 g/hp/h at 11.60 hp, 57.5 mm³ injection.
Maximum injection: 85 mm³. 15.28 hp; maximum stroke: 3 mm.

Injection pressure: 310 atm.

TABLEAU III *a*

$p_{i_2} = 310$ atm $S_7 = 0{,}1256$ mm² — $4 \times 0{,}20$ — $p_{i_1} = 120$ atm.

Engine speed	Tare of balance	Horse-power	Time to burn 100 cm³	Specific consump tion	Return of spring A	Inject'n advance angle	Maximum pres. of cycle
rpm	kg	hp	s	g.hp/h	kg	degrés	atm
800	0,600	0,48	6'26"	1.610	17	10	31
800	3,400	2,72	4'42"	392	17	10	35
800	6,200	4,96	3'39"	277	17	10	35
800	8,700	6,96	2'58"	243	17	10	37
800	9,900	7,92	2'39"	237	17	10	37
800	11,300	9,04	2'24"	231	17	10	40
800	13,500	10,80	2' 1"	231	17	10	41
800	14,700	11,76	1'42"	250	17	10	42
800	15,800	12,64	1'27"	271	17	10	42

Run very smooth, pickup mediocre, idling: 200 rpm.
Minimum consumption: 231 g/hp/h at 10.80 hp, 62 mm³ volume
Maximum injection: 86 mm³, maximum stroke: 2.23 mm.
Weak peak pressures in cycle; no adjustment of advance.

TABLEAU III *b*

$p_{i_2} = 310$ atm $S_7 = 0{,}1256$ mm² — $4 \times 0{,}20$ — $p_{i_1} = 305$ atm.

Engine speed	Tare of balance	Horse-power	Time to burn 100 cm³	Specific consump-tion	Return of spring A	Inject'n advance angle	Maximum pres. of cycle
rpm	kg	hp	s	g/hp/h	kg	degrés	atm
800	0,500	0,40	7'16"	1.715	17	10	36
800	3,300	2,64	5'12"	365	17	10	36
800	5,800	4,64	4' 3"	267	17	10	38
800	8,100	6,48	3'20"	231	17	10	40
800	9,000	7,20	2'57"	235	17	10	38
800	10,900	8,72	2'33"	226	17	10	36
800	13,100	10,48	2'09"	223	17	10	37
800	14,200	11,36	1'56"	228	17	10	39

Run very smooth, engine sluggish, no pickup, idling 200 rpm.
No change in advance. Peak pressure of cycle rises very little with charge.
Minimum consumption: 223 g/hp/h at 10.48 hp, 58 mm³ injected.
Maximum injection: 65 mm³ at 11.36 hp, maximum stroke: 1.7 mm.

TABLEAU III *c*

$p_{i_2} = 310$ atm $S_7 = 0{,}2827$ mm² — $4 \times 0{,}30$ — $p_{i_1} = 120$ atm.

Engine speed	Tare of balance	Horse-power	Time to burn 100 cm³	Specific consump tion	Return of spring A	Inject'n advance angle	Maximum pres. of cycle
rpm	kg	hp	s	g/hp/h	kg	degrés	atm
800	0,600	0,48	6'52"	1.515	17	10	33
800	3,100	2,48	5' 6"	398	17	10	40
800	5,600	4,48	4' 4"	274	17	10	43
800	8,400	6,72	3'20"	222	17	10	44
800	9,500	7,60	3'	219	17	10	44
800	10,800	8,64	2'43"	212	18,5	9	42
800	13,100	10,48	2'15"	211	18,5	9	46
800	14,100	11,28	2' 7"	209	18,5	9	48
800	15,300	12,24	1'51"	221	18,5	9	48
800	16,400	13,12	1'38"	233	18,5	9	48

Run smooth, pickup good, idling: 200 rpm.
Minimum consumption: 209 g/hp/h at 11.28 hp, 59 mm³ injected.
Maximum injection: 77 mm³ at 13.12 hp, maximum stroke: 2 mm.

TABLEAU III d

$p_{i_2} = 310$ atm $S_r = 0,2945$ mm² — 6 × 0,25 — $p_{i_1} = 120$ atm.

Engine speed	Tare of balance	Horse-power	Time to burn 100 cm³	Specific consumption	Return of spring A	Inject'n advance angle	Maximum pres. of cycle
rpm	kg	hp	s	g hp/h	kg	degres	atm
800	0,600	0,48	6'26"	1.625	17	10	35
800	3,400	2,72	4'46"	386	17	10	45
800	6,200	4,96	3'49"	264	17	10	52
800	8,900	7,12	3' 9"	223	18,5	9	54
800	10,000	8,00	2'54"	215	18,5	9	57
800	11,500	9,20	2'37"	207	22,5	5	54
800	13,700	10,96	2'16"	201	22,5	5	54
800	14,700	11,76	2' 8"	198	22,5	5	54
800	16,100	12,88	1'52"	207	24,5	1	54
800	17,100	13,68	1'39"	221	24,5	1	55
800	18,000	14,40	1'25"	245	24,5	1	57

Run faster - by super charge injection at dead center, pickup very good, idling: 200 rpm.
Minimum consumption: 198 g/hp/h at 11.76 hp, 59 mm³ injected volume.
Maximum stroke: 2.27 mm.
Maximum injection: 88 mm³, 14.40 hp .

TABLEAU III e

$p_{i_2} = 310$ atm $S_r = 0,314$ mm² — 10 × 0,20 — $p_{i_1} = 120$ atm.

Engine speed	Tare of balance	Horse-power	Time to burn 100 cm³	Specific consumption	Return of spring A	Inject'n advance angle	Maximum pres. of cycle
rpm	kg	hp	s	g hp/h	kg	degres	atm
800	0,500	0,40	6'58"	1.790	17	10	40
800	3,100	2,48	5' 3"	399	17	10	45
800	5,700	4,56	3'58"	276	17	10	47
800	8,000	6,40	3'15"	240	17	10	52
800	9,200	7,36	2'58"	230	18,5	9	48
800	10,700	8,56	2'40"	219	22,5	5	46
800	12,700	10,16	2'11"	225	22,5	5	46
800	13,800	11,04	2'	227	22,5	5	46
800	15,100	12,08	1'39"	250	22,5	5	46
800	16,200	12,96	1'15"	312	22,5	5	46

Engine sluggish, without pickup, idling: 200 rpm.
Minimum consumption: 219 g/hp/h at 8.56 hp, volume injected 47 mm³.
Maximum injection: 100 mm³, at 12.96 hp, maximum stroke: 2.6 mm.

TABLEAU III f

$p_{i_2} = 310$ atm $S_r = 0,4240$ mm² — 6 × 0,30 — $p_{i_1} = 120$ atm.

Engine speed	Tare of balance	Horse-power	Time to burn 100 cm³	Specific consumption	Return of spring A	Inject'n advance angle	Maximum pres. of cycle
rpm	kg	hp	s	g/hp/h	kg	degres	atm
800	0,500	0,40	6'42"	1.865	17	10	40
800	3,200	2,56	5'	391	17	10	46
800	5,700	4,56	3'58"	276	17	10	50
800	8,400	6,72	3'15"	229	18,5	9	48
800	9,500	7,60	2'59"	221	18,5	9	50
800	10,800	8,64	2'42"	213	22,5	5	52
800	12,900	10,32	2'18"	209	22,5	5	52
800	14,100	11,28	2' 8"	207	22,5	5	52
800	15,400	12,32	1'54"	211	22,5	5	52
800	16,700	13,36	1'38"	229	22,5	5	52
800	17,600	14,08	1'23"	257	22,5	5	52

Run normal, pickup good; idling: 200 rpm.
Peak pressure in cycle shows no increase with charge.
Minimum consumption: 207 g/hp/h at 11.28 hp; injection, 59 mm³.
Maximum injection: 90 mm³ at 14.08 hp,
Maximum stroke: 2.34 mm.

Kerosene density: 0.808
Injection pressure: 610 atm

Table Ib (Kerosene)

p=610 atm S=0.1256 mm²-4×0.20-p=305 atm t=0.00215 mm/s

Engine speed	Tare of balance	Horse-power	Time to burn 100 cm³	Specific consumption	Return of spring A	Inject'n advance angle	Maximum pres of cycle
rpm	kg	hp	s	g/hp/h	kg	degrés	atm
800	0,500	0,40	7'	1.725	20,5	7	38
800	3,200	2,56	5'	376	20,5	7	43
800	5,700	4,56	4' 3"	262	20,5	7	48
800	8,300	6,64	3'20"	222	20,5	7	46
800	9,400	7,52	2'57"	218	20,5	7	50
800	10,800	8,64	2'40"	210	20,5	7	52
800	12,900	10,32	2'17"	205	20,5	7	52
800	14,000	14,20	2' 4"	208	20,5	7	52
800	16,300	13,04	1'45"	211	20,5	7	53
800	17,300	13,84	1'32"	228	20,5	7	56

12. Injection of gasoline.- Injector No. 1-b was tested with a heavy fuel of 0.750 density; starting the engine, hot or cold, was impossible. The engine started with gas oil, since it is well known that in substituting gasoline for gas oil when the hot engine is turning at its normal speed, the injected fuel does not ignite, no matter what the injection advance.

A 50-percent blend of gasoline and gas oil permits spontaneous ignition when the volume injected is great, but the horsepower developed for this volume is very inferior.

Experience seems to indicate that the phenomena of autoignition in gasoline engines result from the compression of the fuel-air mixture, which must have greater knock characteristics than the fuel injected in the hot air of compression. But most often the knock arises from an incandescent point in the working chamber. This hot point is either a carbon deposit on the bottom of the piston, or on the exhaust valve, or some badly cooled metallic roughness.

13. Injection of kerosene.- The same experiment was repeated with kerosene of 0.808 density. Start from cold is instantaneous; the engine turns over; pick-up is vigorous. The recorded curve is as follows:

The engine acts perfectly; the diagrams indicate more moderate explosion crests than when using gas oil. Without changing the point of injection, the peak pressure of the cycle rises but slowly with the charge, and remains within acceptable limits.

The consumption is, in every point, comparable with that recorded with the same device using gas oil. (See table Ib.) The maximum power developed is higher by one-half horsepower. Idling is impeccable at 200 rpm - as for all tests made with gas oil.

14. Comparative measurements.- The basis of comparison was the recorded power and consumption curve of the Winterthur Diesel engine with normal equipment, Bosch pump, and injector.

The plunger of the Bosch pump has a diameter of 6.5 mm. $S_3 = 33.18$ mm². The injection needle (3 and 6 mm D) permits a differential section of 21.20 mm². The spring R of the needle, has a force of 37 kg, which defines the opening of the needle at a pressure $p_{i_1} = 175$ atm.

The section of passage of the nozzle (four holes of 0.20 mm diameter) is $S_7 = 0.1256$ mm². The pump is set for: start of injection, 15° B.T.C.; end of injection, maximum feed, 20° A.T.C.; amplitude at maximum feed, 35°.

TABLE IV. Bosch Calibration Curve

Gas-Oil density, 0.836 at 17° C

Engine speed rpm	Initial force of balance kg	Horse-power	Time for burning 100 cm³	Specific consumption g/hp/h	Injection advance deg	Maximum pressure of cycle atm
800	0.500	0.40	6' 47"	1,840	15	38
800	3.100	2.48	4' 40"	432	15	42
800	5.800	4.64	3' 59"	270	15	48
800	8.200	6.56	3' 12"	238	15	54
800	9.600	7.68	2' 54"	224	15	58
800	10.800	8.64	2' 36"	222	15	58
800	11.800	9.44	2' 16"	234	15	59
800	12.800	10.24	1' 58"	248	15	57

At 10 hp the engine knocks harshly. Idling, impossible to maintain below 400 rpm. Pick-up, mediocre. Volume injected at maximum power of 10.24 hp (100 cm³ in 118 s) is 63.7 mm³, making the effective pump stroke, 1.95 mm=c.

The active angle of the cam controlling the pump is therefore 19° 30', and the duration of maximum injection, 0.00406 second.

Assuming the speed v_p of the plunger as constant for the duration of the injection, $v_p = 0.48$ m/s. The pressure p_{i_2} maintained by the pump is here, as seen previously:

$$p_{i_2} = \left(\frac{S_3}{12}\frac{v_p}{S_7}\right)^2 = 112.4 \text{ atm}$$

The spring R permits the needle to open at a pressure $p_{i_1} = 175$ atm, and we see that this pressure is not maintained by the pump. The injection is by jerks.

For a more complete check on the laws of injection in the light of these experiments, the following changes were made on the Bosch equipment: Start of injection, 10° B.T.C.; end of maximum injection, 8° A.T.C.; amplitude at maximum feed, 18°; spring, R 25.900 kg; opening pressure, p_{i_1} 120 atm.

TABLE V. Bosch Calibration Curve

Amplitude 18° Advance 10° Pressure p_{i_1} = 120 atm

Engine speed rpm	Initial force of balance kg	Horse-power	Time for burning 100 cm³	Specific consumption g/hp/h	Injection advance deg	Maximum pressure of cycle atm
800	0.600	0.48	6' 30"	1,610	10	41
800	3.300	2.64	4' 44"	401	10	48
800	6.100	4.88	3' 50"	268	10	53
800	8.800	7.04	3' 10"	225	10	60
800	10.000	8.00	2' 55"	215	10	62
800	11.300	9.04	2' 39"	210	10	64
800	13.500	10.80	2' 17"	203	10	65
800	14.900	11.92	2' 5"	201	10	66
800	16.100	12.88	1' 53"	207	10	66
800	17.300	13.84	1' 34"	231	10	63

Run lively; pick-up better. The engine does not knock when supercharged. Idling: 350-400 rpm. The peak pressures of the cycle are raised. At the same setting of the Bosch pump, a last curve is plotted by simply doubling the opening pressure p_{i_1}: R = 51 kg; P_{i_1} = 240 atm.

TABLE VI. Bosch Calibration Curve

Amplitude 18° Advance 10° Pressure p_{i_1} = 240 atm

Engine speed rpm	Initial force of balance kg	Horse-power	Time for burning 100 cm³	Specific consumption g/hp/h	Injection advance deg	Maximum pressure of cycle atm
800	0.600	0.48	6' 30"	1,610	10	43
800	3.300	2.64	4' 54"	388	10	48
800	6.100	4.88	3' 58"	260	10	52
800	8.800	7.04	3' 15"	220	10	58
800	10.000	8.00	2' 56"	214	10	58
800	11.400	9.12	2' 39"	208	10	60
800	13.300	10.64	2' 17"	206	10	64
800	14.400	11.52	2' 9"	202	10	62
800	15.800	12.64	1' 53"	211	10	62
800	16.900	13.52	1' 38"	226	10	62

Doubling the opening pressure p_{i_1} does not change the running of the engine. The specific consumption is somewhat better; the maximum power developed, a little poorer. Idling at 400 rpm.

From the last measurement, 13.52 horsepower, where 100 cm³ of gas oil is burned in 98 seconds, we can compute:

The useful stroke of the pump c, 2.35 mm

Amplitude of maximum injection φ, 12° 21'

Duration of maximum injection t = 0.00257 s

Piston speed of the pump v_p, 0.915 m/s

Pressure maintained by the pump p_{i_2}, 400 atm

Measures V and VI, comparable in all points, clearly show that the force R of the spring of the differential needle plays no essential part in the injection and the combustion.

Figures 7, 8, and 9 show the power developed by the Winterthur Diesel for the three sets of consumption curves in comparison with the calibration curves IV and VI.

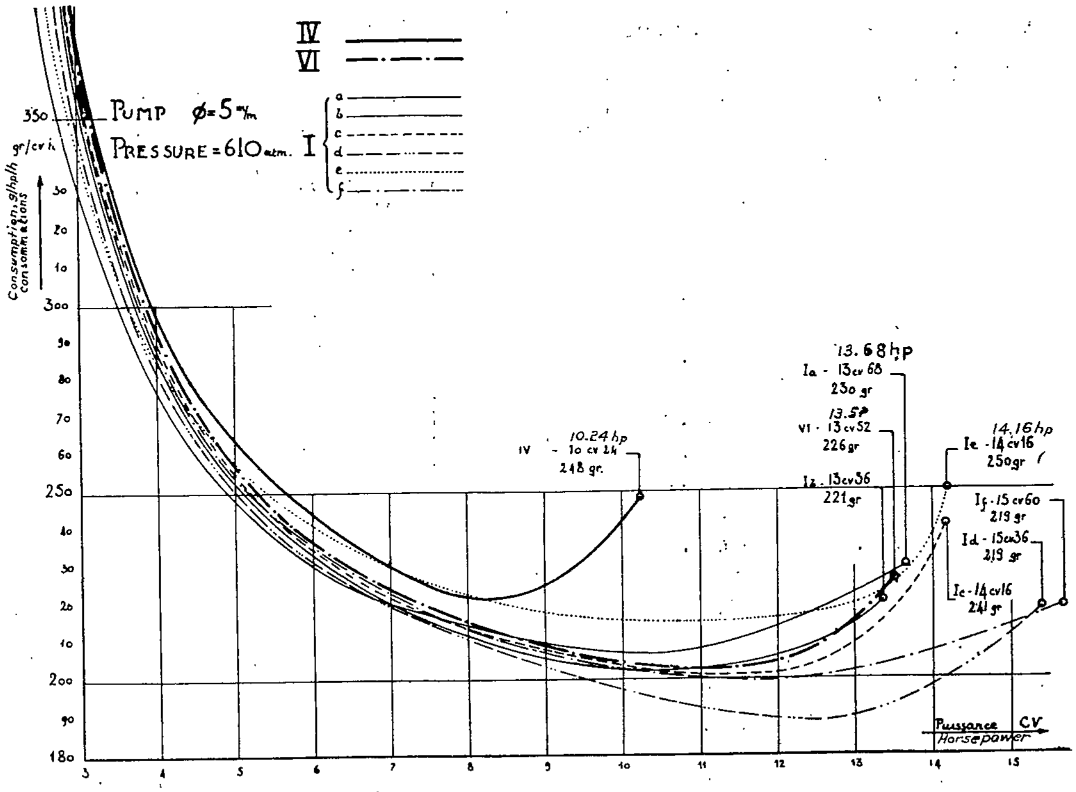


Figure 7.- Power and consumption curves of Winterthur-60 atm. injection pressure.

No.	Injection pressure	Needle opening pressure	Nozzles			Injection period
			Number	Diameter	Section	
	p_i, atm	p_i, atm		mm	mm²	s
IV	112	175	4	0,20	0,1256	0,00406
VI	400	240	4	0,20	0,1256	0,00257
I-a	610	120	4	0,20	0,1256	0,00215
I-b	610	305	4	0,20	0,1256	0,00215
I-c	610	120	4	0,30	0,2827	0,00096
I-d	610	120	6	0,25	0,2945	0,00092
I-e	610	120	10	0,20	0,3142	0,00086
I-f	610	120	6	0,30	0,4240	0,00064

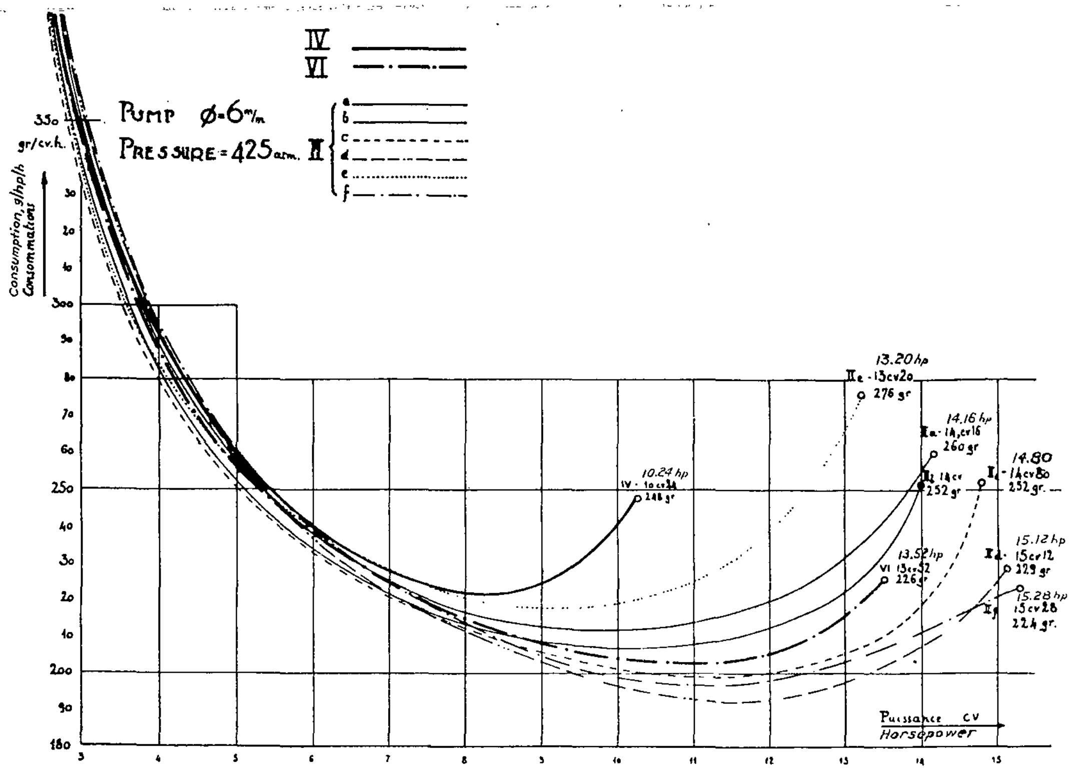


Figure 8.- Power and consumption curves of Winterthur-425 atm injection pressure.

No.	Injection pressure	Needle opening pressure	Nozzles			Injection period
			Number	Diameter	Section	
	pi, atm	pi, atm		mm	mm²	s
IV	112	175	4	0,20	0,1256	0,00406
VI	400	240	4	0,20	0,1256	0,00257
II-a	425	120	4	0,20	0,1256	0,00283
II-b	425	305	4	0,20	0,1256	0,00283
II-c	425	120	4	0,30	0,2827	0,00126
II-d	425	120	6	0,25	0,2945	0,00121
II-e	425	120	10	0,20	0,3142	0,00113
II-f	425	120	6	0,30	0,4240	0,00084

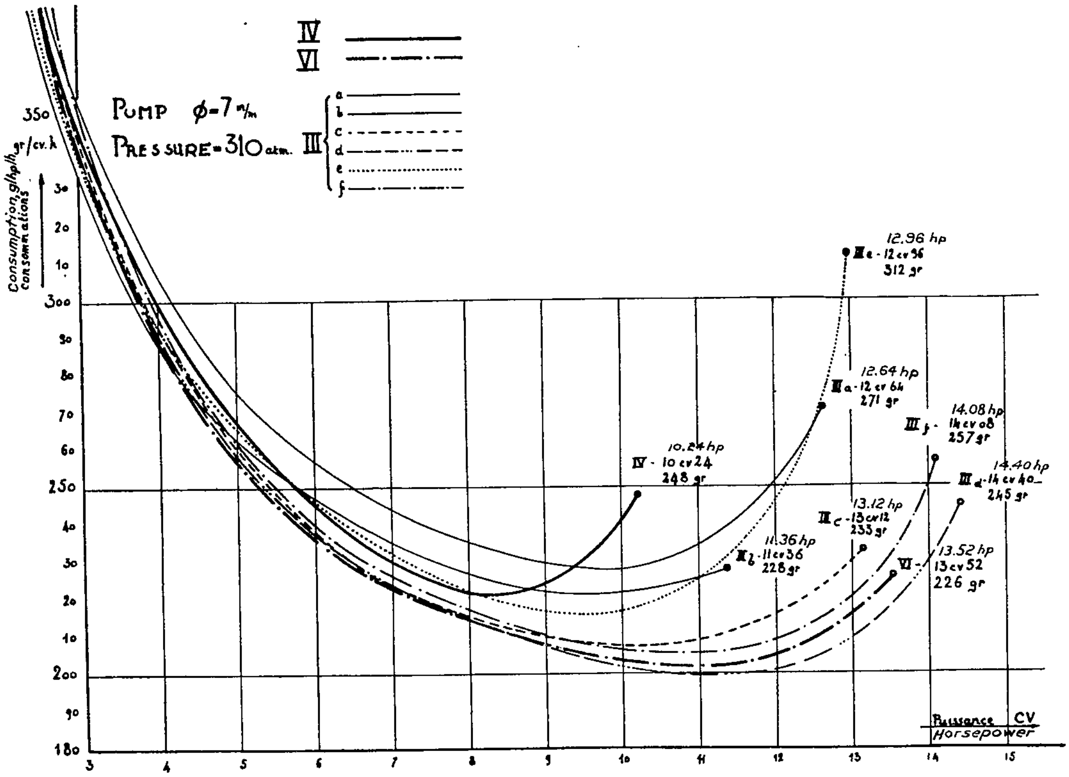


Figure 9.- Power and consumption curves of Winterthur-310 atm injection pressure.

No.	Injection pressure	Needle opening pressure	Nozzles			Injection period
			Number	Diameter	Section	
	p_i, atm	p_i, atm		mm	mm^2	s
IV	112	175	4	0,20	0,1256	0,00406
VI	400	240	4	0,20	0,1256	0,00257
III-a	310	120	4	0,20	0,1256	0,00308
III-b	310	305	4	0,20	0,1256	0,00308
III-c	310	120	4	0,30	0,2827	0,00136
III-d	310	120	6	0,25	0,2945	0,00131
III-e	310	120	10	0,20	0,3142	0,00123
III-f	310	120	6	0,30	0,4240	0,00091

Chapter 3

ANALYSIS OF TEST-STAND DATA

Effects of Pressure, Speed, and Distribution of Injection on the Engine Efficiency

Examination of the curves in figures 7, 8, and 9 affords the following conclusions:

a) For an identical injection pressure, the specific consumption decreases while the power which the engine can develop, increases in relation to the rapidity of injection.

For instance:

Curve I-a...230 g/hp/h at 13.68 hp...duration t=0.00215 s

Curve I-f...219 g/hp/h at 15.60 hp...duration t=0.00064 s

b) Terming the power corresponding to minimum specific consumption, the economical power of the engine, it is seen that this economical power rises in relation to the speed and to the injection pressure.

For instance:

Curve I-f (p_{i_2} = 610 atm, t = 0.00064 s) indicates a minimum consumption of 202 g/hp/h for a developed power of 12.88 hp, while curve IIIa (p_{i_2} = 310 atm, t = 0.00308 s) shows a minimum consumption of 230 g/hp/h at 10.80 hp. Curve I-f is very flat and the consumption may vary between 7 and 16 horsepower and, unlike curve III-a, which rises rapidly above 10 horsepower, and even less for the calibration curve IV (p_{i_2} = 112 atm, t = 0.00406 s), which already rises at 8 horsepower.

c) In the direct injection engine, without turbulence, dealt with here, the choice of nozzles and the distribution of jets in the combustion chamber is not unimportant. In fact, the curves e (S_7 = 10 orifices of 0.20 mm) show a characteristic anomaly from the point of view of specific consumption, as of the horsepower developed. The strainer with 10 holes of 0.20 mm, is poorly suited.

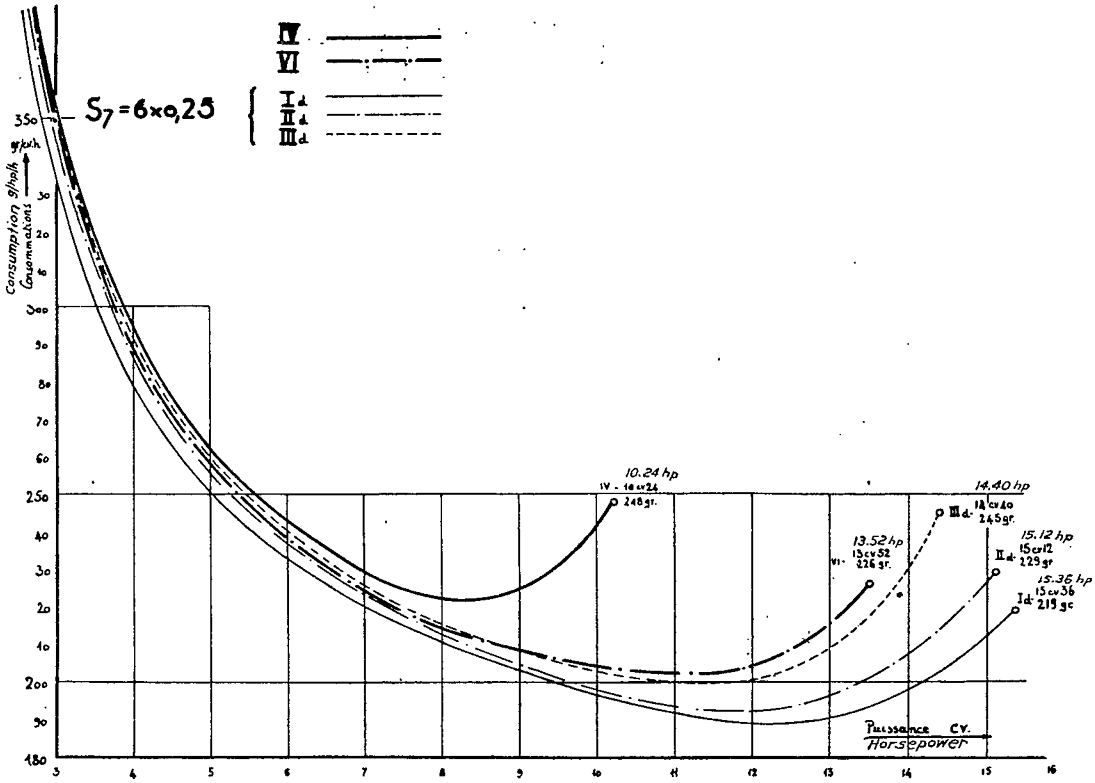


Figure 10.- Power and consumption curves of Winterthur injector d adequate.

No.	Injection pressure	Needle opening pressure	Nozzles			Injection period
			Number	Diameter	Section	
	pi, atm	pi, atm		mm	mm²	s
IV	112	175	4	0,20	0,1256	0,00406
VI	400	240	4	0,20	0,1256	0,00257
I-d	610	120	6	0,25	0,2945	0,00092
II-d	425	120	6	0,25	0,2945	0,00121
III-d	310	120	6	0,25	0,2945	0,00131

On the contrary, the three curves d (S_7 = 6 orifices of 0.25 mm) show that the six nozzles of 0.25 mm are apparently best suited to the Winterthur engine. Figure 10 illustrates these three curves d in comparison with the calibration curves IV and VI.

The recording of the consumptions at maximum power developed at 800 rpm, and the minimum consumptions at economical power, gives the following tabulation:

Consumption at Maximum and Minimum Power

Test No.	Pressure p_{i_2} atm	Duration milli-seconds	Consumption g/hp/h	Maximum horsepower	Consumption g/hp/h	Economical horsepower
I-a	610	2.15	230 at	13.68	208 at	10.40
I-b	610	2.15	221	13.36	203	11.04
I-c	610	.96	241	14.16	200	12.32
I-d	610	.92	219	15.36	189	12.80
I-e	610	.86	250	14.16	212	9.04
I-f	610	.64	219	15.60	202	12.88
II-a	425	2.83	260	14.16	209	10.80
II-b	425	2.83	252	14.00	207	10.40
II-c	425	1.26	252	14.80	199	11.32
II-d	425	1.21	229	15.12	192	11.60
II-e	425	1.13	259	13.92	218	8.64
II-f	425	.84	224	15.28	197	11.60
III-a	310	3.08	271	12.64	231	10.80
III-b	310	3.08	228	11.36	223	10.48
III-c	310	1.36	233	13.12	209	11.28
III-d	310	1.31	245	14.40	198	11.76
III-e	310	1.23	312	12.96	219	8.56
III-f	310	.91	257	14.08	207	11.28
IV	112	4.06	248	10.24	222	8.64
VI	400	2.57	226	13.52	202	11.52

Specific Power of an Injection Engine

<u>The specific power of a fuel-injection engine varies in the inverse sense of the duration of injection</u>.- For a specified injection pressure the maximum power which the engine can develop is obtained when the total charge of fuel is introduced within the shortest time.

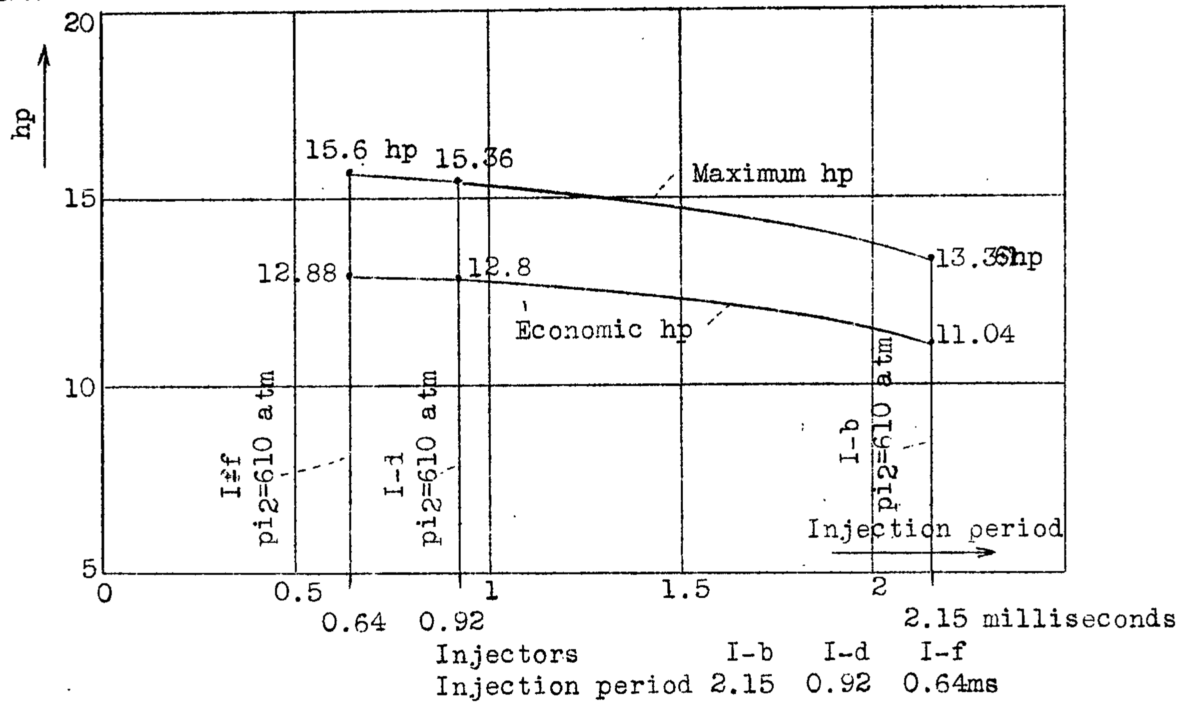


Figure 11.- Maximum and economic power of Winterthur-610 atm injection pressure.

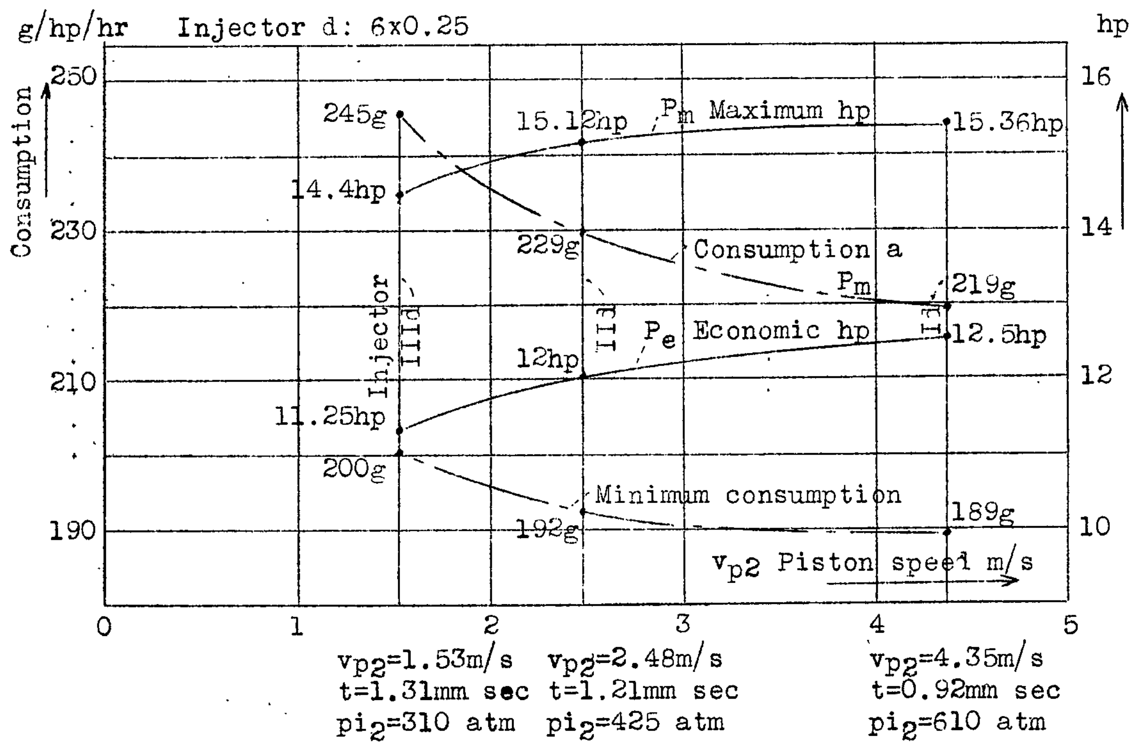


Figure 12.- Power and consumption against injection pressure. Injector d; 6 nozzles 0.25 mm in diameter; injection period t; 1.31, 1.21, 0.92 ms.

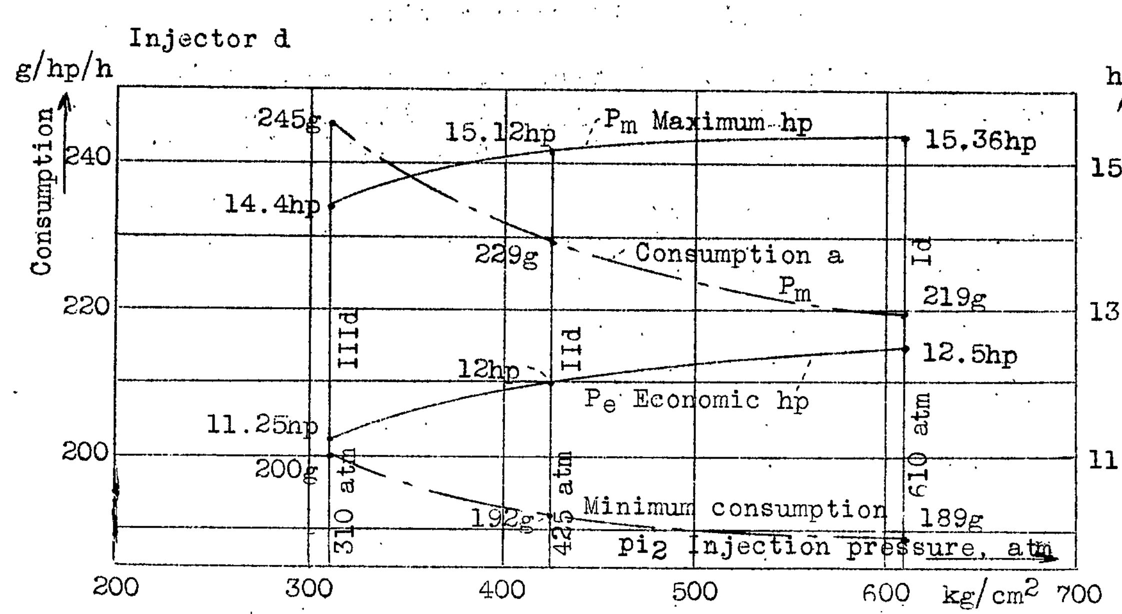


Figure 13.- Power and consumption against speed of injection. Injector d, 6 nozzles 0.25 mm diameter, pressure pi2, 310, 425, and 610 atm.

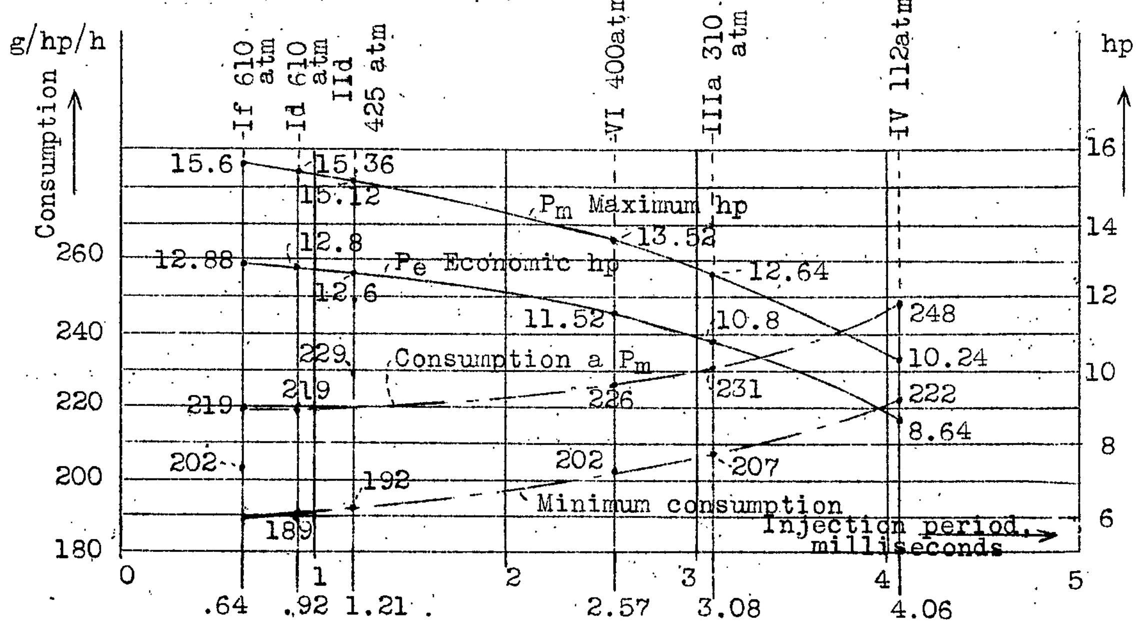


Figure 14.- Power and consumption against injection period. Test No. If, Id, IId, VI, IIIa and IV.

Specific Consumption

The specific consumption of a fuel-injection engine varies in the inverse sense of the injection pressure.- These two laws are related through the following:

The power and consumption operating conditions of a fuel-injection engine improve with a reduction of the duration and an increase in injection pressure.

Likewise, the economic horsepower of the engine increases in relation to the rate and pressure of injection. Reduction of the duration of injection is the chief factor for improving the power and for lowering the specific fuel consumption.

Ignition Lag and Duration of Injection

The duration of injection seems to be intimately related to that of the ignition lag. In fact, torque-stand experiments show that when increasing the volume of injection of any feed system, when the maximum power developed by the engine is reached, it is impossible to increase this power further, and the fuel introduced in excess is decomposed into aldehydes and tars without other result than thick black smoke in the exhaust and an exaggerated specific fuel consumption.

It is believed that, in order to obtain a new power gain, it is necessary either to raise the piston displacement or to improve the turbulence so as to assure a better mixing of the gas oil with the combustion air. Comparison of the torque-stand data proves it is otherwise when the displacement and the turbulence of the Winterthur engine remain equal for all curves, and that the power developed with the device No. I-f is nearly twice the rated engine power when fitted with its normal equipment No. IV.

A simple calculation shows that with No. I-f at 15.60 horsepower, the air-fuel ratio is 23.5/1, with a specific consumption of 219 g/hp/h, while with No. IV, the air-fuel ratio is 33/1 and the specific consumption, 248 g/hp/h.

Hence, with judiciously modified injection, it is possible to raise the power supplied by the engine and at the same time keep the consumption within tolerable limits. With turbulence, the engine manifested, of course, much lower air-fuel ratios.

Since the chief effective factor is the duration of injection, the plausible explanation of this statement is the following:

It is natural to suppose that the particles at the tip of a fuel jet first penetrating the working chamber heat and ignite first equally and form a flame front accompanied by zones of more or less impermeable instantaneous positive pressures in the jets of the fuel nozzles. Ignition being primed, the flame fronts expand and travel very rapidly in the working chamber and, if the ignition has not been terminated when they arrive at the injector nozzles, the fuel introduced later loses part of its penetration, stops burning, or burns very poorly without finding the air necessary, and in no way increases the power developed by the engine.

Whereas, if the total charge of fuel is inducted and distributed properly in the working chamber of the engine before ignition takes place - that is, to say, during the very short period of ignition lag, the penetration of the spray is not disturbed and the maximum power can be drawn from the engine with the minimum air-fuel ratio.

To be sure, it is no less true that a well-conditioned turbulence in the combustion air is the easy way to lower the fuel-air ratio. By reducing the duration of injection and by raising the turbulence, there is nothing to prevent, a priori, the Diesel engine from supplying a specific horsepower comparable with (if not superior to, at equal speed), that of carburetor engines.

The numerous studies made for measuring the ignition lag have shown this lag to depend upon several factors, the chief of which are: the compression ratio, the rate of rotation, the turbulence, and the shape of the combustion chamber. These measures, in general, given in degrees of crankshaft rotation, show that the ignition lag decreases when injection starts about 10° B.T.C. They similarly indicate a substantial reduction in ignition lag by a vigorous turbulence in the engine chamber.

The average ignition lag recorded is from 5° to 12° crankshaft, when the engine speed with vigorous turbulence passes from 500 to 2000 rpm, for a compression ratio of 14.

Now, the angle of injection of a mechanically controlled pump is always greater than 5°, even than 12°. The active part of a cam of the pump records, in general, 20° to 25° crankshaft, in such a way that thus far the total charge of fuel is inducted within a period exceeding that of the ignition lag considerably.

That, we believe - and, as borne out by curves 7 to 14 - is a sufficient reason why the Diesel cycle has required up to now a higher air-fuel ratio than a gas or gasoline engine. The ignition lag measured in degrees of crankshaft when translated into time, gives 0.0017 second at 500 rpm, and 0.001 second at 2000 rpm.

By utilizing the compression for actuating the pump injector, the total fuel charge can be inducted into the engine chamber within the ignition lag period, as shown by I-f, for example, for which the time of injection is 0.00064 second.

Penetration, Dispersion, Distribution, and Pressure of Injection

At the exit of the injector nozzles the pressure of the fuel is transformed into speed. The fuel sprays in the working chamber at the instant of ignition occupy a well-defined position, which can be exactly determined, knowing the duration of the flow, if the speed of the finely atomized particles follow a simple law of deceleration.

The research carried out in various countries to determine the law of penetration of fuel sprays in compressed air, show that the penetration is in function of the injection pressure, but that the resistance of the compressed air undergoes at speed v_i of the sprays at the nozzle exit, a deceleration which is so much faster as the pressure of injection is higher. This should not be surprising since the fineness of the atomized particles is proportional to the injection pressure, so that in the kinetic energy $mv^2/2$ of each droplet, the mass decreases when the injection pressure increases, while the speed v rises, as seen, as function of $\sqrt{p_i}$.

The penetration measurements have been made in containers of cold air or a neutral gas whose density corresponds to that of the working chamber of a Diesel engine at 14 compression ratio. But there is no motion of the gas itself in the container, no disturbance of pressure as set up by the flame front in the chamber of an engine. It is doubtful that such measures correspond exactly with what actually transpires by injection into a combustion chamber with high turbulence - all the more so as the penetration is proportional, not merely to the injection pressure, but is also dependent upon other factors: It grows with the nozzle length to diameter ratio - less dispersion - viscosity of the fuel - larger mass of droplets; and it decreases as the compression increases - stronger resistance of the air. It is difficult to ascertain the exact distance from the tip of the spray to the moment of ignition.

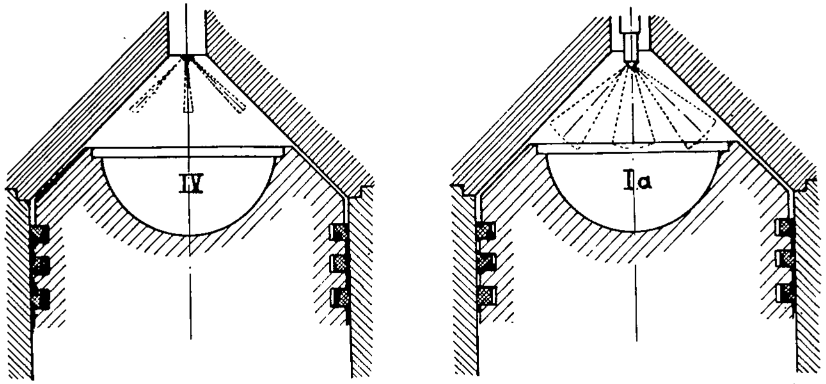
Other than the duration and the penetration, the dispersion likewise determines the quality of a combustion.

Indeed, at the high pressures employed in direct injection, the flow in the spray nozzles is always turbulent and the sprays take the form of cones whose opening angle at the nozzle increases with the injection pressure and decreases in proportion to the nozzle length.

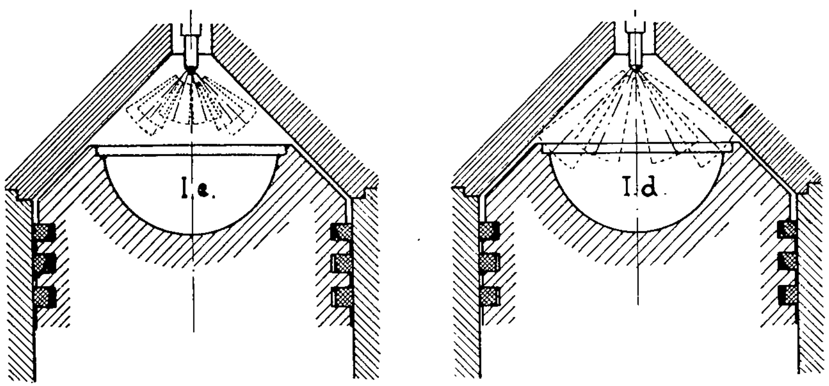
Inducting the entire fuel charge within the ignition lag time interval through nozzles in such number and dimensions that penetration, distribution, dispersion, and fineness of particles permit the injected fuel intimately to affect the entire volume of the compression chamber, the combustion will take place under conditions enabling the engine to develop its maximum power with a minimum fuel consumption. The specific consumptions themselves will be minimum at all charges and speeds of operation of the engine.

There is one well-defined distribution and one injection pressure suitable to the dimensions of a nonturbulent combustion chamber. If the number of nozzles is insufficient - curves I, II, III-a, III-b, IV, and VI - four holes of 0.20 mm diameter - the distribution of the fuel is not sufficient. If the nozzles are too numerous, the injection lacks penetration - curves I, II, III-e - six holes of 0.20 mm diameter.

Likewise, if the pressure is low, the penetration, dispersion, and fineness of particles are not sufficient, the atomization cones fill only part of the combustion chamber, with consequently mediocre power and specific consumption - curves IV, III-a,b,c, etc.



IV.-pi_2 =112atm; S_7:0.1256mm^2-4×0.20; t:0.00406s, P_m: 10.24hp-248g/hp/h. Pressure and distribution to weak-injection period to long.
Ia.-pi_2= 610atm; S_7:0.1256mm^2-4×0.20, t:0.002 15 s, P_m: 13.68 hp-230g/hp/h. Distribution to weak-injection period to long.



Ie.-pi_2= 610atm; S_7:0.3142mm^2-10×0.20, t:0.00086 s, P_m: 14.16 hp-250g/hp/h. Distribution excessive-lack of penetration.
Id.-pi_2= 610atm; S_7:0.2945mm^2-6×0.25, t:0.00092 s, P_m: 15.36hp - 219g/hp/h. Period, pressure, and distribution of spray well adapted.

Figure 15.

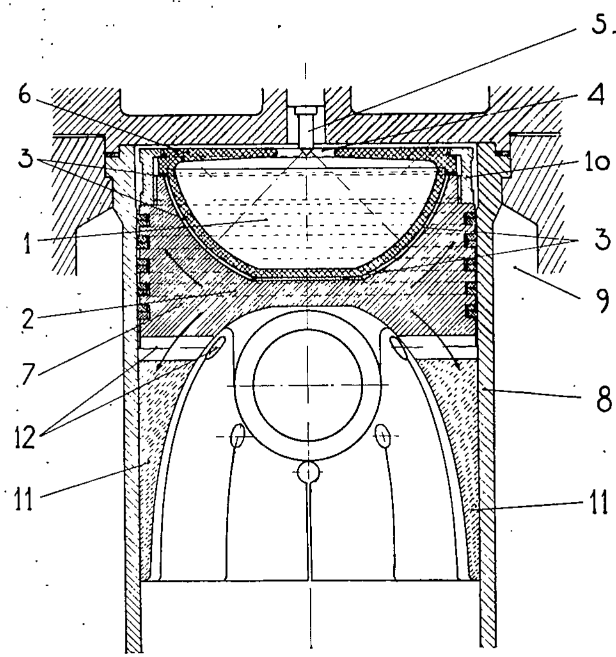
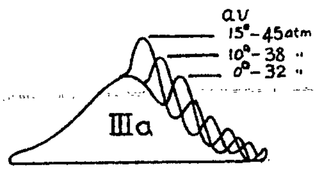


Figure 17. Combustion chamber and piston-cylinder system of a direct injection engine.



IIIa.-pi_2=310 atm;4 nozzles of 0.20mm dia.
IId.-pi_2=425 atm; 6 nozzles of 0.25mm dia.

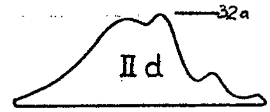
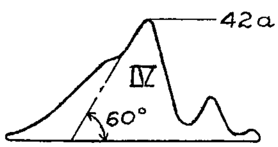


Figure 16a.-IIIa, idling; advance of ignition regulated by lowering the tare of the return spring A; peak pressure of cycle rises with injection advance.

Figure 16a.- IIa, idling;advance of injection:10°. A constant pressure cycle is readily obtainable at idling and low charge.



IV.-pi_2=112 atm;4 nozzles of 0.20mm dia.
VI.-pi_2=400 atm;4 nozzles of 0.20mm dia.

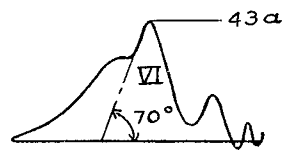
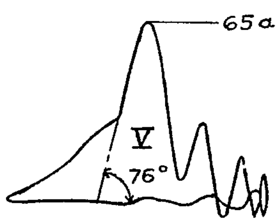


Figure 16b.-IV, 2.48 hp; advance:15°,angle of explosion crater:60°,peak pressure of cycle:42 atm;low injection pressure compensated by greater advance

Figure 16b.-VI, 0.48 hp; advance: 10°, angle of explosion crater: 70°, peak pressure of cycle; 43 atm, injection pressure compensated by greater injection lag. As injection pressure increases, the cycle approaches combustion at constant volume.



V.-pi_2=400 atm;4 nozzles of 0.20mm dia.
VI.-pi_2= 400 atm;4 nozzles of 0.20mm dia.

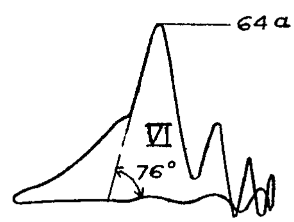
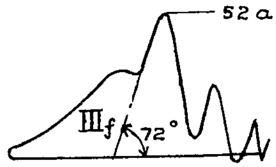


Figure 16c.-V, injection pressure at lifting of needle:pi_1= 120 atm., 10.80 hp; advence: 10°,angle of explosion crater 76°,peak pressure of cycle 65 atm;peak pressure of cycle rises with the charge.

Figure 16c.-VI, injection pressure at lifting of needle:pi_1= 240 atm; 10.64 hp; advance: 10°,angle of explosion crater 76°,peak pressure of cycle 64 atm;compared with the one above,it is seen that the choice of pi_1 has not changed the cycle.



IIIf.-pi_2 =310 atm; 6 nozzles of 0.30mm dia.
Id.-pi_2 =610 atm; 6 nozzles of 0.30mm dia.

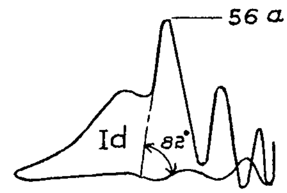
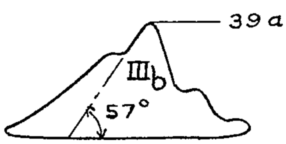


Figure 16d.-IIIf, 13.36 hp,advance: 5°,angle of explosion crater: 72°, peak pressure of cycle: 52 atm.

Figure 16d.- Id, 13.76 hp,advance: 1°,angle of explosion crater: 82°, peak pressure of cycle: 56 atm, the explosion crater is more active as pi_1 increases.The engine runs smoother at low injection pressure,but the efficiency is inferior.



IIIb.-pi_2=310 atm;advance: 10°; 4 nozzles of 0.20mm dia.
IIId.-pi_2=310 atm; 6 nozzles of 0.25mm dia.

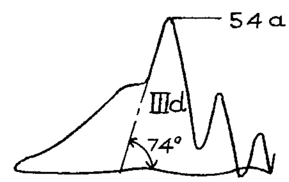


Figure 16e.- IIIb,injection period:0.00308 s; 11.36 hp,advance: 10°; angle of explosion crater;57°,peak pressure of cycle;39 atm.

Figure 16e.- IIId,injection period;0.00131 s; 11.76 hp,advance: 5°; angle of explosion crater;74°,peak pressure of cycle: 54 atm, for an identical injection pressure and engine charge the engine runs more active if the injection period diminishes and the efficiency is improved.

Contrariwise, if the pressure is too high, the tips of the spray reach the hot walls of the piston and produce overheated spots, or reach the cold walls of the cylinder - in which case they do not burn but dilute the lubricating oil and form carbon on the top piston rings.

Figure 15 illustrates, for four experimental devices, the respective position of the sprays in the combustion chamber of the Winterthur engine at the moment of ignition. The distance between the nozzles and the bottom of the piston at dead center is about 50 mm:

No. IV - p_{i_2} = 112 atm, S_7 = 4 nozzles of 0.20 mm diameter, t = 0.00406 sec.

Curve IV shows the distribution and the pressure of the fuel to be too low, and the duration of injection too long to enable the sprays to cover a distance of 50 mm during the ignition lag. Only a small portion of the charge is inducted during this lag. The fuel affects only a small portion of the compression air and its atomization in coarse particles does not enhance the power nor the specific consumption.

No. I-a - p_{i_2} = 610 atm, S_7 = 4 nozzles of 0.20 mm diameter, t = 0.00215 sec.

Here the fuel distributed by the four nozzles is likewise insufficient. Although the pressure of 610 atm assures adequate penetration and dispersion, the duration of injection does not permit induction of the total charge during the ignition lag. The torque-stand measurements are improved.

No. I-e - p_{i_2} = 610 atm, S_7 = 10 nozzles of 0.20 mm diameter, t = 0.00086 sec.

The charge is distributed by a large number of nozzles and, in spite of adequate pressure and a duration of injection shorter than the ignition lag, the fuel lacks penetration and fails to fill the working chamber. The engine runs hard, there are violent explosions, with power output and fuel consumption very mediocre.

No. I-d - p_{i_2} = 610 atm, S_7 = 6 nozzles of 0.25 mm diameter, t = 0.00092 sec.

The conditions necessary for smooth operation are satisfied. Distribution, penetration, dispersion, and duration of injection permit the fuel to mix well with the compression air before ignition takes place. The specific fuel consumptions are very low and the rated 8-horsepower output is almost doubled.

At 12 horsepower, the engine has a 34-percent effective efficiency, the thermic efficiency of the cycle being around 45 percent. These results are remarkable, considering that it involves a two-stroke-cycle, single cylinder with $1\frac{1}{2}$-liter displacement.

Injection Advance and Combustion Cycle

As everybody knows, the increase in injection advance produces a cycle which more and more approaches the cycle with constant volume, and the peak of the maximum pressure is a function of the degree of advance.

The diagrams recorded by Lehmann and Michels indicator for each measurement of these comparative tests, clearly show:

1. That the pressure of the cycle rises with the engine charge.

2. That the craters of explosions are so much more active as the injection pressure is intense.

3. That the pressure of the cycle likewise increases with the speed of injection.

The adjustment of the ignition point is assured in such a manner that the peak pressure of the cycle never exceeds 60 atmospheres.

Figures 16 a, b, c, d, e, illustrate some of the most typical records. The oscillations of the expansion curve are due, of course, to the indicator inertia.

On these diagrams, displaced by 90° on the engine crankshaft, 3 mm abscissas, in direct proximity of T.C., represent 7° 30' crankshaft, or 0.0015 second, making the ignition lag of the Winterthur engine, represented by about 4 mm, 10°, or 0.002 second.

These experiments show that the injection lag should be regulated in relation to the engine charge, so that the peak pressure of the cycle has no substantial rise with the charge.

Idling excepted, for which the specific consumption is improved by an advance of several degrees, the injection should be placed per horsepower on top center, so that the combustion in its entirety takes place during the expansion stroke of the piston. For pump injectors controlled by the compression, the injection-lag limit is, of course, the top center itself.

Under these conditions of timing, and provided the injection pressure is not very high, the peak of the cycle rises slowly with the charge, the engine revolves without any roughness, and the diagrams widen out with the charge, similar to those of a steam cylinder when the intake valve is opened wider. When supercharged, the pressure peak rises, but the engine has no tendency to knock, and sustains this regime without weakness or fatigue.

These experiments show that rapid injection, divided over two or three degrees either side of top center, characterized by a delayed combustion taking place in its entirety during the expansion stroke of the piston, correspond to the best operating conditions of the engine, from the point of view of power developed as of specific fuel consumption.

This practical result may seem paradoxical in comparison with the entropy cards and the heat balance of a normal combustion cycle, without lowering of the compression A.T.C. (diagrams Nos. IV, V, and VI) and of a delayed combustion cycle (diagram Nos. I-f, III-b,d,f, for instance). The reason, it seems, must be looked for in the difficulties of obtaining a complete combustion in a Diesel burning heavy hydrocarbons rather than in thermodynamic laws, which prove the opposite.

In fact, the exhaust-gas analysis of a direct injection Diesel with pronounced pressure peak in combustion proves, if the charge becomes heavy, that the combustion is far from being complete. The smoke and disagreeable odor of the exhaust bear witness to this fact. For complete combustion, the rate of chemical reaction must be sufficient to be terminated before the exhaust opens. The greater the speed of reaction, the more complete the combustion, the better is the power output and the specific consumption. The chemical reaction of the combustion analyzed by calorimeter, does not correspond to the same reaction in the chamber of an engine whose volume is continuously changing.

But, if based on the fundamental laws of chemical reactions which define the variations of the constant of equilibrium, the conditions necessary to accelerate the speed of reaction of a combustion in the calorimeter with variable volume, which represents the working chamber of an engine, can be easily deduced.

This speed of reaction depends on three principal factors, as follows: the concentration of the mixture, the variation of the pressure, and the variation of the temperature.

Qualitatively, Le Chatelier's principle shows that:

"Any change in one of the factors of the equilibrium results in a transformation of the system in such a way that the factor undergoes an inverse change."

Therefore:

1. If the concentration of one of the reaction gases is increased, the impulse is given to a reaction associated with the disappearance of this gas. If the concentration is reduced, the reaction gives rise to the release of this gas.

2. If the pressure is raised, the impulse given to the reaction is tied to a reduction in the number of molecules. If the pressure is lowered, the impulse is given to an increase in the number of molecules.

Lastly, if the temperature of a chemical reaction is raised, an impulse is given to an endothermic reaction and, if lowered, to an exothermic reaction. The speed of reaction toward the new equilibrium increases as the change effected in one of these factors is greater.

Now, the combustion of any heavy hydrocarbon $C^n H^{2n+2}$ which can be burned in a Diesel engine is always accompanied by an increase in the number of molecules, hence accelerated by acting on the three factors of equilibrium - concentration, pressure, and temperature - in such a way that the reaction receives an energetic impulse in the same sense, namely,

1. Increase in concentration by an instantaneous solid injection (during the ignition lag period).

2. Decrease in pressure by an injection at top center, and a great linear piston speed which permits a rapid increase of the volume in the combustion chamber.

3. Decrease in temperature by evaporation of the solid injection, a good cooling of the engine, and a rapid expansion.

With these three conditions satisfied, a complete combustion can be obtained within the shortest time, as amply confirmed by torque-stand test.

The instantaneous induction of the total fuel charge in the working chamber, which corresponds to the increase in concentration of gas mixture, is generally disclosed as being the principal factor acting on the speed of reaction. This condition is, moreover, tied to two others by the induction of fuel in direct proximity of top center, so that the reaction commences and continues during the expansion stroke of the engine. Thus, the volume and the cooling surface of the chamber increase constantly during combustion.

Combustion Chamber and Turbulence

From this analysis of the injection and the ensuing combustion, it is logical to deduce the shape and size best suited for the combustion chamber.

In brief, we have tested and verified that it is necessary to carefully blend the fuel with the compression air before ignition, after dead center. The complete charge should be inducted and distributed in a chamber containing the entire combustion air within the ignition-lag period, and this air animated by a strong turbulence in order to activate its mixture with the fuel. Thus, it seems to us that only direct injection makes it possible to realize the conditions necessary to obtain the best efficiency of a Diesel engine. In fact, direct injection makes it possible to give the combustion chamber a compact shape wherein the whole cylinder charge is collected - the volume of this chamber presenting the surface of minimum exchange of heat.

The other systems with a precombustion or an auxiliary chamber involve a considerably larger wall surface in the working chamber - so much, that some engines cannot start from cold without resorting to a primer. During operation, the walls of the auxiliary chambers become very hot and the turbulence is enormous, so that these systems have offered, up to now, the advantage over direct injection of atomization at low pressure (100 to 120 atm) by an injector, whose operation is less ticklish than that of injectors with several orifices of very small diameter.

By way of compensation such auxiliary chambers are, in general, lodged in the cylinder head, and their thermic effect on the intake air lowers substantially the volumetric efficiency of the cylinder charge - and so much more as the charge and the speed of the engine are greater. For these two reasons - greater area of exchange of heat and lower volumetric efficiency - the injection systems with auxiliary chamber cannot give to the engine its best efficiency.

Figure 17 illustrates one method of obtaining a combustion chamber with direct injection, corresponding to the conditions defined previously.

The compressed air is collected in a stainless steel dome 1, situated in a hemispherical recess of piston 2. This dome holds practically all the combustion air. It is separated from the piston mass by a thermic insulation 3, and is free to expand without causing strains on piston 2. It opens into the engine chamber through a more or less large orifice 4, depending upon the degree of turbulence desired. This orifice is in alinement with injector 5, which distributes and atomizes the fuel. No projection of the fuel injected in the dome on the cylinder wall is produced.

The isolated dome forms a space for storing the heat during combustion for the purpose of restoring it to the air during the compression cycle. This heat chamber is regulated by varying the mass and by the insulation of the dome.

The engine chamber permits the smallest surface of exchange of heat, hence the wall losses are lowered, the volumetric efficiency improved, and starting from cold is assured without the use of a primer. According to our experiments, the running conditions of the engine, the power output, and the fuel consumption are so much better as the injection period is shorter and the injection pressure higher; but, as stated before, these conditions unfortunately correspond to a combustion cycle with very active craters of explosion.

The multiple action of the dome affords a greater freedom in the choice of speeds and injection pressure, assuring smooth running under the conditions of good engine efficiency.

Indeed, the turbulence and the high temperature in the dome shorten the ignition-lag period but permit, at the same time, ignition to follow after ignition, without which the combustion of the charge would be less complete. An appropriate control of the injection could very probably supply a progressive cycle of combustion, without craters of explosions, giving the engine its best efficiency with a very smooth run.

To smoothen the run, the turbulence could be reduced by increasing the section of orifice 4 or drilling other holes - six, suitably distributed over the cover of the dome.

Idling, Pick-up, and Injection Period, Independent of the Engine Speed

In all the injector-pump tests, the idling of the Winterthur engine was controlled at 200 rpm, while with the Bosch equipment, idling cannot drop below 350-400 rpm, and the pick-up is less vigorous.

Chapter 4

OPERATING CONDITIONS OF INJECTOR PUMPS ACTUATED BY ENGINE COMPRESSION

Elastic Balance of Pump Motion

The abrupt pressure changes in the pump chamber and the percussion shocks at the end of the stroke of the moving part, impose two very strict conditions for assembling this equipment:

1. The pump chamber, the injector, and the moving part of the injector pump should have no joints, no threaded parts, if dislocation and unpriming of the pump is to be avoided; the exception is the atomizer, which can be carried on a shoulder of the pump, where it is maintained by the pressure of the fuel.

2. The motion of the pump should be elastically balanced at the end of the stroke, in order to avoid inopportune opening of the needle as a result of the exchange of speeds after percussion.

The elastic balance of the motion is assured by the damper B, which allots the pump a stroke of several hundredths of a millimeter at the end of the injection (fig. 2). If the stop of the motion is not elastic, particles of compressed gas will succeed in penetrating through the injector line into the pump chamber where, if unpriming is not effected, it will form an emulsion with the fuel, causing at the nozzle outlet a sudden expansion of this gas, which causes the jets to burst into very fine particles which quickly lose their penetration, no matter what the injection pressure utilized. The evaporating surface of these fine droplets grows enormously, shortening the ignition-lag period, and disclosing very violent craters of explosions, on the indicator cards.

On the contrary, if the motion of the pump is elastically balanced, the closing of the needle is perfect and no air particle enters the pump line. The jets do not burst at the nozzle outlets, retain their penetration much longer, and produce a less violent combustion without the characteristic explosive wave observed in the injection by mechanically controlled pumps. Indeed, the sudden pressure fluctuations in the line connecting the pump to the injector give rise to shock waves which, if the residuary pressure is high, determine by closing, the well-known secondary injections which often are accompanied by the penetration of compressed gas in the injector.

Control of Feed and Injection Point

These can be controlled jointly or separately (figs. 1 and 2). If the injection-point control is automatic, and it is desired to obtain a combustion cycle without appreciable rise of peak pressure with the charge, the injection lag, as has been stated, should be a function of the injection feed. In this case, the threads of the feed stop 3 and of stop 4 of spring A are in the converse sense, and the pitch chosen for these two threads determine the amount of the retarded injection as function of the engine charge. The movement of the lever of stop 3 engaged in a vertical cavity of feed stop 4, suffices then to assure simultaneous control of the point and the feed of injection. This maneuver demands no effort on the part of the regulator.

In order to be able to utilize the speed, and hence the limit pressure of the pump up to the closing of the injection needle, it is necessary to regulate the lift of damper B over a very short distance. A few hundredths of a millimeter are enough to make the control of the atomizing nozzles of the needle perfect.

Pump Chamber

The chamber of pump 9 below the plunger 8, is condensed in a few millimeters on the end of the bore of the sleeve of pump 7 which penetrates the working chamber of the engine (fig. 3). The injection needle 11 with positive rise is in the axis of the pump where it occupies a perfectly symmetrical position in the sleeve 10 of the injector. The inlet valve 12 obturates the pump chamber upwardly.

The pressure waves at closing of the inlet, enter in resonance between the bottom of the pump chamber and the base of the fixed plunger. The speed of these waves is sonic in the medium where they are propagated. This speed, being of the order of 1500 m/s in hydrocarbons, the frequency of the waves in the pump chamber is 150,000 per second for a chamber 5 mm high, or 150 periods during a 0.001-second injection period. The resonance is so high as to no longer affect the lift or the closing of the needle. Control of intake and delivery of the pump are therefore definitely assured.

The tightness of the pump chamber and the perfect control of intake and delivery have, certainly, a capital effect on the injection period. The solution of the device with positive opening of needle offers much greater security than the solution with negative differential needle, guided in the bore of the fixed plunger and controlling the nozzles drilled in the bottom of the pump chamber.

The escape of fuel along plunger 8 is returned to the hollow of inlet 13 by a set of labyrinth grooves 15 and suitably disposed openings 16, which eliminates the leakage return pipes.

The displacement of the atomized sprays during the stroke of the pump is not particularly favorable because it interferes with good penetration and necessitates probably higher injection pressures than if the nozzles maintained a fixed position. As compensation, the abrupt stop of the mechanism at the end of injection avoids radically the formation of soot craters at the nozzle outlets.

Exchange of Heat in the Pump Chamber

If the section $S_1 - S_6$ of the compression air in gas cylinder 2 is very small, the speed of entry of the hot gases below piston 14 is high and the exchange of heat with the pump chamber is then considerable; so much so, that pump sleeve 7 becomes blue following a ring at the height of section $S_1 - S_6$.

To avoid this heating which, without impeding the operation, may appear excessive, it suffices to increase the diameter of the section of passage of the gases in the cylinder a few tenths of a millimeter. This removes every trace of heat, which proves that the exchange of heat with the walls of a flow is a function of the speed of the fluid and that it operates by molecular contact; i.e., by conduction and not by radiation.

The cooling of the walls of the pump and of the injector by the fuel is so active, that placement of the pump in the exact center of the firing chamber of the engine produces no danger but permits injection of the previously heated fuel and use of heavy fuels without fear of clogging the nozzles.

Piston and Gas Cylinder

In order to obtain a stable and lasting operation of the pump injectors, three conditions must be rigorously observed in the disposition of the piston:

1. The piston and, consequently, the piston rings, must not be subjected to vibrations and the rings must form a perfect seal.

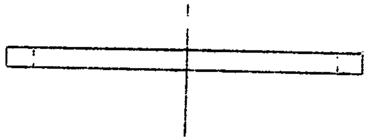


Figure 18.- Piston ring before slotting.

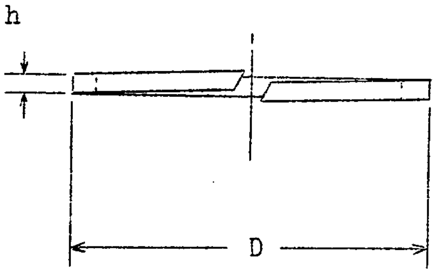


Figure 19.- Piston ring after slotting.

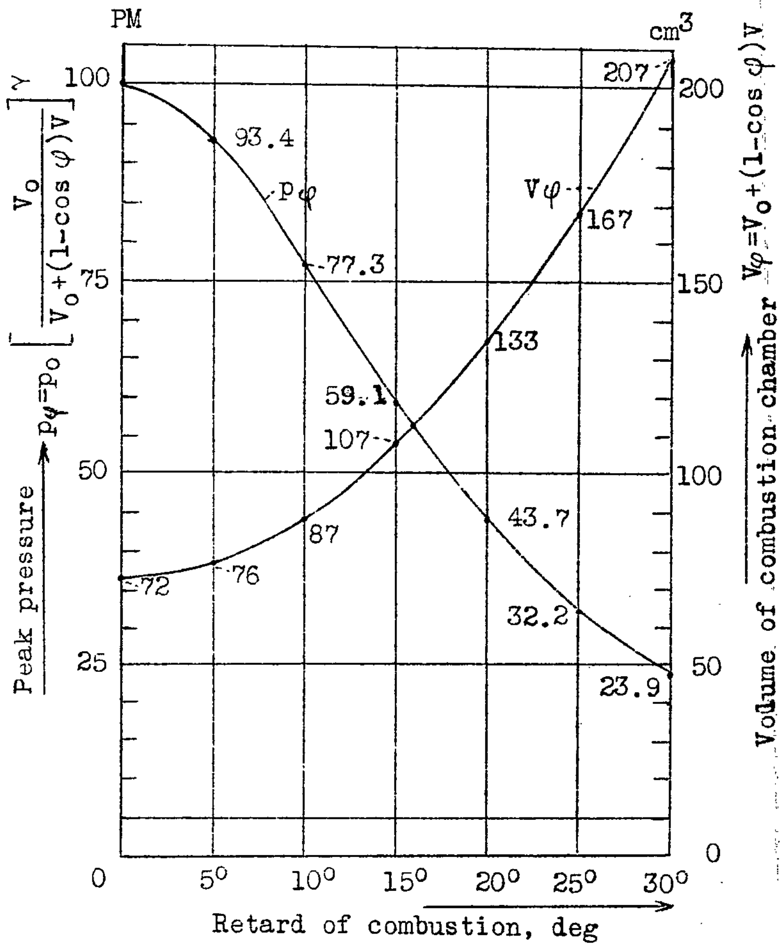


Figure 33.- Retarded combustion cycle.
Decrease in peak pressure and increased displacement in relation to delay in combustion.

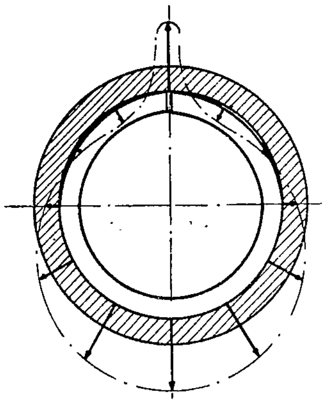


Figure 20.-Before

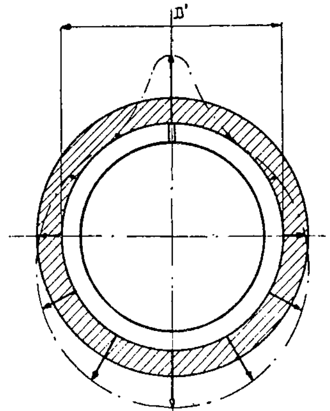


Figure 21.- After

Figures 20,21.- Stress distribution and ring contact with bore, before and after grinding.

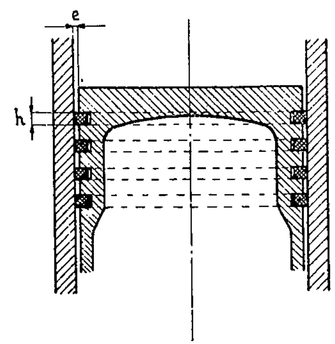


Figure 22.- Piston-cylinder system fitted with elastic piston rings.

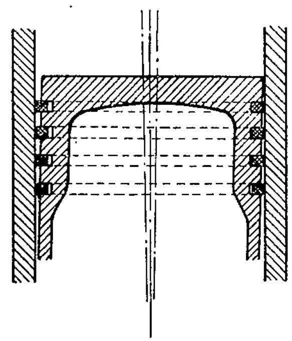


Figure 23.- Deformation of rings following piston knock

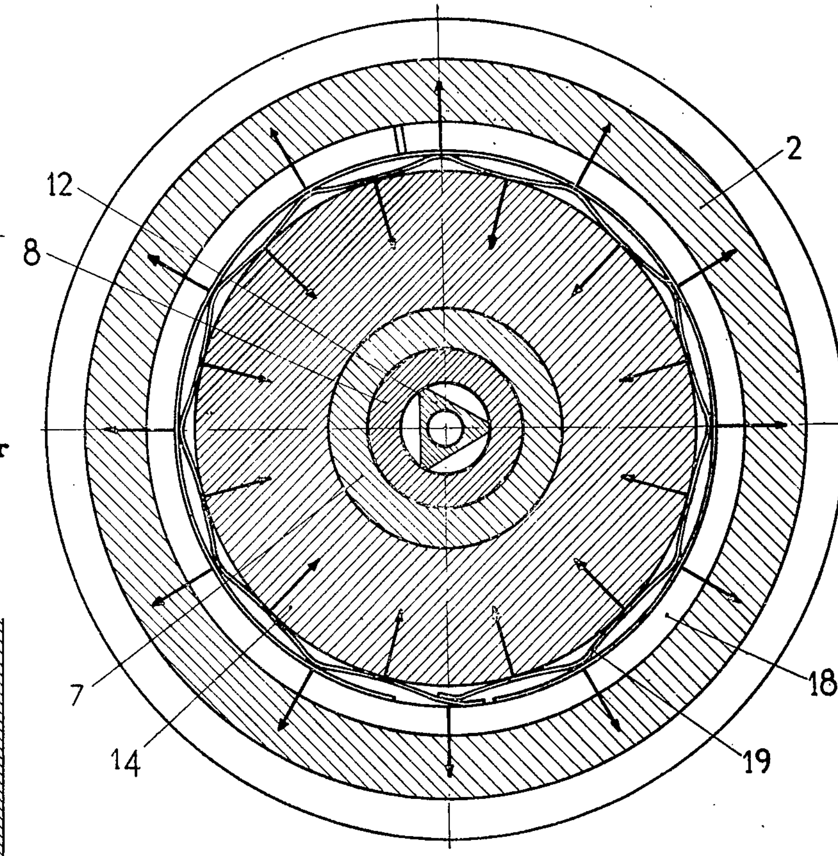


Figure 24.- Cross-section of piston at level of tight rings.
2, gas cylinder; 7, pump sleeve; 8, fixed plunger; 12, inlet valve; 14, gas piston; 18, tight rings; 19, tightening spring.

2. The piston rings should work at a relatively low temperature - at most, at 150° C, preferably less.

3. The cylinder bore must be lubricated.

A piston ring of small diameter, such as that of pump injectors, for example, of little height, machined from a cast-iron mass, is often deformed if the slot is made obliquely to the ring. It then becomes similar to a Grower washer; its permanent deformation no longer permits correct straightening of the face with respect to the bore.

When the small diameter ring is inserted in the cylinder where it is to work, it makes contact with the two tips of the slot and on the opposite half circle. Two slits are visible at 30° and 45° in the other half circle. They become so much more accentuated as the tension of the ring is greater and are proportional to the difference in diameter D of the expanded ring and diameter D' of the cylinder, and to the inertia of the straight section of the piston ring.

It is therefore evident that the tension of the elastic ring on the bore of the cylinder is not evenly distributed over the circumference. The grinding of the ring - much faster at points of high tension, causes the two slits to disappear, little by little, but never produces a uniform tension over the circumference. Its tightness is not uniform, and so produces uneven wear on the bore of the cylinder.

A certain expansion clearance e is provided between piston and cylinder. The piston rings, of themselves elastic, exert no centering action on the piston, with the result that if the latter vibrates in consequence of active explosions in the combustion chamber, for instance, this vibration is transmitted to the piston rings.

The vibration of the piston rings and their deformation on the bore D' of the cylinder, is so much greater as the expansion clearance e and the height h of the rings is greater and the play of the rings in the piston grooves is less. This vibration induces a leakage of hot gas along the piston, and the temperature of the piston rings may rise to the point where it corresponds to the distillation of the oil, resulting in the well-known gumming in the grooves and coking.

These disadvantages are more serious in the Diesel than in the internal-combustion engine. The much higher mean temperature of the cycle and the residues of incomplete combustion of gas or fuel oil employed form at the sides of the piston rings, where they do not perfectly fit the bore, spots of carbon on the cylinder and the piston ring also. The coefficient of slippage of the rings in the bore increases enormously at the side of the spots; the mechanical efficiency and the engine output gradually drop in surprising proportions.

If the piston rings work at high temperature, the carbon deposits become very adhesive and hard, resulting in rapid cylinder wear. In pump injectors, controlled by compression, the carbon deposits on the gas cylinder and piston rings slow up the rate of injection, and thus change the running conditions of the engine.

To prevent piston-ring vibrations consequent to the percussions of the moving part at the end of the stroke, piston 14 is extended upward by elastic slippers 17, cut in a skirt (fig. 3). The end of the slippers is machined with a slight tension on the cylinder bore so that the piston, elastically centered, is free to expand but can no longer vibrate.

To assure perfect tightness, the elastic rings are replaced by split rings 18, whose outside diameter is machined at the cylinder side, without elastic tightening. Tension of the rings on the bore is assured by a set of springs 19, supported on the piston itself. The points of thrust of springs 19 on the rings and on the piston are numerous enough to assure evenly distributed tension and perfect centering of the piston.

To forestall deformation of rings 18 during mounting, they are carried on a shoulder of the piston and retained by a screw 20, forming a labyrinth (fig. 3). Three rings 18 with alternating slots are superposed in the same piston groove; the last toward the top is double, in order to nullify the slot leakage.

The heat flow of the gas on piston 14 is deflected through a large opening in the bottom of the piston toward the fuel prepared for by the inlet cavity 8 in pump chamber 9, and injector 10. The gas cylinder 2 is fitted with cooling fins immersed in the water chamber of the cylinder head.

The piston rings 18 situated in the top of the opening of piston 14, are protected by the labyrinth screw 20, and so operate at low temperature. An oil space 21 is situated between the cylinder and the piston above the double oil ring.

The cavity within the slippers 17 being filled with oil at the time of mounting, it subsequently suffices to squirt a few drops of oil through a hole in base 1, into the cylinder every 10 or 20 hours of running. This can be eliminated with a hole in the pump sleeve 7 above the labyrinth 15 of the leakage return. Capillarity then causes a slight oozing of the gas oil which sinks on the piston, and affords a convenient method of lubricating the wall of the gas cylinder. Such arrangement meets the conditions stated above and permits regular operation of the pump injectors.

The obturation of section S_1 of the entrance of the gases in the cylinder is achieved in such a way that the specific pressure of the obturation cone of the piston on the surface S_1 of the piston is sufficient for a perfect seal; that is to say, a strict control of the injection point by return spring A. The surface remains glossy and free of carbon under any condition and duration of operation. One of the surfaces of S_1 is of pure copper. Thus the percussion at the return to its seat of plastic material nullifies the vibration that would be inevitable if the pieces were of hard steel.

Endurance Test

A 50-hour endurance test at 800 rpm with the Winterthur engine in the Air Ministry laboratory has shown that the power and consumption curves recorded at the start and at the end of the 50-hour test were the same - at least, to within about 2 percent. The pump injector used for this test was similar to design I-b: plunger, 5 mm diameter, p_{i_2} = 610 atm, p_{i_1} = 300 atm, four nozzles of 0.20 mm diameter.

Cylinder-Piston Assembly of Fuel-Injection Engines

The three conditions necessary for the satisfactory operation of a piston of a pump injector must logically be verified for all the cylinder-pump systems of thermic engines.

Figure 17 exemplifies one version of these three conditions in a piston fitted with a combustion chamber with a dome. The solid bottom of piston 2 is doubly spherical or paraboloidal, thus securing a circular opening 7, which constitutes a thermic flywheel for equal distribution of the heat of the piston over a very large surface of cylinder jacket 8, cooled in the water chamber 9. The greatest diameter of engine pistons permits here the use of elastic piston rings without fear of deformations. Two, and even three, rings of small height h are fitted in each groove. The first is protected against high temperature by screw 10, which fixes dome 1 on the piston.

The knock vibrations are prevented by the slippers 11, fashioned in the piston skirt and machined with a little tightening on the bore. The slots of the elastic slippers 11 terminate at holes 12, drilled laterally above the axis of the piston, so that the wrong effect of the piston cannot induce rocking of the piston.

Being thus elastically centered in the bore, the craters of explosions of the combustion cycle cause no knocking; the piston rings, protected from high temperatures, operate properly without risk of gumming, and the mechanical efficiency of the engine is not changed under prolonged operation.

PART II

Chapter 5

INJECTION IN ENGINES WITH TURBULENCE CHAMBER

The Lister Engine

Test Stand

In turbulence engines, the very high speed of the air in the combustion chamber makes it possible to reduce the penetration and the dispersion of the spray. The air swirl assures the mixing and distribution of the fuel charge in the chamber; so the designers of these engines utilize low-injection pressures and single nozzles with much easier and surer operation than is obtainable with several fine nozzles and high pressure.

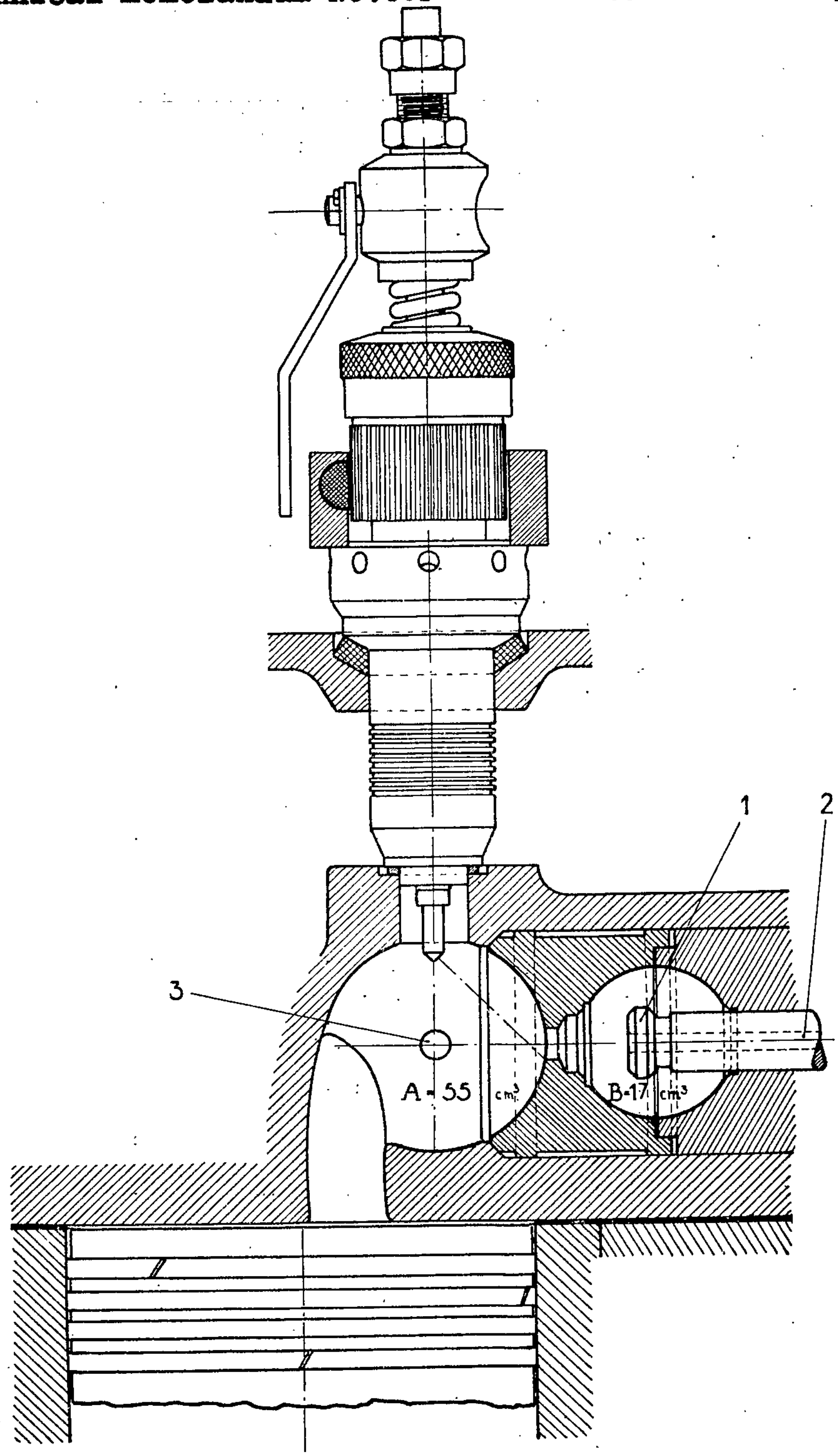


Figure 25.- Tuscher pump-injector unit mounted on Lister engine with double turbulence chambers. A, principal B, secondary turbulence chamber, 1 closing valve of B, 2 hole for Lehmann indicator, 3 hole for Zeiss indicator.

Under these circumstances it may logically be supposed that the period and pressure of injection have not the same effect on the combustion in engines with or without turbulence. Our object was to check this.

For this study, a Lister engine had been gratuitously placed at our disposal by Lister and Co., Ltd (England); of the four-stroke type, it develops 5 hp at 600 rpm, and 6 hp when supercharged; bore 95.3 mm, stroke 139.7 mm, displacement exactly 1 liter.

Figure 25 shows the piston at dead center, the turbulence chamber, and one version of the injectors used in these tests.

The cylinder head, designed by Ricardo, carries a turbulence chamber in two spherical parts A and B, interconnected by an orifice controlled by a valve 1. The first sphere A has a volume of 55 cm^3, the other, B, of 17 cm^3. This arrangement assures easy starting by closing at the start, the small chamber B by valve 1; the compression ratio ρ is then 19. When the engine is charged, valve 1 is opened, which lowers the compression ratio to 15. For recording the diagrams, valve 1 has a hole 2, communicating with the indicator. Another hole 3 (6 mm in diameter, 4 mm long) in the axis of chamber A opens into a chamber in the cylinder head which houses the piezometric quartz capsule of a Zeiss-Ikon indicator.

The engine test stand, as for the Winterthur engine, included a direct-current dynamometer fitted with a 716 mm arm and the identical test instruments. But the exciting current of the dynamo is supplied by the engine itself. At constant speed this current does not change with the engine charge. For the 600-700 rpm speed range, the calibration of this exciting current gives the following values:

Speed of rotation rpm	Amperes A	Voltage V	Power W	Power hp	Power kg/couple	Tare of balance without excitation kg
700	9	185	1,665	2.26	3,230	2,000
650	8.1	168	1,360	1.85	2,840	2,000
600	7	152	1,064	1.445	2,400	2,000

The empty dynamo being excited:

Speed	Actual tare	Tare indicated at balance	Power utilized in excitation	
rpm	kg	kg	kg/couple	hp
700	5,230	4,000	1,230	0.861
650	4,840	3,750	1,090	.709
600	4,400	3,500	900	.540

A correction of 1.230 kg at 700 rpm or of 0.900 kg at 600 rpm was added to the tare measurements recorded at the balance.

Feed Systems

By way of comparison with standard Bosch equipment on the Lister engine, two pump injectors were studied: one fitted with a 6 mm, the other with a 5 mm diameter plunger. They are similar to those used in the Winterthur engine tests. The gas-cylinder bore is 20 mm instead of 24 mm. The feed control is assured by a rack 4, carried in the attachment clamp 5. This rack engages a pinion 6, screwed on frame 7. The shock absorber B, consisting of a stack of synthetic-rubber disks in a slide 8, is maintained in piece 6 by a plug 9, which permits regulation of the force of shock absorber B.

The point where injection starts is doubly regulated by screw 10, screwed on the feed tube and by eccentric 11 which, by medium of a stop 12, acts on the tare of return spring A. The opening S_1 of gas cylinder 13 into the engine chamber, is controlled by pump sleeve 14, itself. A light alloy piston 15, connects with 14.

The lubrication of bore S_2 of the gas cylinder, is automatically assured at each injection by a leakage of compressed fuel in pump chamber 16 along S_3 of plunger 17 (fig. 27). For this purpose, a hole 18 in sleeve 14

at suitable distance from the pump chamber, communicates through groove 19 and other holes 20 suitably distributed in piston 15, with the bore of gas cylinder 13.

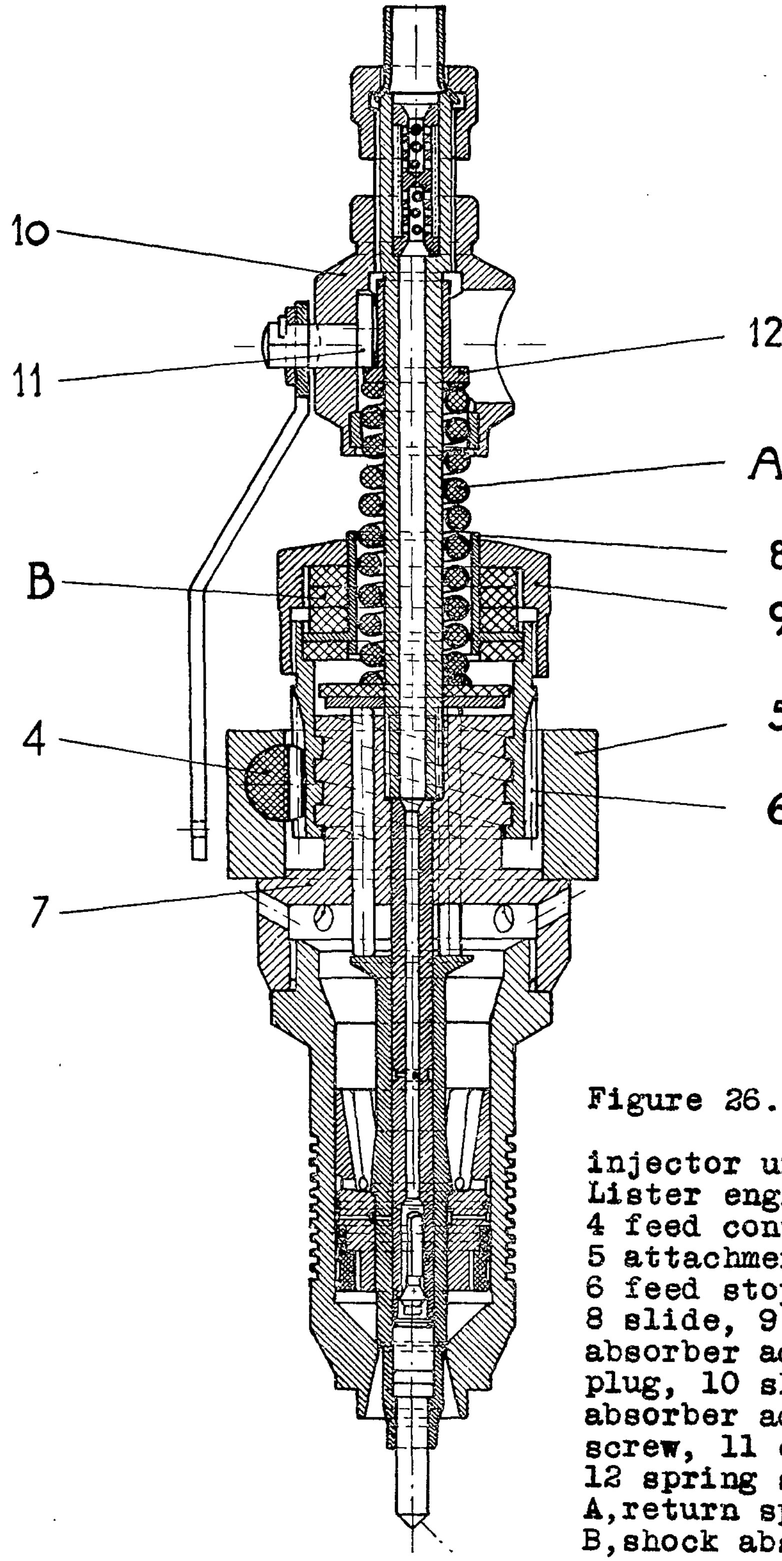


Figure 26.- Tuscher pump-injector unit for Lister engine.
4 feed control rack, 5 attachment clip, 6 feed stop, 7 base, 8 slide, 9 shock absorber adjusting plug, 10 shock absorber adjusting screw, 11 ecentric, 12 spring stop, A, return spring, B, shock absorber.

Piston 15 has a free space 21 of about 1.5 cm³ at the bottom of 13, which assures precise regulation of the point where ignition takes place. Indeed, a slight accidental leak in the closing cone S_1 of the gas cylinder has no effect on the lift of the piston and the regulation of the point of injection is stable up to its limit, at dead center.

Comparative Tests of Injection Systems

Bosch.- The Bosch pump, controlled by the engine regulator, is regulated with an advance of 20° 36' on dead center for the start of injection. The effective angle of the cam has an amplitude of 35° 36' at maximum feed, or 20° 36' B.T.C. and 15° A.T.C.

The plunger has a diameter of 7 mm. The injector with hydraulic control has one single nozzle 0.24 mm in diameter and inclined 45° toward the small turbulence chamber B. The spring of the needle of 6 x 3 mm diameter has a tare of 25 kg, making the pressure p_{i_1} = 118 atm at start of injection.

TABLE I. Standard Bosch Curve

Gas oil, density: 0.835 at 15° C

Speed rpm	Tare kg	Horsepower	Time 100 cm³ s	Consumption g/hp/h	Advance deg	Compression ratio ρ	Peak pressure atm
700	4,100	2.87	8' 19"	210	20° 36'	19	50
650	5,700	3.70	6' 25"	210	20° 36'	15	42
640	6,800	4.35	5' 35"	201	20° 36'	15	42
630	7,800	4.92	5' 3"	203	20° 36'	15	44
620	8,800	5.45	4' 33"	203	20° 36'	15	46
610	9,200	5.60	4' 18"	207	20° 36'	15	49
610	9,500	5.80	4' 4"	212	20° 36"	15	50
600	10,000	6.00	3' 42"	225	20° 36'	15	49

At 6 hp, the volume of one injection, therefore, is 90 mm³, the displacement being equal to 1 liter; the fuel/air ratio is a/c = 16. The useful stroke of the pump is 2.33 mm by 36° crankshaft, the rack of the pump being pushed home by the regulator. The period of maximum injection at 36° crankshaft at 600 rpm, is 0.010 second.

Practical speed of plunger: 0.233 m/s = v_{p_2}

(Fig. 28 is with Fig. 6)

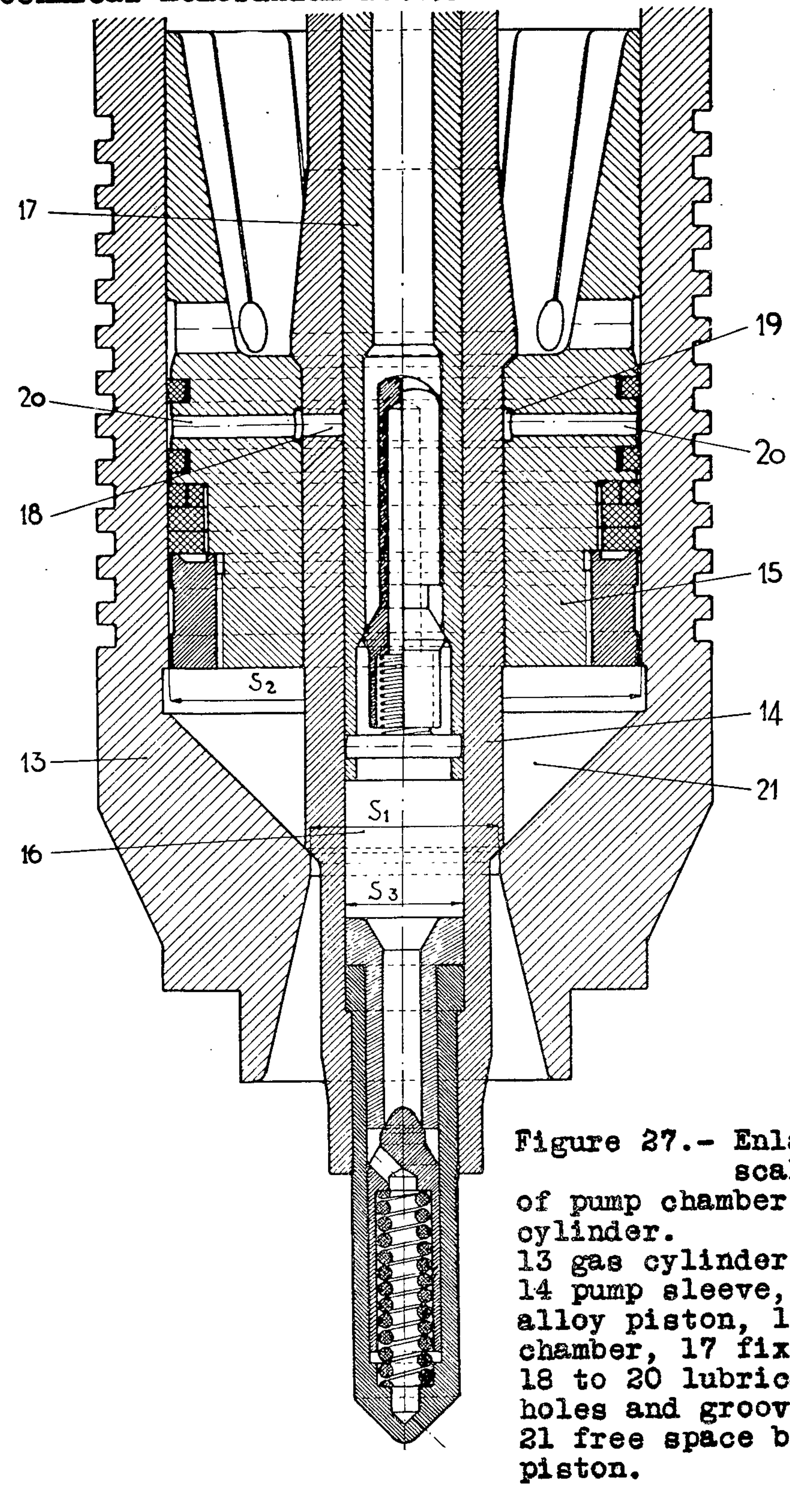


Figure 27.- Enlarged scale view of pump chamber and gas cylinder.
13 gas cylinder,
14 pump sleeve, 15 light alloy piston, 16 pump chamber, 17 fixed plunger, 18 to 20 lubricating holes and grooves,
21 free space below piston.

Injection pressure: $\left(\frac{S_3}{12}\frac{v_{p_2}}{S_7}\right)^2 = p_{i_2} = 245$ atm

Opening pressure: $p_{i_1} = 118$ atm is therefore well sustained

Pump-Injector System, 6 mm diameter plunger

$D_1 = 9$ mm $D_2 = 20$ mm $D_3 = 6$ mm

Opening pressure of needle ..... $p_{i_1} = 175$ atm

Injection pressure .... $p_{i_2} = 285$ atm

Rate of injection of nozzles.... $v_{i_2} = 191$ m/s

Three designs were studied at $p_{i_2} = 285$ atm under the following conditions:

No.	Nozzles		Piston speed m/s	Maximum stroke mm	Period of maximum injection s
	number	diameter mm			
IIa	2	0.25	0.675	3	0.0045
IIb	4	.25	1.350	3	.00225
IIc	7	.25	2.360	3	.00127

TABLE IIa

$p_{i_1} = 175$ atm $p_{i_2} = 285$ atm $S_7 = 2\times0.25$ $v_{p_2} = 0.675$ m/s

Speed rpm	Tare kg	Horse-power	Time 100 cm³	Consumption g/hp/h	Advance deg	Compression ratio ρ	Maximum pressure atm
700	4,000	2.80	8' 25"	212	2	19	45
700	5,500	3.85	6' 12"	211	2	15	35
700	6,600	4.62	5' 15"	205	2	15	40
700	8,300	5.81	4' 24"	195	2	15	42
700	9,400	6.58	3' 45"	202	2	15	42

At 6.58 hp the feed of one injection is 76.5 mm . The

period of injection is 4 milliseconds, or 16° 48' crankshaft. The fuel-air ratio is a/c = 18.5.

TABLE IIb

p_{i_1} = 175 atm p_{i_2} = 285 atm S_7 = 4x0.25 v_{p_2} = 1.35 cm/s

Speed rpm	Tare kg	Horse-power	Time 100 cm³ s	Con-sump-tion g/hp/h	Ad-vance deg	Compres-sion ratio ρ	Maximum pres-sure atm
700	4,100	2.87	8' 20"	209	2	19	48
700	5,600	3.92	6' 3"	211	2	15	42
700	6,800	4.76	5' 18"	198	2	15	45
700	8,000	5.60	4' 34"	195	2	15	46
700	9,400	6.58	3' 55"	193	2	15	48
700	10,400	7.28	3' 27"	198	2	15	48
700	11,600	8.12	2' 42"	228	2	15	50

At 8.12 hp, the injection is 104.5 mm³, the stroke 3.7 mm, and the injection period 0.00272 second, or 11° 30' crankshaft. The fuel/air ratio is a/c = 13.7.

TABLE IIc

p_{i_1} = 175 atm p_{i_2} = 285 atm S_7 = 7x0.25 v_{i_2} = 2.360 m/s

Speed rpm	Tare kg	Horse-power	Time 100 cm³	Con-sump-tion g/hp/h	Ad-vance deg	Compres-sion ratio ρ	Maximum pres-sure atm
700	4,100	2.87	8' 20"	209	2	19	48
700	5,600	3.92	6' 7"	209	2	15	40
700	6,800	4.76	5' 20"	197	2	15	44
700	8,200	5.74	4' 32"	192	2	15	48
700	9,600	6.72	3' 56"	189	2	15	50
700	10,700	7.49	3' 27"	193	2	15	52
700	12,000	8.40	2' 39"	224	2	15	49

At 8.4 hp, the feed at one injection is 107.5 mm³, the pump stroke c = 3.8 mm, the injection period t = 0.0016 second, or 6° 42' crankshaft, the ratio a/c = 13.2.

Pump Injector, 5 mm diameter plunger

D_1 = 8 mm D_2 = 20 mm D_3 = 5 mm

Opening pressure.............. p_{i_1} = 175 atm

Injection pressure .... p_{i_2} = 450 atm

Rate of injection of nozzles: v_{i_2} = 250 m/s

Three different injection periods were studied at this 450 atm injection pressure, as follows:

No.	Nozzles		Piston speed	Maximum stroke	Duration
	Number	Diameter mm	m/s	mm	sec
IIIa	1	0.20	0.40	4.3	0.0105
IIIb	8	.20	3.20	4.9	.00153
IIIc	6	.25	3.80	4.9	.00130

TABLE IIIa

p_{i_1} = 175 atm p_{i_2} = 450 atm S_7 = 1×0.20 v_{p_2} = 0.40 m/s

Speed rpm	Tare kg	Horsepower	Time 100 cm³ s	Consumption g/hp/h	Advance deg	Compression ratio ρ	Maximum pressure atm
700	4,500	3.15	7' 36"	209	2	19	35
700	6,500	4.55	5' 42"	192	2	15	35
700	7,800	5.46	5' 0"	184	2	15	38
700	9,000	6.30	4' 10"	190	2	15	40
700	10,700	7.49	3' 22"	198	2	15	46

At 7.5 hp, the injection is 85 mm³, the stroke 4.3 mm, the injection period 0.0107 second, or 45° crankshaft, fuel/air ratio a/c = 16.7. Fitted with design No. III-a, the Lister engine operated remarkably quietly.

TABLE IIIb

p_{i_1} = 175 atm p_{i_2} = 450 atm S_7 = 8×0.20 v_{i_2} = 3.20 m/s

Speed rpm	Tare kg	Horse-power	Time 100 cm³ s	Con-sump-tion g/hp/h	Advance deg	Compres-sion ratio ρ	Maximum pres-sure atm
700	4,500	3.15	8' 5"	197	2	19	35
700	6,600	4.62	5' 40"	191	2	15	42
700	9,300	6.51	4' 12"	182	2	15	45
700	10,500	7.35	3' 40"	185	2	15	46
700	11,900	8.33	3' 9"	190	2	15	52
700	12,300	8.61	2' 56"	198	2	15	50

At 8.61 hp, the injection is 97 mm³, the effective pump lift 4.9 mm, the injection period t = 0.00153 second, or 6° 24', fuel/air ratio a/c = 14.5/1. The running is harsh but becomes softer as the charge increases.

TABLE IIIc

p_{i_1} = 175 atm p_{i_2} = 450 atm S_7 = 6×0.25 v_{p_2} = 3.80 cm/s

Speed rpm	Tare kg	Horse-power	Time 100 cm³ s	Con-sump-tion g/hp/h	Advance deg	Compres-sion ratio ρ	Maximum pres-sure atm
700	4,300	3.01	8' 5"	205	2	19	38
700	6,200	4.34	5' 59"	192	2	15	40
700	7,600	5.32	5' 8"	183	2	15	44
700	8,800	6.16	4' 34"	178	2	15	44
700	10,300	7.21	3' 50"	180	2	15	48
700	11,000	7.70	3' 34"	182	2	15	49
700	11,700	8.19	3' 18"	185	2	15	54
700	12,500	8.75	2' 56"	195	2	15	52

At 8.75 hp, the feed is 97 mm³, the stroke c = 4.9 mm, the spray period t = 0.00128 second, or 5° 22', fuel/air ratio a/c = 14.5.

Examination of these results show plainly that in an engine with turbulence chamber:

1. The power developed by the engine varies conversely to the duration of injection.

2. The specific fuel consumption varies inversely to the pressure of injection.

These two laws, brought in evidence by the Winterthur engine tests, are in consequence, general, and apply equally to engines with turbulence chambers and those with direct injection.

Concentration - Air-Fuel Ratio

The fuels used in Diesel engines are largely composed of 85 kg of carbon and 14 kg of hydrogen per 100 kg of hydrocarbon. The remaining 1 percent consists of water, sulphur, impurities, etc., and may be disregarded..

The products of complete combustion of the hydrocarbon are carbonic acid and water vapor. The molecular weight of carbon and hydrogen being respectively, 12 and 2, the oxygen necessary for complete combustion of 100 kg of fuel in kilogram/molecule is:

$$\frac{85}{12}(CO^2) + \frac{1}{2} \times \frac{14}{2}(H^2O) = 7.1(CO^2) + 3.5(H^2O) = 10.6 \text{ molecule of } O_2$$

The air necessary for this is:

$$\frac{10.6 \times 100}{21} = 50.5 \text{ molecules of air}$$

Under standard conditions, of 760 mm Hg and 15° C, the volume of a molecule of gas is given by the equation of state: $pV = RT$:

$$V = \frac{848 \times 288}{10000} = 24.4 \text{ m}^3/\text{kg}$$

The air needed for complete combustion of a kilogram of fuel is

$$v = \frac{50.5 \times 24.4}{100} = 12.3 \text{ m}^3/\text{kg of fuel}$$

in weight:

$$G = 12.3 \times 1.19 = 14.5 \text{ kg of fuel/air}$$

For this ratio of 14.5 the mixture is saturated, hence the concentration of 100 percent.

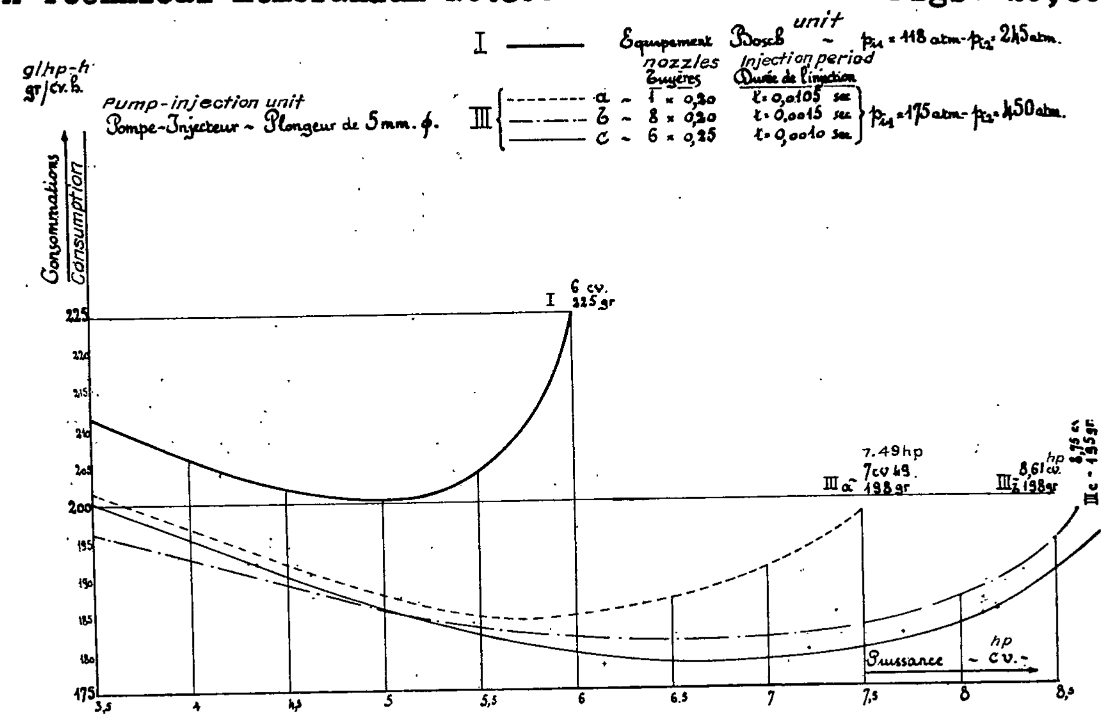


Figure 29.- Comparative power and consumption curves. Plunger diameter 5 mm

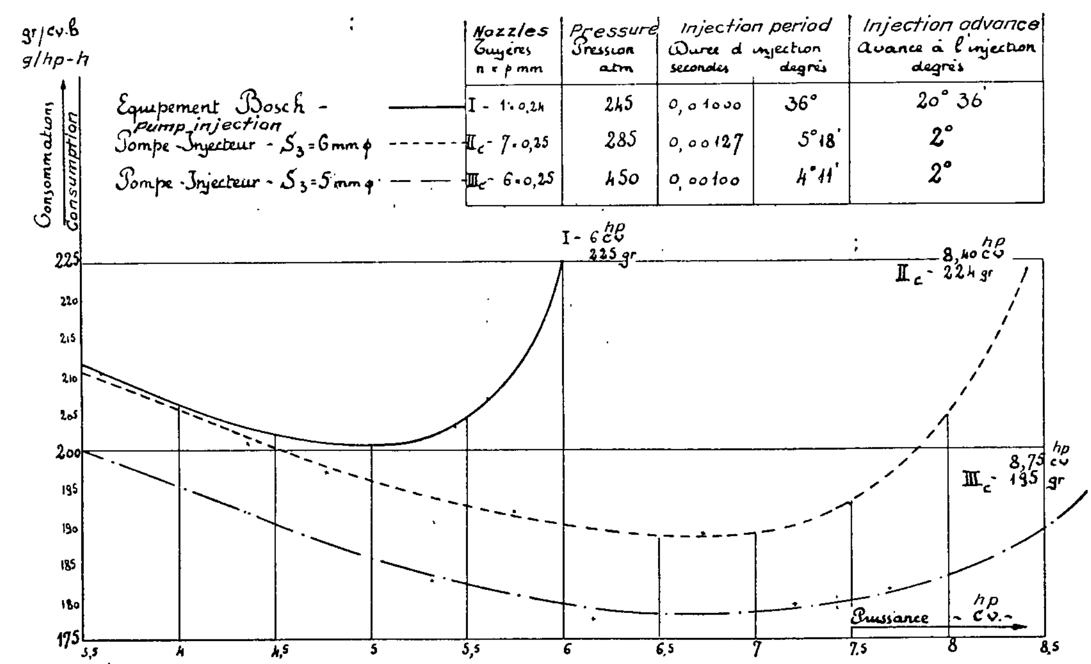


Figure 30.- Comparative power and consumption curves. Numbers I, IIc and IIIc

For the maximum power developed by the Lister engine fitted with the different designs, it was found that:

Design	I	IIa	IIb	IIc	IIIa	IIIb	IIIc
a/c ratio	16	18.5	13.7	13.2	16.7	14.5	14.5
Excess air	+.11	+.28	-.055	-.095	+.15	0	0
Concentration percent	90	78.5	106	110	87	100	100
Horsepower	6	6.58	8.12	8.40	7.49	8.61	8.75
Specific consumption, g/hp/h	225	202	228	224	198	198	195
Effective efficiency	28.2	31.4	27.9	28.4	31.6	32.2	32.6

From this table it is readily apparent that by shortening the injection period, the air-fuel ratio can be reduced to the limit of saturation, 14.5 kg of air per kilogram of gas oil, with an attendant improvement in effective efficiency of the engine.

It is quite evident that for the two points IIb and IIc, where the concentration exceeds 100 percent, the pump feed was excessive; in this case, the excess of fuel cannot burn and produces smoke at the exhaust. The specific fuel consumption for these two points are, for the rest, 30 g higher than for IIIb and IIIc, although for the lower horsepower, the difference amounts to no more than 10 to 20 g/hp/h.

With a 100-percent concentration, and a 75-percent boost charge, the Lister engine manifests no smoke in the exhaust when fitted with designs IIIb or IIIc, and the combustion takes place entirely during the expansion stroke of the piston.

Chapter 6

DELAYED COMBUSTION CYCLE

Rapidity and Efficiency of Combustion Cycle

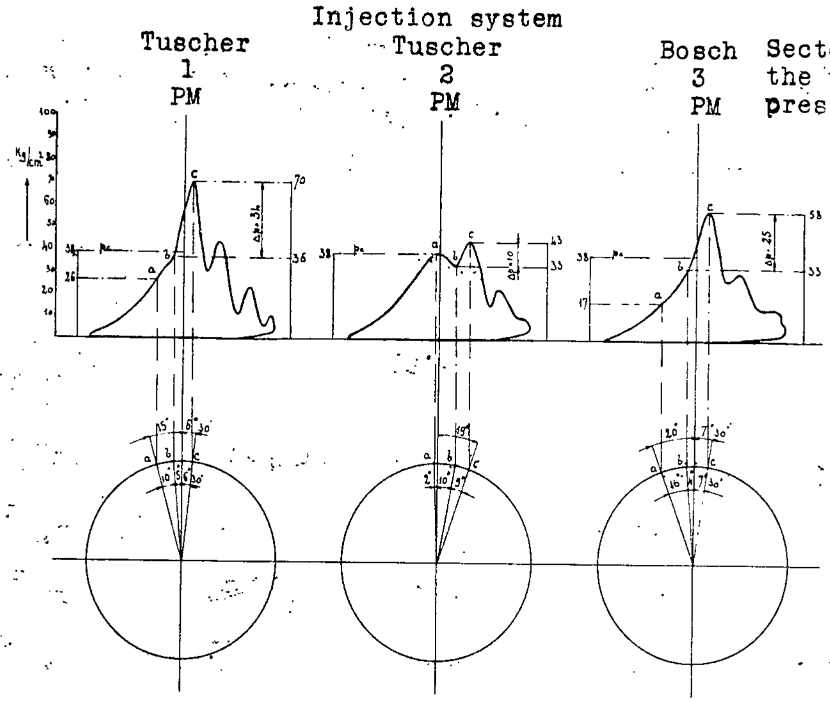
Unquestionably, an increase in compression ratio, in the speed and pressure of injection, tends to increase the quickness of combustion. The cycle approaches that for constant volume, where $\frac{dp}{dt} = \infty$. Under these conditions of optimum thermal efficiency, the fatigue of engine parts, such as pistons, connecting rods, bearings, etc., is no longer compatible with mechanical strength and their satisfactory preservation.

But experience has proved it to be possible to utilize high compression ratios, high speeds, and pressures of injection without promoting greater fatigue in engine components, provided that a delayed combustion cycle is formulated.

As evident proof, we show figure 31. These diagrams were recorded at 700 rpm on the Lister engine at exactly 3 horsepower. They afford a ready means for computing or reading the data appended in the following two tables.

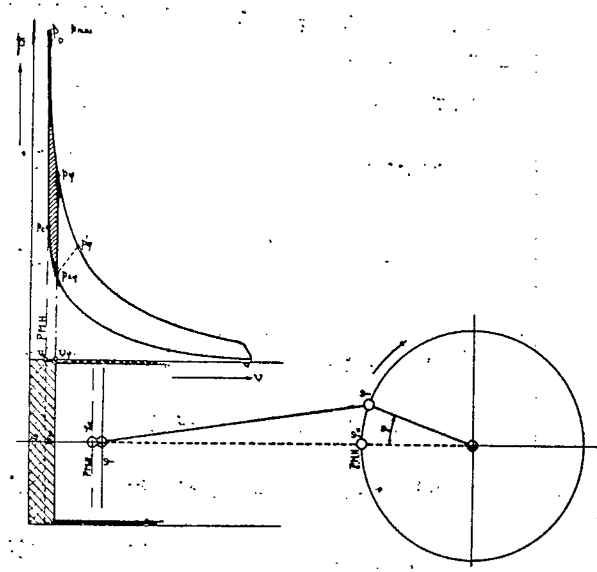
For the first diagram, the engine is fitted with a pump-injector system of D_3 = 5 mm diameter, five atomizing nozzles of 0.25 mm diameter, and injection point set at 15° advance. The second diagram is based on the same equipment, except that the injection advance is reduced to 2°. For the third, the normal Bosch equipment is used, injection starting at 20° B.T.C.

	1	2	3
System	Tuscher	Tuscher	Bosch
Injection advance, deg	15	2	20
Injection pressure, atm	450	450	285
Compression ratio, ρ	19	19	19
Nozzles, n × D	5×0.25	5×0.25	1×0.24
Injection period, s	0.00057	0.00057	0.00345
Injection period, deg	2° 24'	2° 24'	14° 25'
rpm	700	700	700
horsepower	3	3	3
Specific consumption, g/hp/h	212	200	210
Effective efficiency, percent	30	31.7	30.3



For figure 31

Sector a-b is the ignition lag, sector b-c the period of combustion with increase in pressure.



	Tuscher 1	Tuscher 2	Bosch 3	
Injection				
Advance...φ°	15°	2°	20°	degrees
Pressure..atm	450	450	285	kg/cm²
Period....s	0.00057	0.000057	0.00345	seconds
" ....φ°	2°24'	2°24'	14°25'	degrees
Combustion				
Max pressure..atm	70	43	58	kg/cm²
Period ...... s	0.00273	0.00213	0.00274	seconds
" .... φ°	11°30'	9°	11°30'	degrees
Speed .... dp/dt	12.5	4.7	9.1	kg/ms
" .... dp/dφ	2.95	1.11	2.2	kg/deg

Figure 31.— Advanced and retarded combustion cycles, Lister engine at 3 hp, 700 rpm.

Figure 32.— Retarded combustion cycle — pressure-volume card.

For figure 31.

V=displacement   V = 1000 cm³

V_0=volume of combustion chamber; V_0=54cm³ (ρ=19)

a=starting point of injection

b= " " " combustion

c=point of maximum pressure of cycle.

With the minus sign (-) indicating B.T.C., and the plus sign (+) denoting A.T.C., figure 31 affords:

No.	1		2		3	
	Lag a-b	Combustion b-c	Lag a-b	Combustion b-c	Lag a-b	Combustion b-c
Angles, ° ± T.C.	-15 to -5	-5° to +6° 30'	-2° to +10°	+10 to +19	-20° to -4°	4 to +7° 30'
Duration, deg	10°	11° 30'	12°	9°	16°	11° 30'
Duration, sec	0.000237	0.00274	0.00285	0.00213	0.00382	0.00274
Variation V_o cm^3	-30.27	+6.43	+15.2	+39.3	-57.87	+8.56
-- V_o in % V_o	-56%	+12%	+28%	+73%	-109%	+15.8%
Variation pressure	+10 atm	+34 atm	-5 atm	+10 atm	+16	+25 atm
Rapidity of combustion, dp/dt in kg/ms		12.5		4.7		9.1
Rapidity of combustion, dp/dφ in kg/deg		2.95		1.11		2.2
Peak pressure		70 atm		43 atm		58 atm

It is readily apparent that cycle No. 2, with retarded combustion, secures not only the best engine efficiency, but also promotes all the conditions of good preservation of the component parts, while rapidity of combustion and peak pressure of the cycle are reduced to a minimum.

Peak Pressure of Retarded Cycle

Assume a combustion at constant volume at dead center. For a specified concentration, air-fuel = a, this combustion defines a maximum pressure p_o, which may not be exceeded. Since $\frac{dp}{dt} = \infty$, the heat loss by the walls during combustion is zero.

At the point of maximum pressure $p_o\ v_o$ the expansion of the burned gases follows the adiabatic law:

$$p_o\ v_o^{\gamma} = \text{constant}$$

Suppose now the combustion at constant volume takes place after dead center for a crankshaft position φ. In this instance the volume V_φ of the combustion chamber is

$$V_\varphi = V_o + (1 - \cos\varphi)\ V$$

where V_o and V denote the volume of the compression chamber and of the displacement, respectively.

The pressure p_φ of combustion at constant volume in point φ of the piston, cannot be other than that obtained by an expansion of p_o to p_φ on the adiabatic $p_o\ v_o$ = constant. Hence, the relation:

$$p_o\ v_o^{\gamma} = p_\varphi \left[V_o + (1 - \cos\varphi)\ V\right]^{\gamma} = \text{constant}$$

whence

$$p_\varphi = p_o \left[\frac{V_o}{V_o + (1 - \cos\varphi)\ V}\right]^{\gamma}$$

The calculation of this value p_φ of the peak pressure of a retarded combustion for the Lister engine with $\rho = 15$ compression ratio:

$V_o = 72\ \text{cm}^3$ $\quad V = 1072\ \text{cm}^3$ $\quad \gamma = 1.4$ $\quad p_o = 100$ atm

affords:

Constant volume combustion A.T.C.		Volume of combustion chamber increase		Peak pressure of cycle $p_\varphi = p_o \left[\frac{V_o}{V_o + (1 - \cos\ \ )V}\right]^\gamma$	Decrease of peak pressure
Retarded deg	V_o cm^3	ΔV_o cm^3	ΔV_o percent	atm	percent p_o
0	72	0	0	100	0
5	76	4	5.5	93.4	6.6
10	87	15	21	77.3	22.7
15	107	35	49	59.1	40.9
20	133	61	85	43.7	56.3
25	167	95	132	32.2	67.8
30	207	135	187	23.9	76.1

A 10° lag in combustion A.T.C. reduces the peak pressure by a fourth of its theoretical value for top center, and 41 percent for a 15° lag.

In reality, the combustion is not realized at constant volume, it has an appreciable duration of several milliseconds, and the reduction in the peak pressure of the cycle is still more accentuated than indicated by the theoretical calculation.

The real peak pressure is a point $p_\varphi' < p_\varphi$ on the adiabatic $p_o \cdot v_o^\gamma$ = constant.

On comparison of the diagrams 1 and 2 of figure 31, it is found that the peak pressure drops from 70 to 43 atm for a 10° delay of cycle (plot 2). The drop in peak pressure is 38.5 percent.

We also find that the rapidity of combustion dp/dt itself, is considerably reduced when the cycle is retarded: plot No. 1, $\frac{dp}{dt}$ = 12.5 kg/ms; plot No. 2, $\frac{dp}{dt}$ = 4.7 kg/ms.

$$\text{Ratio of reduction, } \frac{12.5}{4.7} = 2.7$$

For a 10° retarded combustion at top center, the quickness of combustion and, consequently, the fatigue of the engine is almost three times less.

It further was found that this 10° delayed cycle defines the best effective efficiency of the engine: η_e = 31.7 percent (200 g/hp/h) instead of η_e = 30 percent (212 g/hp/h) when the combustion is situated normally at top center (plot No. 1).

Contrary to these empirical findings, it is seen on plot pv (fig. 32) that the energy supplied by the cycle of retarded combustion p_{c_φ}, p_φ is inferior to that of a combustion $p_c - p_o$ at constant volume in T.C. The shaded area on the left-hand side gives the value of the energy lost by the delay in combustion.

The theoretical thermal efficiency of a delayed combustion cycle therefore being plainly inferior to that of a combustion at top center, it seems logical to admit that the kinematics of our engines - pistons, connecting rods, crankshaft - have a deplorable instantaneous mechanical efficiency when the peak of the pressure of a cycle exceeds a certain limit p_o or the quickness dp/dt of the combustion passes beyond a specified value. Therefore, a delayed combustion cycle of inferior thermal efficiency practically insures the engine a better effective efficiency; hence, finally, a better utilization of the burned fuel.

It goes without saying, that by merely reducing the compression ratio, one cannot anticipate the same results obtained by the use of a retarded combustion.

The temperature change in a cycle follows substantially that of the pressure. From this point of view, the retarded combustion cycle offers the advantage of reduced heat stresses in the pistons and liners of the engines, and lower heat loss through the walls. But this advantage alone could not give a rational explanation of the gain made from the use of this form of cycle, if the mechanical efficiency of the engine were not at the same time improved considerably.

To our knowledge, no study of the instantaneous variation of the mechanical efficiency of an engine in relation to the quickness dp/dt of combustion has ever been published. Such research should throw some light on the difficulties encountered the past few years with high-power, high-speed Diesels for aircraft use. Such studies might be carried out by measuring the effective efficiencies of an engine under varying degrees of delayed combustion, the peak pressure p_φ and dp/dt, and computing the entropy diagrams and heat balances of realized cycles. A comparison of the two sets of results would make it possible to determine the best p_φ and dp/dt along with the simultaneous change in thermal and effective efficiencies of the engine. Then it would be a simple matter to deduce the laws of instantaneous variation of the mechanical efficiency in relation to the two values of p_φ and dp/dt. Unfortunately, we have not the means to undertake this research.

Records of Delayed Combustion Cycles Obtained with the Zeiss-Ikon Cathodic Indicator

It was believed that the Watt indicator had too much inertia to give an exact check of the marked pressure changes within the brief interval of a few milliseconds. So the Air Ministry placed at our disposal a Zeiss-Ikon indicator with quartz crystal with which to check the plots recorded so far with the Lehmann-Michels instrument. These records were made simultaneously with the two indicators.

The pressure tap of the first (fig. 25) is a hole 6 mm in diameter in the axis of the first Ricardo sphere, the quartz crystal being located some 4 or 5 mm from the sphere; the pressure tap of the second is across a 3.5 mm hole, about 250 mm long in the axis of the valve closing the second sphere B.

For these tests:

1. The engine speed was kept exactly at 700 rpm, by correcting the pump feed manually.

2. Three systems of injection were essayed:

a) Bosch: injector with nozzle 0.24 mm in diameter, at 45° toward chamber B, the pump being set at 20° advance on top center.

b) Tuscher: pump injector with one nozzle of 0.20 mm diameter at 40° toward chamber B, injection advance varied from 3° to 6°.

c) Tuscher: pump-injector unit, 8 nozzles of 0.20 mm diameter, fanwise toward chamber B. One of the nozzles points toward center of sphere A; injection point varied from 0° to 3°

a) Bosch unit	b) Tuscher unit ran smoothly	c) Tuscher unit ran hard
1 x 0.24	1 x 0.20	8 x 0.20

3. Four points on the power curve were studied:

	Horsepower developed	Compression ratio
1. - - - - - - - - - -	3	19
2. - - - - - - - - - -	3	15
3. - - - - - - - - - -	5	15
4. - - - - - - - - - -	maximum at 700 rpm	15

The four graphs, 34 to 37, represent the diagrams recorded in this experiment. The measurements and results are given in tabular form.

On comparing these results, it is seen that the measurements supplied by the two types of indicators agree to within 2 to 3 percent. Except for the adiabatic expansion of the cycle, the study of which holds nothing new, the inertia of the equipment and the marked flattening of the pressure lead of the Lehmann indicator do not distort the records appreciably. The conclusions so far drawn from the reading of the diagrams may be considered as correct.

Effect of Compression Ratio ρ

Measurements I and II, $\rho = 19$ and $\rho = 15$, 3 horsepower, have shown that:

1. The effective engine output is a function of the compression ratio ρ. The gain is about 2.5 percent for the three injection systems analyzed, when ρ passes from 15 to 19.

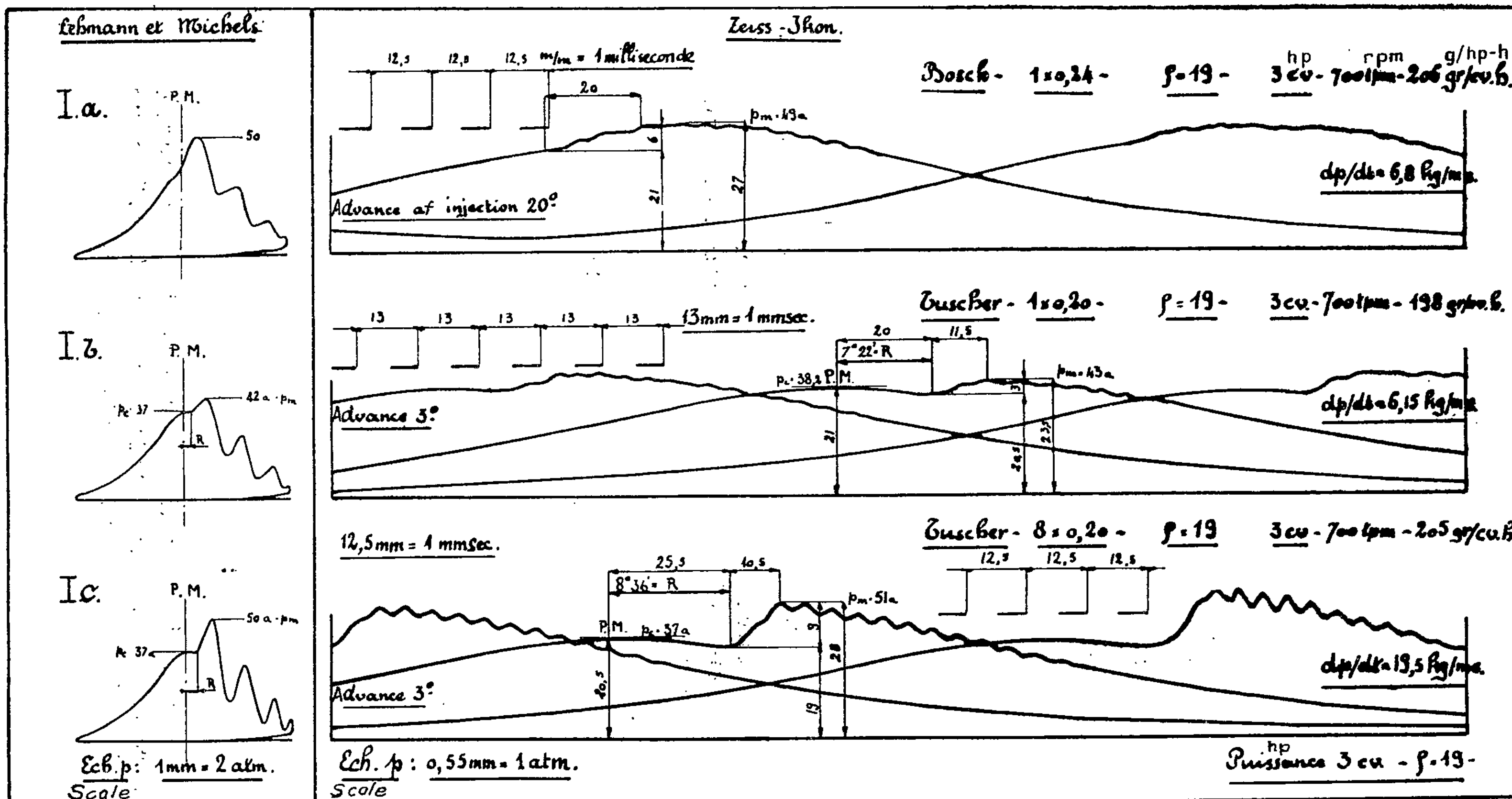


Figure 34.- Lehmann-Zeiss cards, 3 hp, 700 rpm, compression ratio: ρ =19.

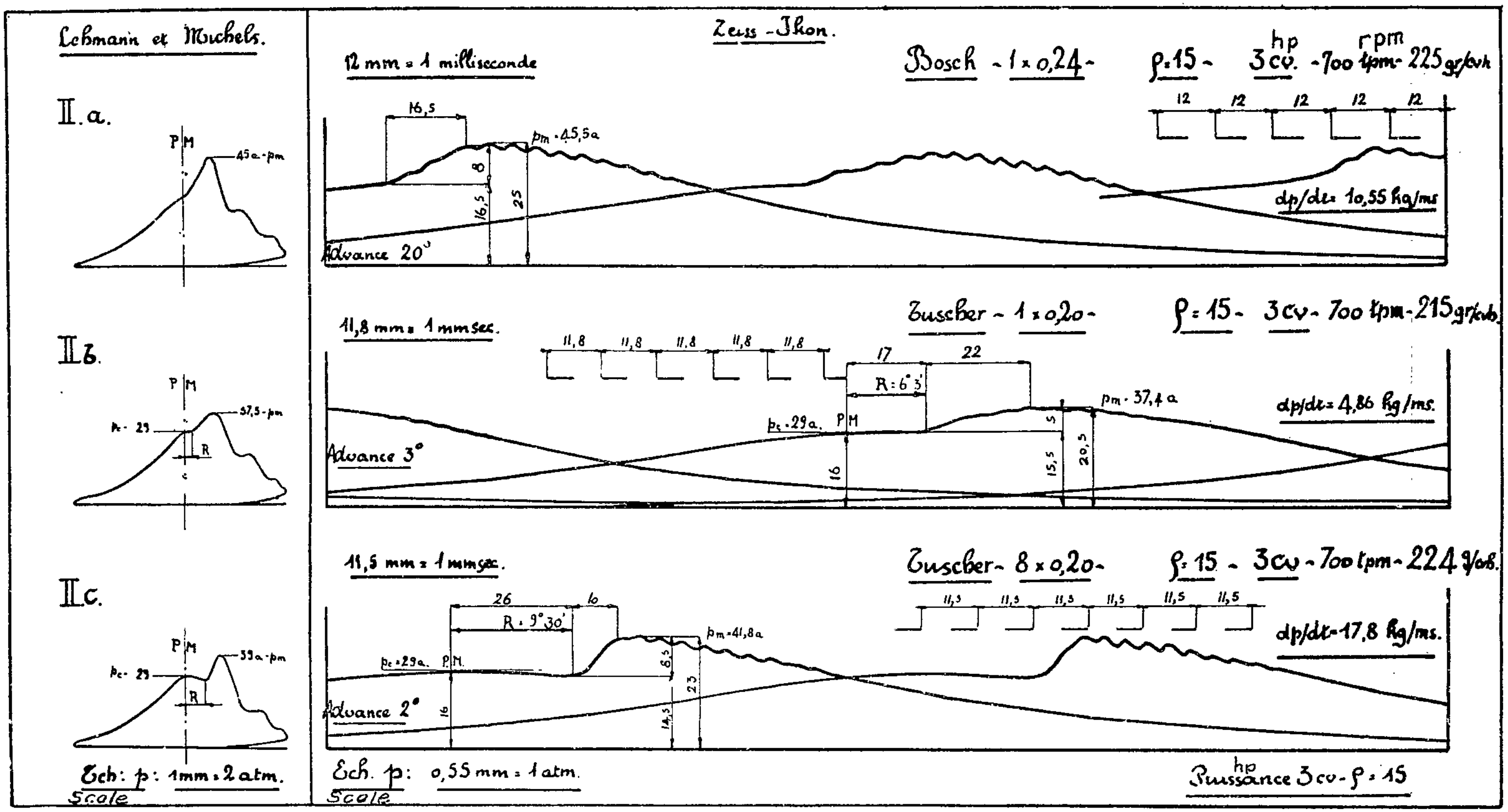


Figure 35.- Lehmann-Zeiss cards, 3 hp, 700 rpm, compression ratio: ρ = 15.

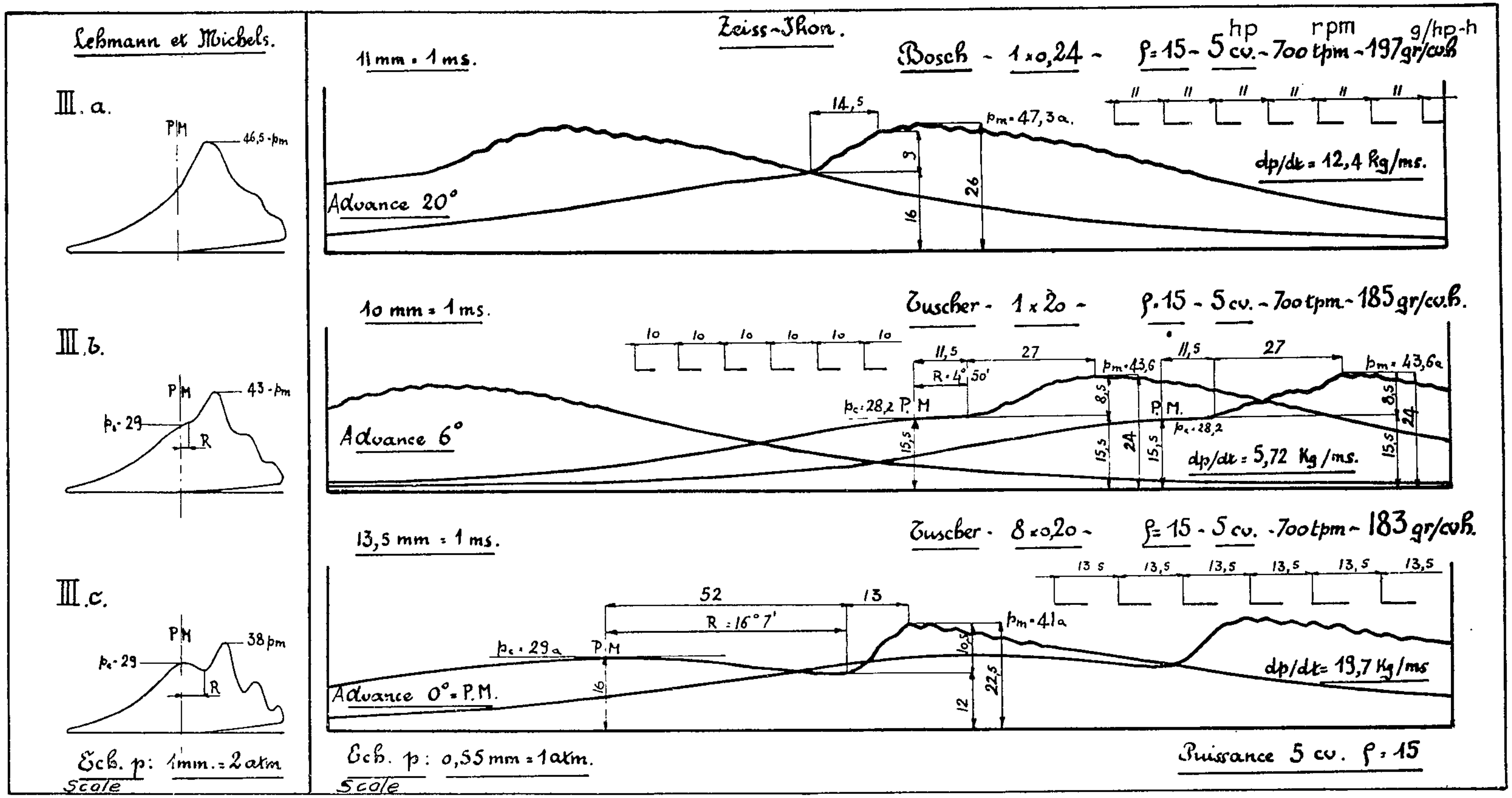


Figure 36.- Lehmann-Zeiss cards, 5 hp, 700 rpm, compression ratio: ρ = 15.

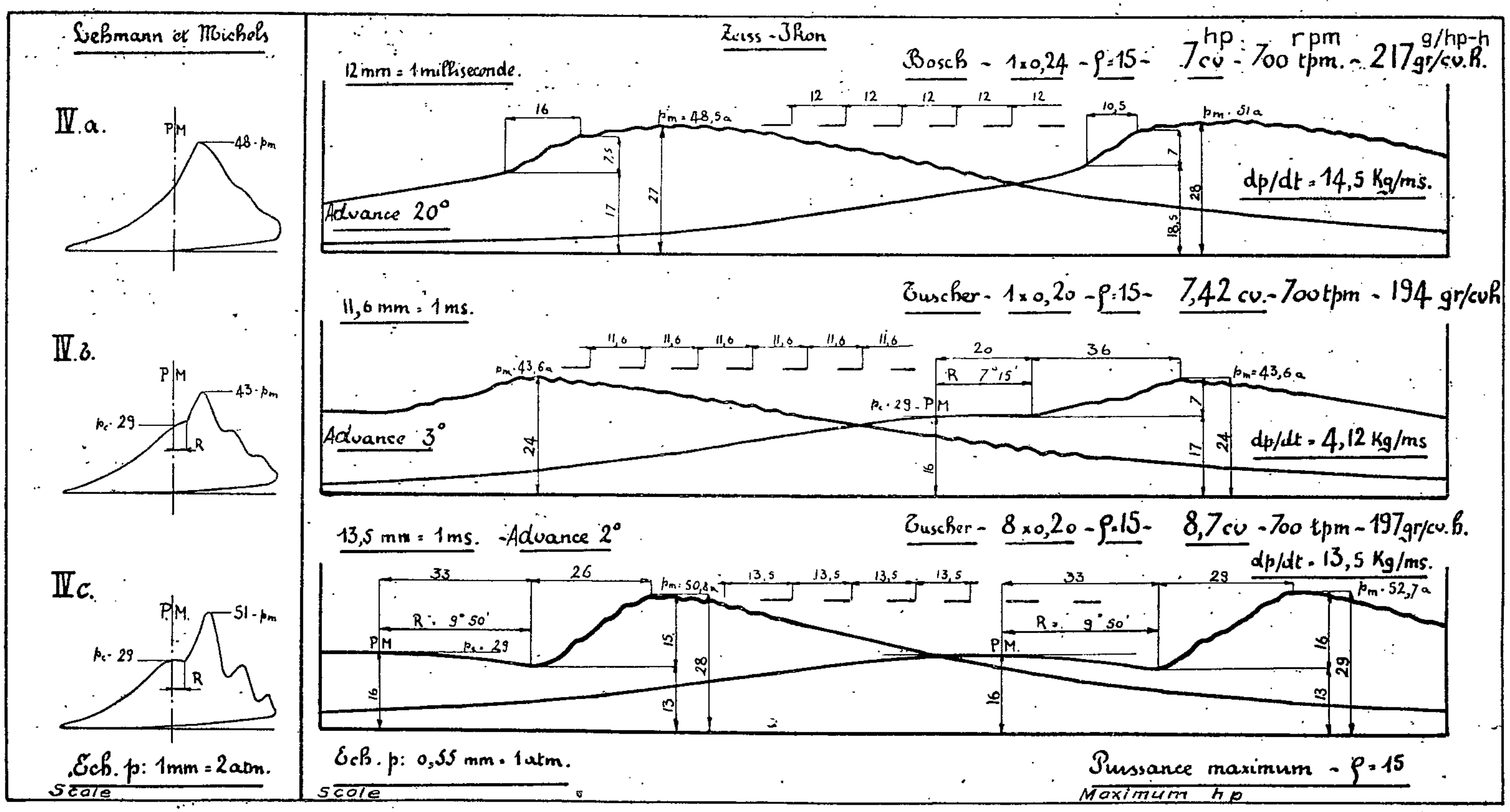


Figure 37.- Lehmann-Zeiss cards, max. hp, 700 rpm, compression ratio: ρ=15

2. The peak pressure of the cycle rises with ρ; it amounts to 3.5 atm for the advanced cycle given by the Bosch unit (p_n = 45.5 atm for $\rho = 15$ and 49 atm for $\rho = 19$). It is even more accentuated when the cycle is retarded: 6 atm for unit b) with smooth running, and 9 atm for unit c) with hard running. In fact, in a unit such as b) or c), controlled by the compression of the engine, the injection pressure is proportional to the compression; the penetration of the atomized spray is therefore much greater when ρ is increased. It is therefore natural that the peak pressure of the cycle and the quickness of combustion rise even more rapidly in relation to the compression ratio.

3. The quickness of combustion rises with ρ when the cycle is retarded (units b) and c)).

4. Contrariwise, for the Bosch unit with advanced cycle, the quickness is 6.8 kg/millisecond for $\rho = 19$, and rises to 10.55 kg/millisecond for $\rho = 15$. By listening to the noise while the engine runs, this anomaly is audible. The running seems to be smoother and more regular when the second chamber B in the Lister engine is closed.

This finding, seemingly abnormal at first glance, can be explained by the fact that the Bosch injects the fuel at a pressure independent of the compression of the engine. In fact, the injection pressure being identical in both cases, the penetration and dispersion of the spray are less when the compression ratio is higher, the portion of the combustion air involved by the gas oil is smaller, has a greater penetration, and consequently a less active combustion of the mixture.

From this it is logical to conclude that the quickness of combustion varies inversely with the mixture concentration. This fact is confirmed even more clearly by the Zeiss records of the retarded cycles - units b) and c) - where the rapidity of combustion is seen to decrease as the engine charge and, consequently, the saturation of the air-fuel ratio is increased.

An examination of the Zeiss records shows, in a general way, that a greater rapidity of combustion is accompanied by more accentuated pressure undulations during combustion and expansion. In our opinion, these undulations, which come and go for no apparent reason, are indicative of a turbulent regime of the combustion. The

Table B

	I			II			III			IV		
Compression ratio, ρ	19			15			15			15		
hp	3			3			5			Maximum		
rpm	700			700			700			700		
Units	a	b	c	a	b	c	a	b	c	a	b	c
Tare of balance, kg ..	4,200	4,300	4,400	4,200	4,300	4,300	7,000	7,200	7,300	10,000	10,600	12,400
hp developed	2,94	3,01	3,08	2,94	3,01	3,01	4,90	5,04	5,11	7,00	7,42	8,68
To burn 100cm³ gas-oil	8′15″	8′23″	7′53″	7′34″	7′43″	7′25″	5′12″	5′20″	5′20″	3′18″	3′28″	2′55″
Spec. consump.,g/hp/h	206	198	205	225	215	224	197	185	183	217	194	197
Effective output,% ..	30,7	32	31	28,2	29,5	28,3	32,3	34,4	35	29,2	32,8	32,3
Feed, mm³............	34,7	34	36,2	37,8	37	38,5	54,5	53,6	53,6	86,5	82,5	96
Stroke, mm	0,9	1,73	1,84	0,98	1,88	1,96	1,42	2,73	2,73	2,25	4,20	4,88
Injection period,ms ..	3,44	4,32	0,575	3,74	4,70	0,612	5,43	6,83	0,855	8,6	10,5	1,523
Period, degrees.......	14°24′	18°15′	2°25′	15°40′	19°45′	2°35′	22°42′	28°40′	3°35′	36°	44°6′	6°24′
Air fuel ratio,a/c ...	39,7	40,5	38,2	36,6	37,2	36	25,3	25,8	25,8	16	16,7	14,5
Excess air,ε	1,74	1,8	1,66	1,53	1,56	1,5	0,76	0,78	0,78	0,11	0,15	0,00
Concentration, % ...	36,5	36	37,6	39,5	39	40	57	56	56	90	87	100
Compress. A.T.C.:												
Zeiss measure, atm ...	—	38,2	37	—	29	29	—	28,2	29	—	29	29
Lehmann measure, atm .	—	37	37	—	29	29	—	29	29	—	29	29
Peak pressure:												
Zeiss measure, atm ...	49	43	51	45,5	37,4	41,8	47,3	43,6	41	51	43,6	51
Lehmann measure, atm .	50	42	50	45	37,5	39	46,5	43	38	48	43	51
Speed of combustion:												
dp/dt ,kg/ms	6,8	6,15	19,5	10,55	4,86	17,8	12,4	5,72	19,7	14,5	4,12	13,5
dp/dφ ,kg/deg	1,6	1,46	4,64	2,51	1,16	4,24	2,96	1,36	4,68	3,45	0,98	3,21
Injection advance,deg	20°	3°	3°	20°	3°	2°	20°	6°	0°P.M.	20°	3°	2°
Retarded combustion:												
ms A.T.C.	—	1,75	2,04	—	1,44	2,26	—	1,15	3,85	—	1,725	2,45
deg A.T.C.	—	7°22′	8°36′	—	6°3′	9°30′	—	4°50′	16°7′	—	7°15′	9°50′
Ignition lag:												
Duration in ms	—	2,48	2,77	—	2,17	2,75	—	2,58	3,85	—	2,45	2,82
" " deg	—	10°22′	11°36′	—	9°3′	11°30′	—	10°50′	16°7′	—	10°15′	11°50′
Combustion:												
Duration in ms	1,6	0,885	0,84	1,375	1,87	0,87	1,32	2,7	0,97	0,875	3,1	2,15
" " deg	6°44′	3°43′	3°31′	5°48′	7°50′	3°39′	5°33′	11°21′	4°4′	3°40′	13°	9°

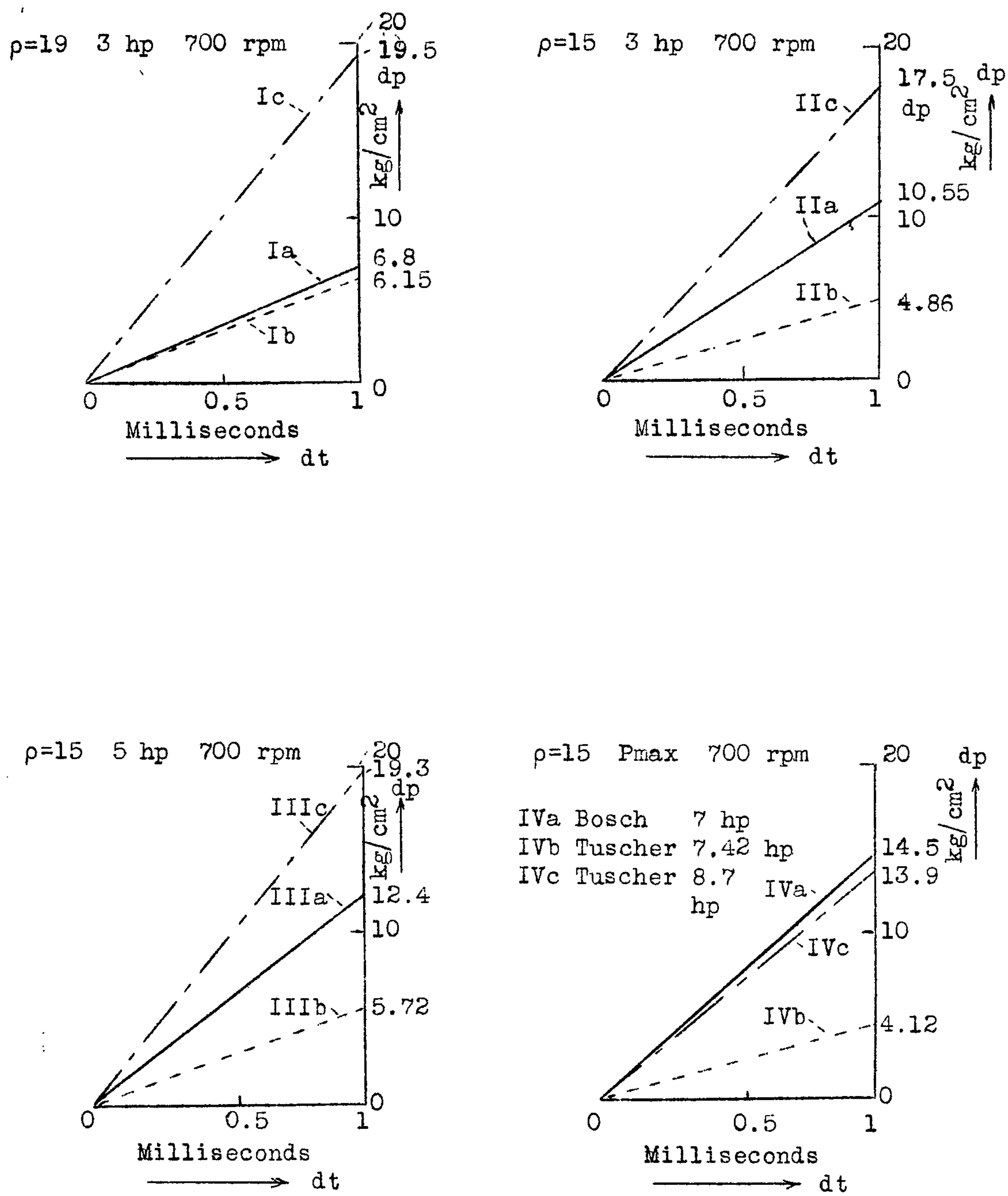


Figure 38.- Speed of combustion, speed and peak pressure represented (Fig. 39) plotted against power developed; No. II, III, and IV, ρ=15.

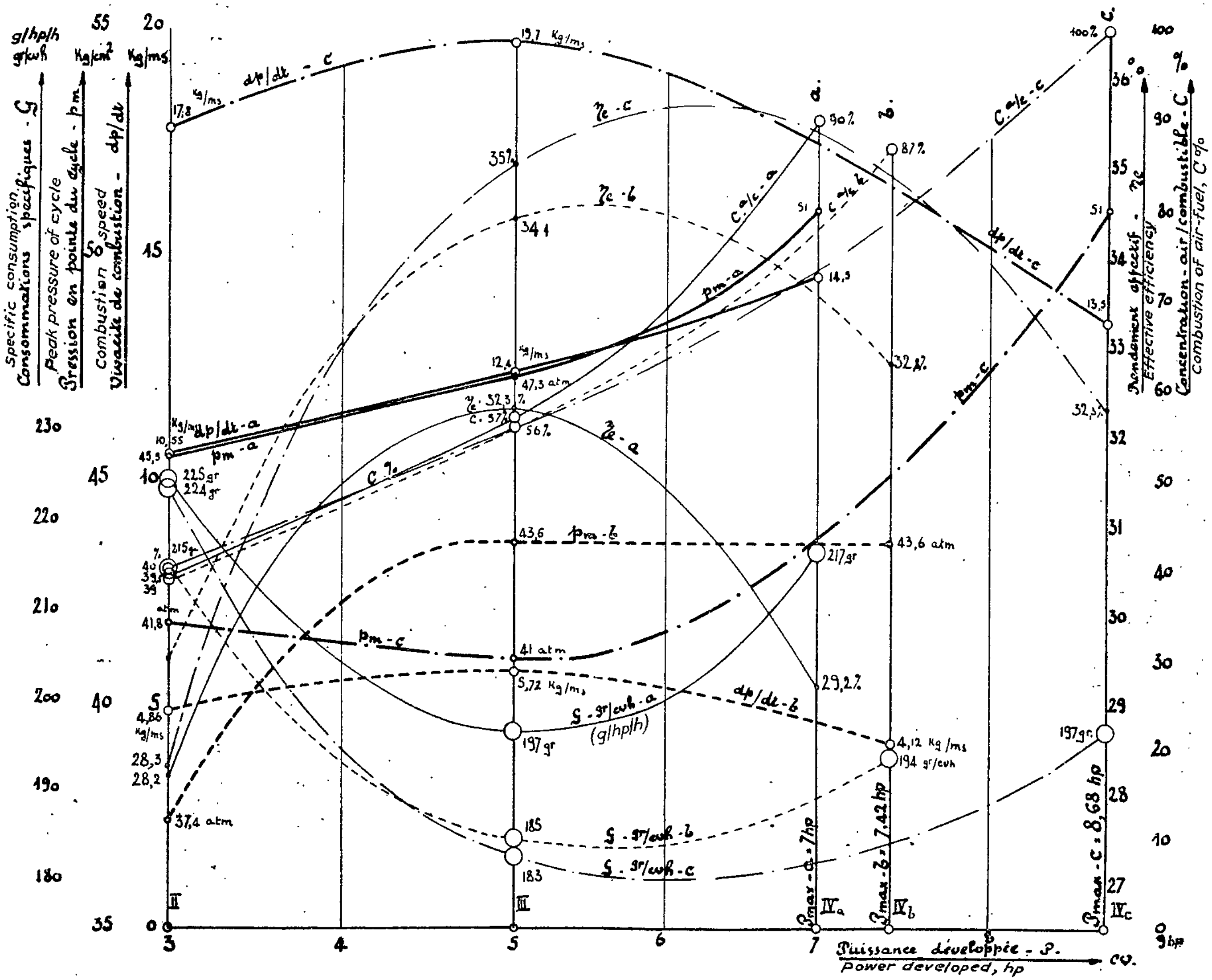


Figure 39.- Speed and pressure of combustion-consumption and effective efficiency, concentration of air-fuel plotted against developed horsepower; ρ = 15 .

causes of this phenomenon are unstable; indeed, no two successive cycles recorded with the Zeiss indicator ever match exactly. Not only may the peak pressure vary 2 or 3 atmospheres, but the rapidity of the pressure variations also always show some difference. These differences are the result of a change in the way the spray penetrates the fire chamber; the flow of the fuel being equally turbulent, makes this possible. A more searching experimental analysis will perhaps show perfect accord between resonance of the pressure of combustion and that of the flow of the injection.

For the present, it can only be said that the amplitude of resonance, for equal engine charge, increases with the rapidity dp/dt of combustion. It reaches 7 atm for $\frac{dp}{dt}$ = 19.5 kg/ms, graph I-c, and drops 50 percent for $\frac{dp}{dt}$ = 17.8 kg/ms, graph II-c.

We also see that the amplitude of these resonances diminishes with the engine charge (graph IV-c). Their frequency, on the other hand, always seems to be around 3000 periods per second. Furthermore, it appears as if these phenomena of turbulence in combustion were the cause of the noisier running of the engine.

Plate 38 is a graphical representation of the quickness of the explored cycles.

Variations in Quickness of Combustion

5. Quickness of combustion is, as already stated, inversely proportional to the delay in the combustion and to the period of injection (curves dp/dt a, b, and c).

6. For an advanced combustion cycle (Bosch unit, curve dp/dt a)), rapidity increases with the engine charge.

7. For a delayed combustion cycle (units b) and c)), rapidity starts on a slight increase of the charge, then drops quickly with the supercharge.

To illustrate: At 8.7 hp (point IV-c), the rapidity is only 13.5 kg/ms, whereas it reaches 17.8 kg/ms at 3 hp (point II-c) and 19.7 at 5 hp (point III-c).

This feature of the delayed cycle is, we believe, of utmost importance for high-speed Diesel development.

In effect, up to now, high-speed Diesel research has run against this "rock" of quickness of combustion increasing with the engine charge. Mechanical accidents due to the abnormal violence of the cycle, have forced constructors, in their attempts to obtain a longer and more progressive combustion, either to increase the period of injection, or resort to preignition, which consists in atomizing a small amount of fuel, in the operating chamber, several degrees before the principal injection.

These artifices invariably reflect on the power developed by the engine, as demonstrated on two engines of different types and of absolutely different design; the specific horsepower of a Diesel engine is inversely proportional to the injection period.

It is here, we believe, that we must try to find the reason why the Diesel engine has not afforded the advantages which were, by rights, expected.*

It seems to us that this situation might be modified in favor of the Diesel by using the cycle of delayed combustion which permits, as seen, higher speeds and pressures of injection without increasing the peak of the cycle, while occasioning a reduction in the rapidity of combustion in relation to the charge.

This peculiar quality of the delayed cycle might, it seems, furnish a solution for the best specific horsepower and best effective efficiency, without fear of dangerous fatigue.

For the present, it appears difficult to exactly define the relation of cause and effect which explains this decline of quickness in function, of the charge of the engine.

---

*La Technique Automobile et Aérienne, No. 188, February 1939: "It has been established that, for engines of equal displacement and weight, the Diesel cycle supplies 33 percent less power than a carburetor engine, but its fuel consumption is also 15 percent less. This difference in consumption between two engine types, assumes importance only when it involves distances of from 2500 to 3000 km; but as a result of the high supercharging adopted in modern aircraft, the power necessary at take-off renders flight impossible."

Three conditions tend toward this object:

1. A greater amount of heat employed for the vaporization of a greater fuel charge.

2. A longer period of injection, which extends the period of combustion.

3. A much greater and faster volume V of the chamber during combustion.

The delayed cycle alone possessing this characteristic, the third condition is quite certainly dominant.

Peak-Pressure Variations of the Cycle

8. In an advanced cycle, the maximum pressure rises with the engine charge (curve p_m-a).

9. In a retarded cycle the peak pressure is a function of the advance of combustion (curves p_m b and c).

Even for high speeds and pressures of injection, the peak pressure of the retarded cycle can always be maintained within acceptable limits, by regulating the delay in injection in relation to the engine charge (point IIIc, for example; injection at T.C.; delay of combustion 16° 7' peak pressure 41 atm).

Fuel Consumption, Power, and Effective Efficiency of an Engine with Delayed Cycle

10. The specific fuel consumptions and the horsepowers developed are improved by the use of the retarded cycle; they always follow the laws outlined in sections on pages 37 and 40, and appear to be little affected by a greater or lesser delay in combustion.

11. At low engine charge the efficiency and, hence, the specific consumption of the engine are improved by a less pronounced rapidity of combustion (curves η_e and G, points IIb).

Concentration - Air-Fuel Ratio

12. With supercharge, the possibility of concentration of the fuel-air mixture under favorable conditions of

thermal efficiency increases with the speed of injection (curves a/c, a, b, and c) (fig. 39).

This statement is verified in an absolutely general manner for all systems of injection and for all combustion cycles used. Therefore, when the Bosch pump is controlled by the regulator, the engine speed drops slowly with the charge. It was found that, under these conditions, the Lister engine developed 6 hp at 600 rpm, supercharged 20 percent of the rated horsepower, with a concentration of 90 percent and a specific consumption of 225 g/hp/hr. The injection period is then 10 milliseconds. If, under those conditions, the rack of the pump feed is pushed to the bottom, the engine does not change speed - neither does the power of 6 hp rise any further; only the specific fuel consumption increases and the exhaust becomes smoky. Contrariwise, if the rise in charge occurs while its speed of 700 rpm is maintained, by manual regulation of the pump feed, the power by supercharge reaches 7 hp (curve a). The injection period at 700 rpm is not more than 8.58 milliseconds, and for a 90-percent concentration the specific consumption has dropped to 217 g/hp/h, while the supercharge is 40 percent instead of 20.

It is seen, on the other hand, that for a 1.5 millisecond injection period, the supercharge reaches 75 percent (point IVc) with a specific consumption of 197 g/hp/h, a 32.3-percent efficiency, and 100-percent concentration; the theoretical mixture is 14.5 parts of air to one part of fuel.

This 100-percent saturation cannot be obtained under adequate conditions of engine efficiency except by the use of the retarded combustion cycle.

The effect of a delayed combustion on the possible concentration of the fuel-air mixture and, consequently, on the specific horsepower per liter of displacement, can be explained, as seen in section on Injection Advance and Combustion Cycle, page 48, by the chemical theories of the variation of the equilibrium factor, though a physical interpretation of the phenomenon seems evident and leads, in any case, to the same conclusions.

In fact, if the combustion starts as usual, before top center, it continues in a gas in active compression; the movement and turbulence of this gas tend to become slower, to the detriment of a good contact of air and fuel.

The combustion is completed before the total fuel is burned. In the delayed cycle, on the other hand, combustion starts and continues in an atmosphere of vivid expansion - that is, to say, of increasing turbulence, thus assuring better contact of air and fuel. The quickness of the combustion should be greater, but it is largely compensated by the increased chamber volume, so that the concentration is able to reach its theoretical limit of saturation easily.

Thus, without having investigated it as yet, it seems logical that the advantages - greater horsepower, thermal and mechanical efficiency - obtained by a delayed combustion, should be even more substantial in high-speed engines, in which the greatest linear piston speed permits a more rapid increase in the volume V_a of the working chamber and, consequently, a more active expansion of gases.

Operating Noises

Obviously, the operating noise of the Dièsel is a function of the quickness of combustion. To reduce the noise, a delayed combustion is therefore indicated. The rate of injection could also be reduced but that would be to the detriment of the horsepower developed.

The Lister engine equipped with pump-injector unit b (one nozzle, 0.20 mm diameter) has remarkably smooth operation at all loads; the developed horsepower has a limit of 7.42 horsepower.

The same engine fitted with unit c (eight nozzles of 0.20 mm diameter), on the contrary, runs roughly, especially at low loads; at a maximum horsepower of 8.7, the operation is less noisy than at 5 horsepower, which proves that the quickness of the retarded cycle decreases when the concentration increases.

The nozzle sprays being toward the axis of the turbulence - that is, toward chamber B - the engine runs more quietly than if directed at 90° or 180°. The position of the nozzles, pointing toward chamber B, also gives the best valmes at the test stand. A symmetrical spray distribution in chamber A is accompanied by harder running and less favorable consumption values.

In short, it does not seem possible to get the utmost from fuel-injection engines if this demand is accompanied by silent operation. Some choice must be made between smooth running and smaller supercharge, and hard running under optimum conditions of specific horsepower and efficiency.

Translation by J. Vanier,
National Advisory Committee
for Aeronautics.