Performance and Combustion in Gasoline Injection Engine

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Introduction

In gasoline injection spark ignition engine, two types are defined by the charging method. The one is cylinder injection and the other is manifold injection. The purpose of the present work is to determine the engine conditions, which would give the best performance for each fuel system. Another object of the test is to clarify the combustion properties in gasoline injection engine and to compare them with the results obtained in carburetor system.

1. Performance of Cylinder Injection Engine

A. Apparatus and Method

The single cylinder four stroke engine used in this test has a bore of 90 mm, a stroke of 120 mm, and a compression ratio of 6 : 1, and is rebuilt from a Diesel engine. The cylinder head is shown in Fig. 1, and tests can be made varying the location of the injector and the spark plug.

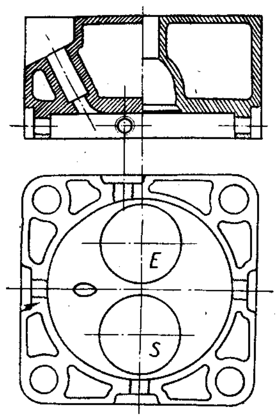

Fig. 1. Cylinder head
E : Exhaust valve
S : Suction valve

A Junkers injector for Diesel engine is used throughout the tests. The injection pump is of a Bosch type, having a plunger of 10 mm diameter. When the engine is running with carburetor, a Carter carburetor was applied.

Throughout all the tests, the pressure change in the cylinder was recorded by a photo-electric cell indicator. To obtain an indication of the mixture ratio supplied to the engine, an Orsat's gas analyser was used.

The gasoline is a regular commercial one for automobile use. (its specific gravity being 0.73 at 23°C)

B. Test Results

(a) The location of the injector and the ignition plug for the best performance.

With cylinder injection system, the distribution of fuel vapour at the moment of spark is not uniform as pointed out in the foregoing report.[1] Therefore, the location of the injector and the spark gap is of the most importance. First, in order to determine the best locations of the injector and spark plug, several arrangements as shown in Fig. 2 were tried at the engine speed of 1500 rpm and the start of injection of 60 crankshaft degrees after TDC. These investigations show that the best performance can be obtained with the arrangement (9), in which the spray is directed horizontally across the combustion chamber from the exhaust side towards the inlet valve.

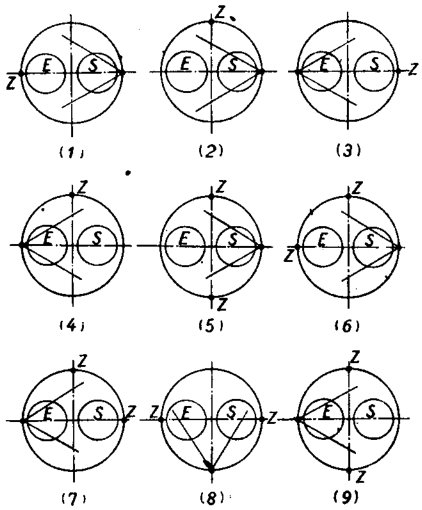

Fig. 2. Direction of fuel spray and position of spark gap.
E : Exhaust valve S : Suction valve
Z : Ignition spark
(1) misfired
(2) high exhaust gas temperature
(3) misfired
(4) high exhaust gas temperature
(5) maximum output 7.9 HP
(6) maximum output 8.0 HP
(7) 〃 8.2 HP
(8)
(9) 〃 8.4 HP

Further, in order to check the performance of the engine using the Bosch type injector, it was tried with several kinds of nozzles and spark gap locations, varying the injection pressure and the start of injection. The desirable performance, however, could not be obtained.

As the dispersion characteristic of gasoline is better than that of heavy oil, the injectors for Diesel engine can be applied successfully as far as the dispersion is concerned. The direction and distribution of fuel spray is especially of importance and it should be determined taking the air turbulence and the location of the spark plug into consideration. The experiments show that the desirable form of spray is not obtained for any spark locations and the favourable mixture can not be formed at the neighbourhood of the spark gap, when the Bosch type injector is used.

The better performance is obtained by spraying the fuel from exhaust side towards inlet valve than the opposite arrangement. Spraying the fuel in this direction has following merits. As the spray penetrates the relatively hot air existing at the neighbourhood of exhaust valve and moreover it is directed right against the stream of incoming air, it makes the heat transmission to the droplets of fuel effective and promotes the vaporization. Further, the result shows that the disturbance near the injector is not effective on the mixing of fuel vapour and air.

As shown in Fig. 2, it is unfavourable to locate the spark gap opposite to the injector, because it is wetted by the spray.

(b) Comparative performance of cylinder injection and carburetor system.

After the preliminary tests described above, the engine performance was investigated in detail for both fuel systems over a range of engine speed from 1000 to 2000 rpm. In this test the time of spark and the charging quantity of fuel were controlled so as to obtain the best performance.

In Fig. 3, the performance characteristics for both systems are shown. At the practical speed range, slightly higher output is obtained with cylinder injection than with carburetor, and the difference is clearer at higher speed. This increase is caused by the following reason. In carburetor system, the engine loss increases, owing to the resistance of air flow through the throat of the carburetor and the decrease of inlet air due to the pressure drop. As the running speed is increased, the resistance of air flow through the throat of carburetor increases and the difference of engine loss in the two types becomes remarkable.

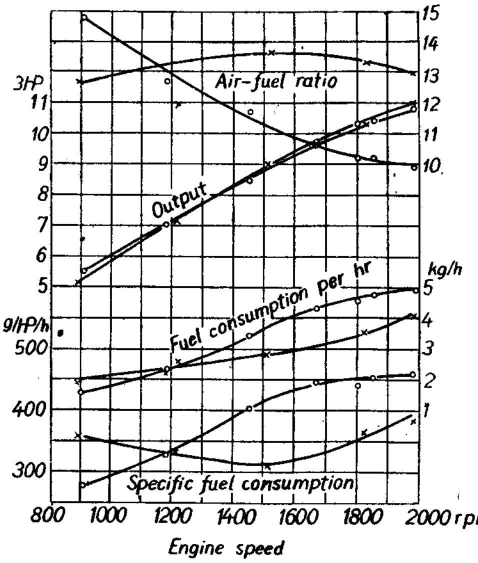

Fig. 3. Performance characteristic curves for each fuel system
o Carburetor system, × Injection system
Throttle: Full open
Injection start: 20° after TDC
Cooling water temperature: 80±12°C

At lower speeds, the fuel consumption is lower with carburetor than with injection into cylinder, whereas at higher speeds, it is reversed. Generally, when the fuel is supplied with carburetor, the quantity of fuel is determined by the velocity of air at the throat. Hence, under the same openings of the throttle valve and mixture regulating valve, the mixture ratio becomes rich as the engine speed increases. Therefore, if it is required to control the mixture ratio according to the running condition, the design and adjustment of carburetor becomes difficult. On the contrary, it is an important advantage of engines with cylinder injection that the quantity of charge is controlled at will independent of engine speed. In carburetor, of course, various devices to remove these defects are adopted in practice, but with the use of injection, the regulation of charge is extremely easy and the diffi-culties encountered in the carburetor can be entirely avoided.

(c) Effect of start of injection.

In the injection system, mixing of fuel with air becomes incomplete as the injection is delayed, and the combustion properties change as pointed out in the foregoing paper.[1] It is important to determine the optimum start of injection. In order to clarify the effect of the start of injection, tests were made varying the start of injection from 2 to 78 crankshaft degrees after TDC.

The experimental results are shown in Fig. 4. The output increases slightly at first as the start of injection is delayed until 40 crankshaft degrees, but it decreases suddenly at the further delay. This fact means that such a late injection does not allow enough time interval necessary for the sufficient mixing of fuel and air, and therefore the combustion becomes unfavourable causing the decrease of the engine output.

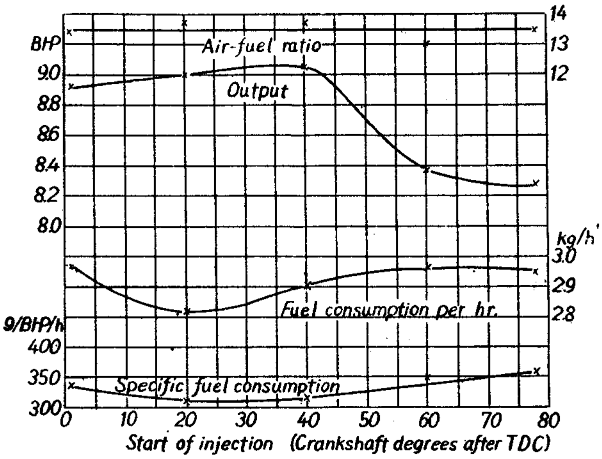

Fig. 4. Effect of injection timing
Engine speed ; 1500 rpm Throttle : Full open
Air-fuel ratio : (13.3±0.4) : 1 Spark advance : 38°
Cooling water temperature : 81±5°C

As shown in the diagram, fuel consumption is minimum when the fuel is injected at about 40 crankshaft degrees after TDC.

(d) Comparison of engine loss of the two fuel systems.

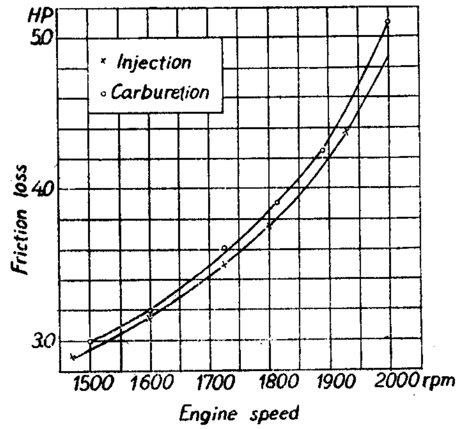

Fig. 5. Friction loss

It is expected that the friction loss is higher with carburetor system than with injection, because the carburetor offers some restriction to the flow of air. In order to compare the friction loss, it was determined by motoring the engine of both types at various speeds. During the test, the temperature of the lubricating oil and the cooling water were maintained at the average in the power runs. The test result is shown in Fig. 5.

2. Performance of Manifold Injection Engine

A. Apparatus

A C.F.R. fuel testing engine with bore of 82.5 mm and stroke of 115 mm was used in this experiment. An injection system was installed and runs were made with manifold injection. A diagrammatic sketch showing the arrangement of the injector is given in Fig. 6. In the case of (A), the fuel spray is directed toward the inlet valve, while in (B), it is assembled to direct the fuel spray against the air flow.

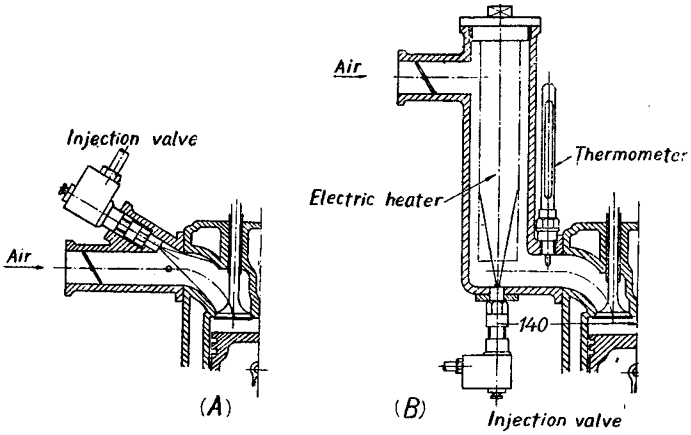

Fig. 6. Arrangement of injection valve

The measuring instruments are the same as in the previous test.

B. Test Results

(a) Selection of injector for the best performance.

At first, several nozzles were tried to determine which is the best for power and economy. Tests were made with injection pressure varied from 50 to 200 atm keeping the start of injection at 60 crankshaft degrees after TDC. It showed that the manifold injection is not so sensible to the injection nozzle as the cylinder injection.

(b) Effect of various conditions on engine performance.

Before the final tests, the effect of various running conditions on the performance was investigated.

(1) Effect of start of injection. Fig. 7 shows the test results. With both injection systems of (A) and (B), the output is maximum when the start of

injection is at about 100 degrees after TDC, and the fuel consumption is not affected by the injection timing. However, in practice, the start of injection has little significance as compared with the case of cylinder injection.

(2) Effect of injection pressure. The results are shown in Fig. 8. When the spray is directed against the incoming air, output is not affected by the injection pressure. But when the spray is directed toward the inlet valve, the lower injection pressure than 100 atm shows a slight decrease in power. The fuel consumption decreases with the increase of injection pressure.

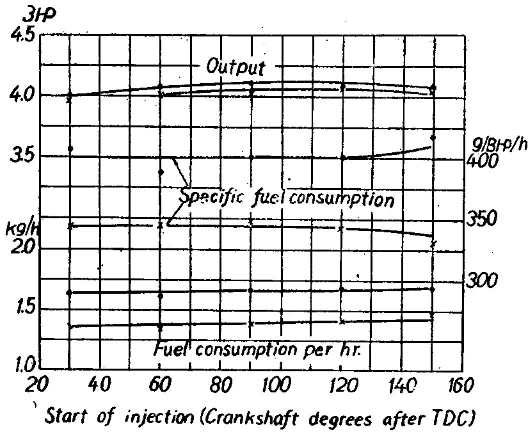

Fig. 7. Effect of start of injection
× Injection system A • Injection system B
Compression ratio: 6:1 Spark advance: 18°
Air-fuel ratio: (11~13):1 Engine speed: 900 rpm

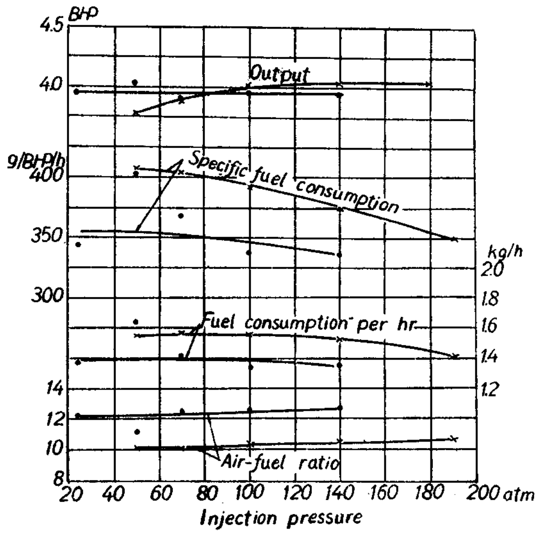

Fig. 8. Effect of injection pressure
× Injection system A • Injection system B
Compression ratio: 6:1 Spark advance: 18°
Injection timing: 60° after TDC
Engine speed: 900 rpm

When the spray is directed against the incoming air, the vaporization of the fuel spray is better than when the fuel is injected toward the inlet valve, because the relative velocity of fuel droplets to the air flow is greater in the former than in the latter. Therefore, the state of distribution and dispersion of fuel spray has little influence upon the mixture performance when the spray is directed against the incoming air. Owing to this reason, output does not decrease even at low injection pressure. But, when the fuel is sprayed toward the inlet valve, the power is affected by the injection pressure, though little, and decreases with lower injection pressure.

(3) Effect of spark timing. Fig. 9 shows the results. The greatest power is obtained with the ignition at 20 crankshaft degrees before TDC, when the manifold injection is used, whereas, with carburetor, the best performance is obtained by the slightly earlier ignition.

(c) Comparative performance for each of the fuel systems.

Finally, the comparative performance with each of the three methods of mixing fuel with air was determined over a range of engine speed from 600 to 1800 rpm. All tests were conducted with full open throttle, and the spark timing and mixture concentration were adjusted to give optimum performance. The performance characteristics are shown in Fig. 10.

Higher output is obtained with manifold injection than with carburetor. The difference of output between the use of carburetor and manifold injection increases as the engine speed becomes high. This is because the restriction of air flow in the carburetor increases as the engine speed becomes high. With manifold injection, output is slightly higher when the fuel is sprayed toward the inlet valve than when injected against the incoming air. When the spray is directed toward the inlet valve, some parts of the fuel droplets will be introduced into the cylinder,

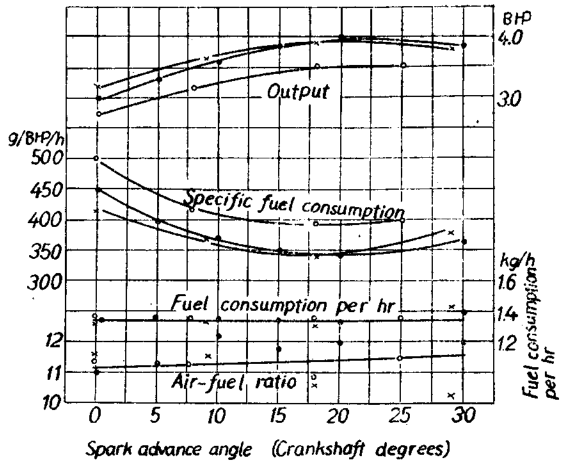

Fig. 9. Effect of spark timing
o Carburetor system × Injection system A
• Injection system B Compression ratio: 6:1
Injection timing: 60° after TDC
Engine speed: 900 rpm

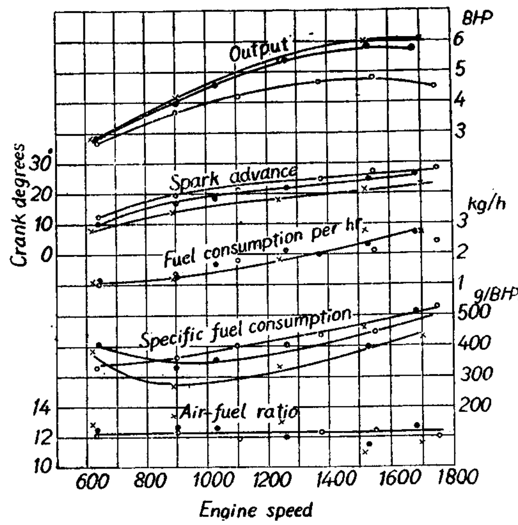

Fig. 10. Performance characteristic curves
o Carburetor system × Injection system A
• Injection system B Compression ratio: 6:1
Injection timing: 60° after TDC

therefore the temperature of the charge will drop owing to the vaporization of fuel, resulting in the increase of mixture induced in the cylinder. The higher output with injection toward the inlet valve is caused by this increase of the charge.

With the use of manifold injection, slightly later ignition is favourable than when carburator is applied. This fact is due to the rapid progress of combustion at the initial stage, when the fuel injection system is used.

The specific fuel consumption is minimum for the injection toward the inlet valve, while it is maximum with carburetor.

(d) Comparison of knocking tendency for each fuel system.

The engine was adjusted at the standard operating condition which is applied in the fuel testing, and the knocking tendency for each fuel system was compared. At first, the original compression ratio, at which the standard knocking occurs, was determined for carburetor system. In the next place, the engine was driven under the strictly same condition, varying only the fuel system, and the critical compression ratio was measured. In this test, the starting period of injection and injection pressure were kept the same as that in the performance test.

The experimental results are listed in Table 1.

From this result, it is obvious that the anti-knock characteristic is most excellent with the cylinder injection, and is worst in carburetor system.

Table 1.

Kind of fuel system	Critical compression ratio at which the standard knocking occurs.
Carburetor	4.7
Manifold injection (A)	*—
Manifold injection (B)	4.8
Direct injection	4.9

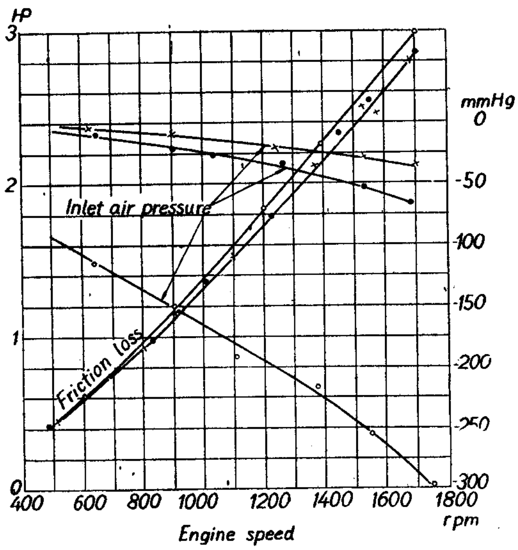

Fig. 11. Friction loss
o Carburetor system × Injection system A
• Injection system B Compression ratio: 6:1

* The standard operating condition can not be obtained owing to the constructive reason.

(e) Comparison of engine loss for each fuel system.

The engine loss was determined by motoring the engine, varying the fuel system. Test was carried out immediately after the power runs. The temperature of the cylinder was kept at the same as that during the performance test by decreasing the supply of the cooling water.

As shown in Fig. 11, it is obvious that the pressure drop at the throat of carburetor increases as the engine speed becomes high, and the increase of the loss follows.

The difference of the loss among the three cases is caused only by the difference of the restriction in the intake system.

3. Rate of Combustion and Thermal Efficiency

A thermodynamic analysis of the indicator cards secured in the previous tests was made to obtain information on the evolution of heat, and the thermal efficiency of gasoline injection engine was discussed.

A. Rate of Combustion

(a) Estimation of the rate of combustion.

Let Q_G be the heat given to the gas in the cylinder due to combustion, then

$$\frac{dQ_G}{d\varphi}=\frac{A}{\kappa-1}\left(\kappa p\frac{dV}{d\varphi}+V\frac{dp}{d\varphi}\right)$$

where,
p = pressure in the cylinder
V = cylinder volume
φ = crank angle
A = heat equivalent of mechanical work
κ = adiabatic exponent

In this equation, if the relation between p and φ is known, $dQ_G/d\varphi$ can be calculated easily. The value of $dQ_G/d\varphi$ indicates the rate of the effective heat supplied to the gas in the cylinder, and will be called the effective rate of combustion in this paper.

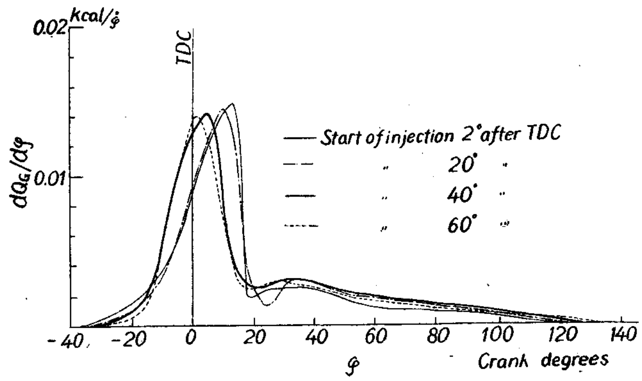

Fig. 12. Variation of combustion rate with start of injection

(b) Effective rate of combustion.

The effective rate of combustion was calculated using the indicator cards obtained in the previous performance tests.

(1) Cylinder injection. Fig. 12 shows the effective rate of combustion in the cylinder injection system as affected by the injection timing. The results are plotted in Fig. 13. $(dQ_G/d\varphi)_{max}$ decreases and its presenting period becomes earlier as the start of injection is delayed. However, with early start of injection, the descent of $dQ_G/d\varphi$ after its maximum is rapid, the completion of combustion becomes early and therefore after-burning is reduced.

(2) Manifold injection. The effective rate of combustion is shown in Fig. 14. The earlier the injection, the more reduced the value of $(dQ_G/d\varphi)_{max}$, when the spray is directed against the flow of the incoming air. However it remains almost constant with the injection toward the inlet valve. For both injection systems, the earliest occurrence of $(dQ_G/d\varphi)_{max}$ takes place when the start of

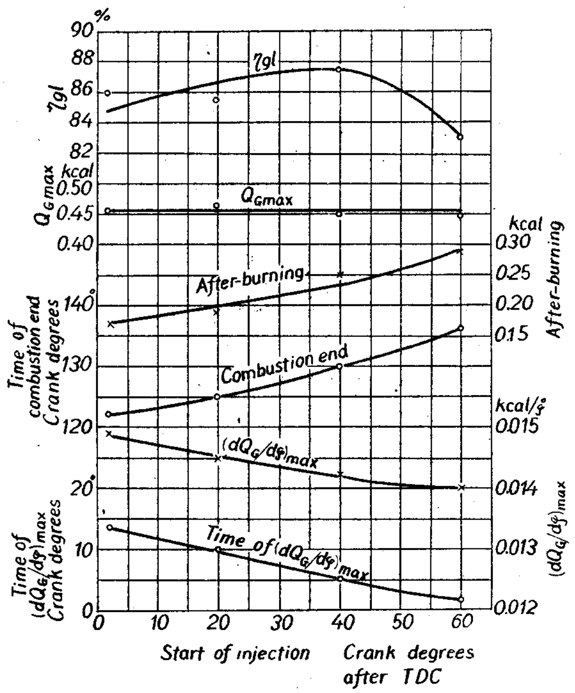

Fig. 13. Effect of injection timing on effective rate of combustion and equivolume degree
Q_{Gmax}: Total heat evolution

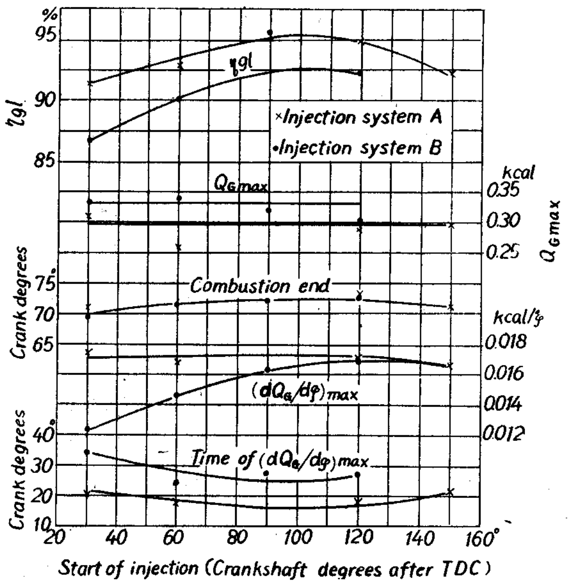

Fig. 14. Effect of injection timing on effective rate of combustion and equivolume degree

injection is at about 100 crankshaft degrees after TDC. The period of the end of combustion is almost the same for the different start of injection, and no difference can be observed for both fuel systems. Generally, the rate of combustion is not affected by the starting time of injection so much with manifold injection as with cylinder injection.

B. Thermal Efficiency

(a) Relation between effective rate of combustion and thermal efficiency.

In an actual cycle, as the whole of the charge cannot be burned instantly, the influence of the combustion time on the thermal efficiency must be investigated. Assume that the polytropic exponent m is the same for compression and expansion stroke, and let η_φ be the thermal efficiency of the elementary Otto cycle as indicated in Fig. 15, then

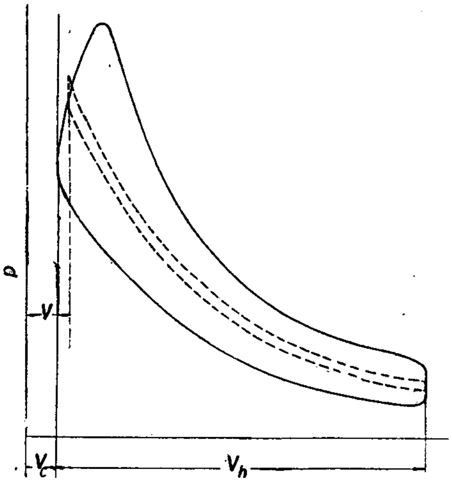

Fig. 15.

$$\eta_\varphi = 1-\left(\frac{1}{\varepsilon_\varphi}\right)^{m-1}$$

where,

$$\varepsilon_\varphi = \frac{\text{clearance volume } (V_c)+\text{stroke volume } (V_h)}{\text{cylinder volume at any crank angle } \varphi(V)}$$

Therefore, the thermal efficiency of the actual cycle is

$$\eta = \frac{1}{Q_{G\,max}}\int_{\varphi_0}^{\varphi_e} \eta_\varphi\left(\frac{dQ_G}{d\varphi}\right)d\varphi$$

where,

$Q_{G\,max}$ = total amount of heat supplied per cycle

φ_0 = spark advance angle

φ_e = period of the completion of combustion

Now, let η_{th} be the theoretical indicated thermal efficiency, then η/η_{th} indicates the influence of the rate of combustion on the thermal efficiency in an actual engine. It is called equivolume degree and is denoted by η_{gl}. It is expressed as follows:

$$\eta_{gl} = \frac{1}{Q_{G\,max}}\int_{\varphi_0}^{\varphi_e}\left(\frac{dQ_G}{d\varphi}\right)\left\{\frac{1-(1/\varepsilon_\varphi)^{m-1}}{1-(1/\varepsilon)^{m-1}}\right\}d\varphi$$

where, ε indicates the compression ratio.

(b) Equivolume degree in the gasoline injection engine.

Using the experimental data of the effective rate of combustion obtained in the foregoings, the equivolume degree in the gasoline injection engine was investigated. In the calculation, the polytropic exponent m is assumed to be 1.3, which is the mean value estimated from the actual cycle.

(1) Cylinder injection. In Fig. 13, η_{gl} is plotted against the starting period of injection. As the experiment was carried out by changing the injection timing only, the value of $Q_{G\,max}$ is constant. But the value of η_{gl} is varied due to the variation of the process of heat evolution. As shown in the diagram, η_{gl} is maximum when the injection starts at 40 crankshaft degrees after TDC. Owing to this fact, the maximum output is obtained with this injection start. The fact that such an improvement of η_{gl} is realized in a slightly later start of injection results from the quick extent of combustion as pointed out in the previous paper.[1] With too delayed start of injection, however, the after-burning increases, and η_{gl} decreases. From this point of view, the start of injection at 40 crankshaft degrees after TDC is most favourable in the cylinder injection system.

(2) Manifold injection engine. In the manifold injection system, η_{gl} is higher for the injection toward the inlet valve than injection against the incoming air as indicated in Fig. 14. With carburetor system η_{gl} is small, which causes the higher specific fuel consumption.

Further, it is maximum with the start of injection at 100 crankshaft degrees after TDC, because the period at which the effective rate of combustion reaches maximum becomes earliest at this start of injection. For this reason, the high output can be obtained with the start of injection at 100 crankshaft degrees after TDC.

Summary

The test results may be summarized as follows:

(1) Slightly higher output can be obtained with cylinder injection system than with carburetor system. The carburetor offers restriction to the flow of air, therefore the engine loss is higher in carburetor system than in cylinder injection.

(2) When carburetor is used, the amount of fuel supply depends inevitably on the velocity of the incoming air in the throat of the carburetor, it increases as the engine speeds up. Whereas, with cylinder injection engine, the mixture strength can be controlled independently under any speed. Thus the latter enables the economical driving at any speed.

(3) The interrelation of the direction of the spray and the position of spark plug is of the most importance in cylinder injection engine, which should be determined taking the air turbulence into consideration. It is more effective to disturb the spray at a distance from the nozzle than near the injector; that is, it is desirable to spray the fuel from the exhaust valve side toward the inlet valve.

(4) With the cylinder injection, the start of injection of 40 crankshaft degrees after TDC on the inlet stroke is desirable. The sudden decrease in output follows with later start of injection.

(5) With manifold injection, better performance can be obtained than with carburetor system. In this fuel system, the direction and shape of spray, injection pressure, and the start of injection have not so serious effect on performance as in the cylinder injection. When the spray is directed against the incoming air, good performance can be obtained even at low injection pressure. The start of injection of 100 crankshaft degrees after TDC on the inlet stroke is optimum.

(6) Fuel injection is superior to carburetion with respect to antiknock characteristic. The cylinder injection is the best.

(7) In the injection system, slightly later ignition is favourable as compared with carburetor system.

Reference

1) Fujio Nagao, Syun'iti Ohigashi: Memoirs of the Faculty of Engineering (Kyoto University) Vol. XIV, No. 1 (1952)
Ditto: Ditto Vol. XIV, No. 2 (1952)