A Fuel Injection System for Diesel Engines by Injection Pressure Control
1st. Report, Theories, Influence upon Theoretical Thermal Efficiency and the Result of Basic Experiments

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In order to ensure the efficiency of Diesel engines and to adjust the exhaust gas for environmental requirements, it is desirable to control not only the timing of fuel injection beginning but also the period or duration of fuel injection positively. With the conventional fuel injection systems however, it has been extremely difficult to carry out especially the latter control in all operation modes. The authors, based on a new idea of controlling the amount of fuel injection by means of injection pressure and thereby regulate the injection period freely and at its optimum value in principle, made some basic experiments and obtained satisfactory results on the whole. The results will be reported briefly in the following.

Key Words: Internal Combustion Engine, Diesel Engine, Fuel Injection System, Electronic Fuel Injection Control, Injection Pressure, Injection Period Control

1. Introduction
Conventional fuel injection system based on the reciprocating plunger mechanism has the advantage of keeping the rates of fuel injection and heat release theoretically constant referring to the crank angle of the engine. Its function is quite stable. Therefore, it has been used widely for Diesel engines. However, it is built up of intricated parts precision machined, and there have been many problems in its maintenance to prevent the lowering of performance due to aging and abrasion. It can not satisfy the requirements for sophisticated fuel injection control from the viewpoints of energy saving and environmental pollution. Since the reciprocating motion is imparted from the cam shaft which is driven by the crank shaft of the engine, it is difficult to provide a variable phase difference between the angular positions of both shafts and thereby control the injection beginning point, and thus the control of the injection period is impossible. Besides, it is also difficult to give satisfactory solution to the problem of intermittent injection in low speed ranges and secondary injection in high speed ranges. Based on a new idea of applying electronic control of fuel injection wherein the injection quantity is controlled by the injection pressure and thereby not only the beginning but also the period of injection will be freely set in principle, the authors carried out a series of basic experiments. As the results were satisfactory on the whole, essential points will be described in the following.

2. Principle
Currently used reciprocating pumps control the injection quantity by changing the effective plunger stroke or the length of injection period. Therefore, it is impossible to adjust the injection period itself irrespective of the injection quantity. From this point of view, we tried to regulate the quantity by varying the injection pressure. The injection pressure $P_i$ or the pressure difference between the nozzle and the cylinder required to inject the fuel of quantity $q$ and density $\rho$ through time $t$ from a nozzle of area $A$ and the coefficient of discharge $C_d$ is given by

$$P_i = \frac{\rho q A^2 t}{2CA_d}$$

(1)

The injection period $\phi [\text{CA}]$ referred to the crank angle, time $t$ [s] and engine speed $N[\text{rpm}]$ are correlated by

$$t = \frac{N}{360} \phi$$

(2)

From Eq. (1) and Eq. (2), $P_i$ is written as

$$P_i = \frac{\rho q A^2}{C_d \phi t}$$

(3)

In other words, Eq. (3) gives the required injection pressure to be set in order to inject the fuel quantity $q$ per cycle at the engine speed $N$ for the injection period $\phi$.

When the injection pressure is controlled as above, combustion in the optimum injection period can be made at any engine speed and injection quantity. Such optimizing control has been practicable.
for the conventional systems only in specific mode (at the maximum output in most cases), contrary to the expectations over so many years.

The optimum value of injection period differs according to the type and output of the engine. It should not be regarded as a matter simply dealt with here. Therefore, detailed discussion of the above control will be deferred to some future occasion. Instead, we will describe a control method for keeping \( \phi \) constant irrespective of the injection quantity and engine speed, as an example of controlling \( \psi \) positively by means of adjusting the injection pressure based on our principle.

Since \( \psi \) is constant in this case, \( P_1 \) in Eq. (3) will be

\[
P_1 = K - \psi + N^2
\]

where \( K \) is a constant. Under this condition, it is assumed that \( \psi_1, \psi_2 \) and \( \psi_3 \) are the quantities of injection at high, medium and low levels of injection respectively. Then the relation between the corresponding injection pressure \( P_{1i}, P_{12}, P_{13} \) and the engine speed \( N \) will be given by the curves in the upper part of Fig. 1.

![Fig. 1 Injection quantity controlled by injection pressure (in case of \( \psi = \text{constant} \)](image)

3. Effects on Thermal Efficiency

3.1 Comparison of both systems in general

The difference between the conventional injection system and ours with regard to thermal efficiency will be described in the following. Fig. 2 shows a change in the shape of indicator diagram due to the variation of injection quantity \( V \) in the conventional injection system. As shown in Fig. 2, the rate of explosion \( P_{1/2} \) is originally constant irrespective of the injection quantity, and only the cut-off ratio varies, such as: \( V_{oh}/V_e \) at high level, \( V_{oh}/V_e \) at medium level and \( V_{oh}/V_e \) at low level of injection respectively.

![Fig. 2 Indicator diagram of conventional injection system](image)

The rate of explosion in our system varies in response to the injection quantity. The cut-off ratio can also be set freely by regulating the injection period. Consequently, the formation of indicator diagram becomes very elaborate. For example, when \( \psi \) is kept constant as in Fig. 1, the cut-off ratio \( V_{oh}/V_e \) can also remain constant in principle irrespective of the injection quantity as shown in Fig. 3, and the rate of explosion \( P_{1/2} \) only varies in response to the injection quantity.

![Fig. 3 An example of indicator diagram for injection pressure control system](image)

Next, the change of theoretical thermal efficiency due to the difference between the injection controls of both systems will be compared with the aid of generally accepted formulae and constants. The condition of comparison are based on
JNR type DMH 17C engine for Diesel car, specifications of which are given in Table 1. The conditions are: the compression ratio = 16, the rate of explosion = 17 and the ratio of specific heat = 1.4.

Table 1 Specifications of JNR Diesel engines for railway cars

<table>
<thead>
<tr>
<th>Type</th>
<th>DMH17C</th>
<th>DMH15SH</th>
</tr>
</thead>
<tbody>
<tr>
<td>No. of cylinder</td>
<td>120 x 160</td>
<td>140 x 160</td>
</tr>
<tr>
<td>Bore x stroke</td>
<td>168 x 200</td>
<td>150 x 200</td>
</tr>
<tr>
<td>Displacement volume (cc)</td>
<td>16980</td>
<td>14450</td>
</tr>
<tr>
<td>Pre-combustion chamber</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Fig. 4 Cut-off ratio and theoretical thermal efficiency in conventional injection system

As shown in Fig. 4, theoretical thermal efficiency $\eta_t$ in the conventional system decreases as the cut-off ratio (injection quantity) $\beta$ increases, and $\eta_t$ falls down to its lowest value at the highest injection ($\beta = \text{maximum}$), where the highest efficiency should be obtained. In our system, when the injection quantity is controlled only by the injection pressure at the constant cut-off ratio $\beta$ (injection period $\alpha$), $\eta_t$ shows only a minute change irrespective of the rate of explosion $\alpha$ (injection quantity) as shown in Fig. 5.

Fig. 5 Rate of explosion and theoretical thermal efficiency in injection pressure control system

As above described, the conventional system gives good efficiency at low injection levels. In our system, when the maximum value of $\beta$ is set equal to that of the conventional system ($\beta = 1.7$) because of the limitation to maximum combustion pressure $P_{\text{max}}$, it is necessary to take the same value of $\beta$ as that of the conventional system ($\beta = 3.0$) in order to maintain the injection quantity equal to that in the conventional system. Under such circumstances, $\eta_t$ can only be equal to the lowest value of the conventional system.

3.2 Problems in the conventional system

When viewed only from the aforementioned results, even though the efficiency of conventional system is the lowest at the maximum output, it seems that the system is the best injection control theoretically applicable to Diesel engines and that nothing is left for improvement, so far as the value of $\alpha$ is limited to the maximum combustion pressure $P_{\text{max}}$.

In the actual engines, however, recorded specific fuel combustion does not always coincide with the theoretical value, and there are not a few examples of recorded fuel consumption turning out higher than that at high output. One of the causes for such contradiction is the difficulty of delicate control or correction of injection angle to compensate for the complex ignition lag by means of the current mechanical devices. Another cause may be attributed to an occasional fall of the rate of explosion at low injection level, which should be constant irrespective of the injection quantity. This is shown in the theoretical indicator diagram in Fig. 6, where the part of the diagram which should remain on the line CD giving $P_{\text{max}}$ at high injection level falls from C'D'C' at low injection level. In this way, the decrease of the rate of explosion and the accompanying increase of the cut-off ratio jointly decrease the thermal efficiency.

Fig. 6 Decrease in rate of explosion at low injection level in conventional system

In order to prevent the loss in thermal efficiency at low injection level because of the aforementioned reasons, it is necessary to keep $P_{\text{max}}$ at the highest value in the permissible range, even in the case other than high level injection. It is well known that the common method to expect an improvement in $P_{\text{max}}$ without changing the injection quantity is to increase the heat release rate or the fuel injection rate referring to the crank angle. For most engines, the injection rate is specified on the basis of $P_{\text{max}}$ at maximum injection rate. Therefore, some allowance for the rate is left in case of the low injection level. In such a case, an increase in the injection rate is effective to prevent the drop of $P_{\text{max}}$. However, the control of this kind has been difficult with the conventional systems.

In our system, both the injection pressure and the injection period can be freely set in principle. Therefore, it seems quite promising to prevent the loss in theoretical thermal efficiency $\eta_t$ by means of increasing the injection rate at the low injection level and thereby preventing the drop of $P_{\text{max}}$. 
4. Problems in the Practice of Injection Pressure Control

In the following, fundamental problems in the practice of injection pressure control based on the principle in Sec. 2 will be discussed.

The theoretical injection pressure $P_i$ in our system must be varied in proportion to the square of engine speed ($N^2$) as is given by Eq. (3), when $\phi$ is kept constant. However, as is generally known, there is a lower limit $P_{\text{m}}$ to the injection pressure. The upper value $P_{\text{max}}$ should also be limited because of the capacity of the hydraulic devices. In case of the engines for railway cars, the highest speed $N_{\text{max}}$ is about four times the lowest speed $N_1$. Therefore, if $P_i$ is set at 120 MPa for $N_{\text{max}}$, $P_i$ for $N_{\text{max}}$ should theoretically be set at about 190 MPa (120 MPa x 4). However, it is difficult to obtain such a high pressure by means of the current hydraulic devices, and the group pattern of the injection pressure as shown in Fig. 1 can not be optionally established.

We here assume the practically available values of $P_{\text{m}}$ and $P_{\text{max}}$ as above, and consider the way to make the same control as in Fig. 1, for the following case.

![Fig. 7 An example of controls when the injection pressure is limited](image)

When the injection pressure $P_i$ at the low injection level coincides with $P_{\text{m}}$ at $N_1$ and the pressure $P_i$ at high injection level coincides with $P_{\text{max}}$ at $N_1$ respectively as shown in Fig. 7, $P_i$ should be fixed at $P_{\text{m}}$ in the range of speeds lower than $N_1$ and at $P_{\text{max}}$ in the range higher than $N_1$. In these speed ranges, the injection control can not be done by means of the injection pressure. It can be done only by means of varying the injection period $\phi$.

In such cases as above, when the high level injection is maintained at a speed $N_1$ higher than with a specific injection period $\phi_i$ obtained by shifting the injection end $\theta_1$ backward to $\theta_2$, referring to the crank angle $\theta$ since the time $\phi$ and the period $\phi_i$ of injection at $N_1$ are correlated by Eq. (2) or $\theta_1=\theta_i-F(N-N_{\text{max}})+N\phi/N_1$ by the equation

$$\theta_1=\theta_i-F(N-N_{\text{max}})+N\phi/N_1$$

Thus, as shown in Fig. 7, control may be done through varying the injection end by the amount $\theta_2$, given by Eq. (5) for high level injection in the speed range $N\geq N_1$, and by $\theta_2$ of Eq. (6) for low level injection in the range $N<N_1$.

Our injection control appears quite complicated from the above description. However, the injection period at speeds lower than $N_1$ or higher than $N_2$ is identical with that at $N_1$ or $N_2$ respectively. As the pattern of $\theta_1$ or $\theta_2$ in case of setting back the injection end at such speed is a group of straight lines converging at a point $\theta_i+F\theta_{\text{m}}$ on crank angle at $N=0$, the processing in the electronic control to generate the pattern of this kind is nothing difficult.

In the speed range where the injection pressure is at the limiting value, the control to set the value of $\phi$ also becomes impossible, and the characteristic features of this system can no more utilized. However, it must be mentioned that, in the conventional system, intermittent or secondary injection used to take place in those speed ranges. Therefore, elimination of such irregular injection, especially the controlling of intermittent injection when starting the engine or at low speed running of the engine will be of great importance.

5. Experimental Equipment

The experimental equipment to operate the engine in our above system is outlined in the following.

5.1 Test engine

A single cylinder experimental engine based on the JR type DMH 15HS, specifications of which are given in Table 1 is directly coupled to a DC motor controlled by Ward-Leonard system. The motor is used for starting the engine, absorption of the engine power and speed control.

5.2 High speed change-over valve

The high speed change-over valve is a 3-way control valve with quick response, controlling the supply and stop of high

![Fig. 8 Sketch of high speed Change-over valve](image)
pressure fuel to the injection valve. As shown in Fig. 8, it is composed of a 3-way sliding spool valve and a torque motor actuating the spool. Its rated pressure is 24.5 MPa, flow rate 0.015 m³/min and change-over time is approximately 0.2 ms.

5.3. Injection control system

The control system is composed of the following units i.e.: a fuel pump for compressing the fuel to high pressure; an electro-hydraulic conversion valve (termed "servo valve" hereafter) for adjusting the high pressure of fuel to the specified injection pressure; a high speed change-over valve above mentioned for supplying the fuel to injection valve at specified timing; an electronic control unit for detecting and generating the signals of \( P_1 \), \( \theta \), \( \delta \), and \( \phi \) in response to the instructions for engine output. A block diagram of the system is shown in Fig. 9.

The electronic control unit performs the following two functions i.e.: the injection pressure control, the main component of which is a pattern generator for detecting the injection pressure suitable for the engine output and speed; the injection timing and period control for detecting the injection timing corresponding to \( P_1 \), \( N \) and crank angle. Each of these functions is provided with an independent microprocessor, and \( P_1 \), \( \theta \), \( \delta \), etc. are detected for every stroke. \( \phi \) can be controlled either in connection with or independently of the output instruction and \( P_1 \).

Various means are available for the regulation of fuel to the injection pressure detected by the electronic control unit. One of them is to couple a variable capacity pump directly to the engine and regulate its delivery. We employed a high pressure servo valve and an independent motor driven fuel pump of large size as a reliable means for easy, free and quick setting of \( P_1 \). However, the means for the efficient control of the injection pressure should also be investigated as an important subject in the future.

In order to secure the pressure fol-

low-up capacity in changing \( P_1 \) and to prevent the pressure drop at injection, accumulators of 1000 cm³ and 100 cm³ were attached respectively to the delivery sides of pump and servo valve.

6. Results and Discussion

In order to confirm the feasibility of operating the engine in the pressure controlled injection system using the above described devices, research on the stability of injection system and basic test for combustion were carried out. The results are described in the following.

6.1 Stability of injection

The injection quantity is controlled by means of the injection pressure \( P_1 \) and period \( \phi \) in our system. Therefore, it seems undeniable that the stability of our system in injections is not always sufficient as compared with the conventional systems using the volumetric injection pumps including the overflow type. Accordingly, we tested the quantity, period and state of fuel injection by spraying the fuel into the open air. Injection valve is an automatic valve with throttle nozzle, which is the same as one on the engine. Its opening pressure is 11.8 MPa, and the injection pipe is 226 mm long.

![Fig. 9 Block diagram of injection pressure control system](image)

![Fig. 10 Electronically instructed injection time and injection quantity](image)

Fig. 10 shows an example of the relation between the electronically instructed time \( t_{inj} \) and the injection quantity \( \Delta Q \) per stroke, corresponding to three values of parameter \( P_1 \) and \( \phi \), and the control of injection quantity in itself involves no problem. However, it must be mentioned that the maximum injection pressure of our equipment is 25.5 MPa and that a considerably long time \( t_{inj} \) or \( \phi \) is taken to obtain the maximum injection quantity (0.16 cm³/cylinder) of the engine.

Fig. 11 shows some examples of the action of the nozzle needle valve in response to the electronic instruction of
injection when \( P_1 \) is 20.6MPa. The state of injection is quite stable and no sign of irregular injection is observed.

\[
\begin{align*}
2.1\text{ms} & \quad 2.9\text{ms} & \quad 3.6\text{ms} & \quad 4.0\text{ms} \\
4.6\text{ms} & \quad 5.4\text{ms} & \quad 5.9\text{ms}
\end{align*}
\]

Fig. 11 State of nozzle needle valve motion following the electronic instruction of injection

6.2 Combustion test

An example of the pressure change in the injection devices and in the engine cylinder recorded during the combustion test is shown in Fig. 12. Test conditions are: \( N = 1200\text{rpm}, P_1 = 15.7\text{MPa}, \phi = 4.2\text{ms}, \) supercharging pressure \( = 0.087\text{MPa} \) and \( \dot{\alpha} = 0.08\text{cm}^3/\text{cy} \cdot \text{cyl} \). In response to the electronic instruction of injection, pressure at the nozzle inlet \( P_H \) and at the outlet \( P_{H\text{out}} \) rise successively, which lead to the actuation of needle valve and the starting of combustion. The lapse of time from the electronic instruction of injection starting to each of these actions changes is given in the figure (except the value estimated for the time of combustion starting), showing that everything is going quite well.

The fall of \( P_H \) following the electronic instruction for the injection end also takes place very quickly. Generation of shock waves is observed in \( P_{H\text{out}} \). But it does not lead to any secondary injection.

6.3 Instructed injection time and actual time

All of the above described injection period and timing were referred to the electronically instructed timing of injection \( \tau_{\text{inj}} \). However, from the test results in Fig. 10 showing that the injection quantity \( \dot{\alpha} \) is not proportional to \( \tau_{\text{inj}} \), we presumed that \( \tau_{\text{inj}} \) or the instructed injection period \( \tau_{\text{inj}} \) would not always coincide with the actual time \( \tau \) or period \( \phi \).

If these instructed values do not coincide with the corresponding actual values, it will be necessary to investigate thoroughly the relation between the instructed values and the actual values, because it is the actual value that has close influence upon the performance of the engine.

In the throttle type nozzle used in our experiment, the state of full injection is not attained until the nozzle needle is actuated to nearly half or more of its lift. Expressing the time or period of full injection by the "Effective injection time" \( \tau_e \) or "Effective period" \( \phi_e \), we investigated the relation between these values and the instructed values of \( \tau_{\text{inj}} \) or \( \phi_{\text{inj}} \) in the same way.

The relation between \( \tau_{\text{inj}} \) and \( \tau_e \) or \( \phi_e \) corresponding to \( P_H \) at \( N = 1200\text{rpm} \) is shown in Fig. 13. The relation between \( \tau_{\text{inj}} \) and \( \tau_e \) or \( \phi_e \) is illustrated in Fig. 14. As shown in these figures, actual injection time or period is considerably shorter than the instruction within this scope. How to compensate for such difference is the subject to be investigated hereafter.

Fig. 12 State of injection and combustion following the electronic instruction of injection

Fig. 13 Electronic instruction of injection time and actual injection time

Fig. 14 Electronic instruction of injection time and effective injection time

6.4 Test of combustion controlled by the injection period -- an example

Finally as an example of displaying the characteristic features of our system i.e., "In principle, \( \phi \) can be freely controlled under any operating condition", data obtained from the combustion tests at constant \( \phi \) within the permissible range of \( P_1 \) are shown in the following.
Fig. 15 Indicated specific fuel consumption in case of controlled injection period

Fig. 15 shows the change of indicated specific fuel consumption $\dot{f}_i$ in relation to $N$, for the injection rate set at about 1/3 (0.055 cm$^3$/cc-yo) of the maximum quantity and with the values of $\phi_m$, $\phi_r$ and $\phi_e$ kept constant. In Fig. 16 are given the changes of $P_i$ and $R_m$ in relation to $N$ with $\phi_m$ = 30°CA shown by a full line $\phi_m$ in Fig. 15.

Fig. 16 Method of injection pressure and period control (for two ranges of engine speed)

As shown in Fig. 16, in the speed range $N > N_i$, $\phi_m$ is kept constant by means varying $P_i$ and $R_m$ while $\phi_r$ and $\phi_e$ are controlled simply in proportion to $N$. If $P_i$ coincides with $P_{in}$ (11.8MPa) in the speed range $N < N_i$, control is done by varying $\phi_m$ and $\phi_e$ with the values of $P_i$ and $R_m$ kept constant, as already shown in Fig. 7.

It is needless to say, in the controls of Fig. 15 where $\phi_r$ and $\phi_e$ are kept constant, controls similar to those of Fig. 16 are done on $P_i$ and others.

Although the change of $f_i$ obtained in the above injection control experiment does not coincide strictly with the theory as shown in the figures, it shows that the influence of the control of $\phi_r$ upon $f_i$ is very important. This can be regarded as verifying our method of injection control. It must also be mentioned that $\phi_r$ among various $\phi$ is the most closely connected with the change of $f_i$, so far as our data of experiment are concerned, and on the whole, the values of $f_i$ are satisfactory.

7. Summary

The results of research on combustion in Diesel engine by means of the injection pressure control are summarized as follows.

1. In principle, optimum injection period can be established freely in any operation mode by means of controlling the injection quantity in terms of the injection pressure.

2. When the injection pressure required for controlling the injection quantity is beyond the range of appropriate values for injection characteristics or performance of the injection devices, proper level of injection pressure can be obtained in almost all modes by means of controlling the injection quantity through adjusting the injection period while keeping the injection pressure within the range of above appropriate values. Such measures are effective to prevent irregular injection.

3. Loss in theoretical thermal efficiency resulting from the drop of explosion pressure experienced at the low injection level in conventional injection pumps of overflow and other types can be reduced theoretically by increasing the rate of injection in our system.

4. Even in the case of high speed Diesel engines with short injection period for railway cars, stable control of injection period can be done by means of controlling the pressure and period of injection.

5. In the basic experiments of an engine employing this system for injection control, the states of fuel injection and combustion were quite satisfactory.

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