An Analytical Study on the Performance Analysis of a Unit-Injector System of a Diesel Engine

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A numerical algorithm is developed to analyze the performance of a Unit-injector (UI) System for a diesel engine. The fundamental theory of the algorithm is based on the continuity equation of fluid dynamics. The loss factors that should be seriously regarded on the continuity equation are the compressibility effect of liquid fuel, the wall friction loss in high-pressure fuel lines of the system, the kinetic energy loss of fuel in the system, and the leakage of fuel out of the control volume. For an evaluation of the developed simulation algorithm, the calculation results are compared with the experimental outputs provided by the Technical Research Center of Doowon Precision Industry Co. (DPICO); the maximum pressure in the plunger chamber ($\boldsymbol{P_p}$) and total amount of fuel injected into a cylinder per cycle ($\boldsymbol{Q_f}$) at each operational condition. The result shows that the average error rate (%) of $\boldsymbol{P_p}$ and $\boldsymbol{Q_f}$ are 2.90% and 4.87%, respectively, in the specified operational conditions. Hence, it can be concluded that the analytical simulation algorithm developed in this study can be reasonably applied to the performance prediction of newly designed UI system.

Key Words : Unit-injector System, Injected Fuel Volume, Plunger Chamber Pressure, Duration of Injection, CAM Rotational Speed, Nozzle Hole Diameter, Bulk Modulus, Discharge Coefficient, Compressibility Factor

Nomenclature

A : Cross-sectional area
C : Discharge coefficient
D : Effective diameter
d : Diameter
f : Friction factor
h_p : Plunger piston lift
K_f : Bulk Modulus of fuel
L : Length
m_f : Mass flow rate of fuel
N : Rotational speed of plunger cam
n : Number of hole
P : Pressure
t : Time
V : Volume
Vo : Injection velocity
β : Compressibility factor of fuel
ρ_f : Density of fuel
δ : Diameter ratio of nozzle inlet to outlet
ζ : Effective diameter ratio

Subscripts

sp, in, out : Spill line, inlet, outlet
N, pp : Injection nozzle, plunger piston
S : Solenoid valve
O : Orifice
01, 02, 03, 04 : Model number

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sv	: Solenoid valve seat
1, 2	: High-pressure line number
$inj, bulk, leak, fric$	: Injected, compressed, leaked, friction loss
cyl, p, s, fp	: Engine cylinder, plunger chamber, solenoid chamber, fuel pump
1, 2, m_f, $total$	: High-pressure line 1, 2, injected fuel, total volume of injection system

1. Introduction

The emissions from automotive engines were not an attractive issue for study by engineers until the 1960s. When people came to realize that the main source of the harmful emissions ; HC, CO, NO_x, SO_x and particulates in metropolitan areas, is the power generation unit for the city transportation system such as taxies, buses, sedans and trucks, it was decided to embark on serious considerations of these environmental problems. Since an emission standard for automotive engines was first introduced in the United States in early 1960s, much research work related to the performance improvement of engines has been conducted to improve the atmospheric conditions in urban areas. Awareness of emission control has since spread throughout the world.

The compression ignition (CI) type of engine is preferable to the spark ignition (SI) type for medium and large power outputs in many industrial fields. However, the CI engine is a major source of HC, NO_x and particulates. One of the technologies used to reduce the harmful emissions from CI engines is a higher-pressure injection system. For this, an electronically controlled high-pressure injection system is required. There are two kinds of injector systems, which are commonly used in diesel engines ; the common-rail injector system and the unit-injector system (Pulkrabek, 1997). Even though the former system shows stable and steady characteristics in the injection pressure at the initial stage of injection, the latter can build up a much higher injection pressure, reaching up to 2000 bar at the current stage.

This study is a part of a research project for the development of a new UI system that will be applied to a medium size diesel engine (Model No. 3176B, 366hp @1600 rpm) manufactured by the Caterpillar Co. Ltd. The final goal of this study is the development of the analytical design algorithm of a UI system. For this purpose, an analytical algorithm for the performance prediction of newly designed UI system will be developed as the first step.

2. Theoretical Background

In the case of small and medium-size high speed diesel engines, the combustion process occurs under constant-volume and pressure conditions. When liquid fuel is injected into a cylinder through an injection nozzle, it is broken up into small parcels of droplets and diffused into the combustion chamber. When a fuel droplet travels through the compressed hot air layer, evaporation occurs on its surface and a flammable gas layer is formed in the chamber (Kim, 1995). The flame starts in the space where the ambient condition is well developed for self-ignition. In the case of a CI engine, the ignition starts as a rapid pre-mixed combustion locally around the droplets and late turns into a turbulent diffusion combustion.

The combustion efficiency of a CI engine is greatly affected by the condition of the mixture in the cylinder. The liquid fuel should be broken up into small droplets that penetrates widely and deeply into the compressed air field to form a homogeneous well-mixed gas within a limited time. Because of this, an electronically controlled high-pressure UI system is beneficial to the diesel engine.

In this section, the theoretical background of the UI system is introduced which is based on the conservation of mass with an assumption of adiabatic process.

2.1 Structure of the UI system and its operational procedures

Figure 1 shows the schematics of the UI injector developed by DPICO (Lee et al, 1997). It

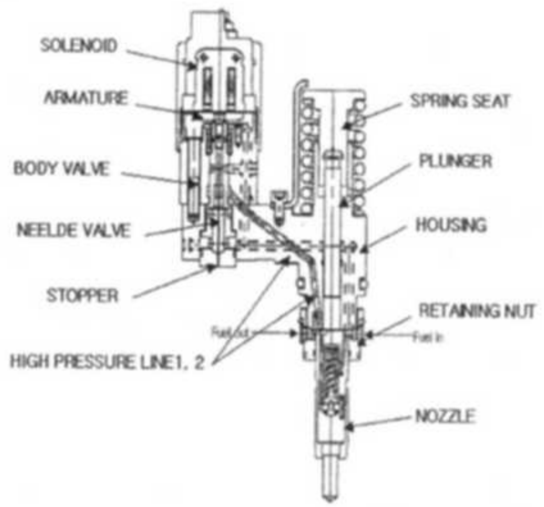

Fig. 1 Schematics of a UI system and its components

consists of three main components; an electric controlled solenoid valve to turn on and off the fuel supply line to the injection nozzle, a mechanically operated plunger pump to build up the higher injection pressure in the system, and an injection nozzle to introduce the atomized fuel into the combustion chamber. In this study, the diameter and number of the injection nozzle holes, the duration of injection, and the rotational speed of the plunger cam are the important parameters for the performance analysis of a newly designed UI injector.

2.2 Continuity equation of the UI system

The amount of the fuel volume injected into the engine cylinder per cycle should be equal to the extracted volume of the plunger chamber by the plunge piston in the UI system, if it is assumed that there is no fuel volume loss during the injection process in the system. However, it is hard to ignore the fuel volume loss in the system during the process. The most serious causes of the fuel volume loss considered in the system are;

- The compressibility effect of liquid fuel in the system
- Kinetic energy loss of fuel flow during injection process
- Leakage loss of liquid fuel in the system

Therefore, the governing equation of the fuel injection of the UI system can be obtained from the continuity equation of fluid flow considering the above loss factors in the system. The heat transfer is ignored in the system because the injection process can be assumed an adiabatic process.

From the law of mass conservation, the continuity equation for the unsteady state is; (Irving, 1996)

$$\iiint_{c.s.} \rho v_n dA + \frac{\partial}{\partial t}\iint_{c.v.} \rho d\forall = 0 \tag{1}$$

The first term of Eq. (1) is the difference between the fuel mass supplied into the UI system from the fuel pump per cycle and the fuel mass ejected from the system through the fuel injection into an engine cylinder and fuel leakage from the system. The transient term of Eq. (1) is the variation of the fuel mass in the UI system due to the compressibility effect of the liquid fuel, and the pressure drop in the high-pressure fuel line during the injection process. It can be rewritten as follows;

$$(\dot{m}_{f_{in}} - \dot{m}_{f_{out}}) + \frac{\partial}{\partial t}\iint_{c.v.} \rho d\forall = 0 \tag{2}$$

where

$\dot{m}_{f_{in}}$: Mass flow rate of fuel in accordance with the volume obtained in the plunger chamber

$\dot{m}_{f_{out}}$: Mass flow rate of fuel, including that injected into the engine cylinder and that leaked from the control volume of the UI system

3. Governing Equations and Analytical Algorithms

3.1 Governing equations

In the continuity equation for the system, Eq. (1), the transient term should be exactly defined and formulated to estimate the exact amount of fuel injected into a cylinder at any given operational condition. The term can be divided into three major physical terms due to the physical phenomenon in the system as given below.

(1) Compressibility effect of liquid fuel
(2) Leakage of the fuel from the system
(3) Surface friction loss of fuel flow

In the case of the surface friction loss of the fuel running through high-pressure injection line during the injection process, it has an indirect effect on the reduction of the injected fuel mass because some of the pressure energy generated in the plunger chamber disappears and the injection pressure is reduced.

These are defined as the transient loss terms of fuel mass in the system and can be written respectively as follows;

$$\dot{m}_{f_{bulk}},\ \dot{m}_{f_{fric}},\ \dot{m}_{f_{leak}}$$

Therefore, the continuity equation of the system can be rewritten,

$$\dot{m}_{f_{dis}} - \dot{m}_{f_{inj}} = \dot{m}_{f_{bulk}} + \dot{m}_{f_{leak}} + \dot{m}_{f_{fric}} \qquad (3)$$

3.1.1 The Compressibility effect of liquid fuel

In general, the normal operation pressure range in UI system is 200~2000 bar. In this case, the compressibility effect of the liquid fuel in the system cannot be ignored and even becomes an important fact that affects the mass change in the system. The level of injection pressure developed at the injector is determined by the volume change in the plunger chamber. The pressure in the plunger chamber can be calculated from Eq. (4) (Kim, 2001),

$$\beta \cdot (V_{total} + V_{m_f}) \cdot \frac{dP_p}{dt} = \frac{(A_p \cdot h_p - V_{m_f})}{dt} \qquad (4)$$

where $\beta = 1/K_f$ (compressibility factor)

- **Bulk modulus of diesel fuel** (K_f) (White, 1994)

The bulk modulus of diesel fuel is very sensitive to the fuel pressure and temperature. In this study, a linear polynomial equation was derived for the compressibility factor (β) of diesel fuel to include the effect of its pressure. It is based upon the experimental results (Rose, et al, 1977). A numerical curve-fitting program, Table Curve 2D (v5.0) (Table Curve 2D, 2001) was incorporated. The derived equation is given below,

$$\beta = a + b \cdot P_p + c \cdot P_p^{1.5} + d_p^{0.5} \qquad (5)$$

where, a=7.8377948 b=−1.529135e-4
c=4.2339265e-5 d=−8.7225419e-2

3.1.2 Surface friction loss of fuel flow

The pressure in the dead volume of the UI system increases continuously with the change in lift of the plunger piston until the needle valve of the injector is opened. It may be thought that the plunger chamber pressure (P_p) is the same as the injector pressure (P_{inj}) at each time step of the compression process of the plunger piston. However, a pressure difference occurs when injection starts because of the surface friction loss of the fuel flow in the control volume. At the end of the injection process, the solenoid valve is opened in order to return the residual fuel in the plunger chamber to the fuel storage. Because of the above two physical phenomena, the injection pressure is lower than the pressure in the plunger chamber during the injection process.

$$P_{plunger} = P_{injector} + \Delta P_{fric_{1,2}} \qquad (6)$$

Here, the frictional loss of pressure in the high-pressure fuel line 1, 2 in Fig. 1 is;

$$\Delta P_{fric_{1,2}} = \rho_f \cdot f \cdot \left(\frac{dL_{1,2}}{d_{1,2}}\right) \cdot \left(\frac{V_{1,2}^2}{2}\right) \qquad (7)$$

The friction factor in a smooth, circular pipe (White, 1994) is;

$$f = \left(1.8 \log \frac{R_n}{6.9}\right)^{-2} \qquad (8)$$

3.1.3 Leakage loss of liquid fuel in the system

The pressure of the liquid fuel in the dead volume is around 200~2000bar during the compression process. Therefore, some of the fuel leaks from the control volume and in turn, could affect the drop of the injection pressure in the system. The total volume of the leaked fuel per cycle is not easy to measure; however, an empirical equation provided by DEPCO (Lee et al, 1997) is used in the continuity equation for the leakage effects of this study.

$$Rate_{leakage}=1.0-\frac{0.02}{t_{tot}}\Delta t_{st} \quad (9)$$

where t_{tot} : Total injection time at a given rpm and duration of injection
Δt_{st}: Time step for calculation

3.1.4 Injected fuel mass

The total amount of fuel injected into a cylinder per cycle by an injection nozzle varies with the pressure difference between the injector and the engine cylinder. The injected fuel mass per cycle is calculated by (Heywood, 1988),

$$\dot{m}_{f_{inj}}=C_N \cdot A_N \cdot n \cdot \sqrt{2 \cdot \rho_f \cdot (P_N-P_{cyl})} \quad (10)$$

Here, the discharge coefficient (C_N) of the nozzle is calculated by Eq. (11) with an assumption that the nozzle is a circular orifice in shape (ASME, 1981).

$$C_N=\alpha[f(\delta_N)+97.91 \cdot \delta_N^{2.5} \cdot R_n^{-0.68}] \quad (11)$$

where

$$f(\delta_N)=0.5959+0.03312 \cdot \delta^{2.1}-0.18 \cdot \delta^8,\ \alpha=f(\zeta)$$

3.1.5 Returned fuel mass

The pressure of the plunger chamber is also affected by the returning process of the fuel, which is controlled by a solenoid valve. The amount of fuel returned per cycle is decided by the pressure difference between the fuel pump line and the high-pressure line of the system.

$$\dot{m}_{sp}=C_s \cdot A_{sp} \cdot \sqrt{2 \cdot \rho_f \cdot (P_p-P_{fp})} \quad (12)$$

3.2 Operational conditions for calculation

The calculation conditions for the analytical model of the UI system are given below. These are the same conditions given to the experimentation.

- Solenoid valve (S.V.) is completely shut off at 132° of the plunger cam angle.
- Duration of injection (t_{inj}) is fixed on three different cam angles ; 2.5, 10.0 and 17.5 degree
- 0.3 ms time delay exists between the electrical signal access and the activation of the solenoid valve.

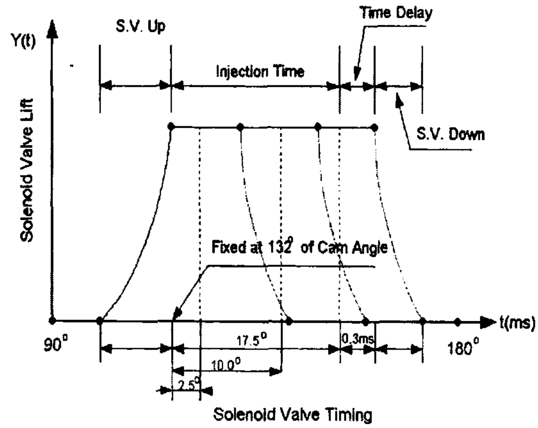

Fig. 2 Operational time schedule of the fuel return valve controlled by an electric solenoid valve

Figure 2 shows the details of the operation schedule of S.V. per cycle (Lee et al, 1997).

The total operational time of the S.V. per cycle depends on the rotational speed of the plunger cam. The time step (Δt_{st}) is the basic component used to calculate the plunger pressure and the fuel amount injected into the cylinder at each step. The operational procedure per cycle consists of four different steps ; S.V. Opening, Injection, Shut-off Delay, S.V. Shut-off. The time at each cam angle is obtained by Eq. (13).

$$t=\frac{(cam\_angle) \cdot 1000}{6 \cdot N}(ms) \quad (13)$$

4. Parameters for Analytical Calculation and Performance Analysis Methods

4.1 Parameters for analytical calculation

In the previous study (Kim, 2001), the rotational speed of the plunger cam was the only parameter incorporated to evaluate the reliability of the simulation program for the performance analysis of the UI system. In the present study, the duration of injection (t_{inj}) and the effective diameter (D) of the injection nozzle are included for a comprehensive analysis of the performance of the simulation program. The parameters and their ranges are given in Table 3.

Table 3 Important parameters and their ranges for the analytical study

	Operation Range			
Rotational speed of plunger cam	400～1300 rpm			
Duration of injection (t_{inj})	2.5, 10.0, 17.5 degree (cam angle)			
Injector Model	Model-1 D_{01}	Model-2 D_{02}	Model-3 D_{03}	Model-4 D_{04}
(Hole diameter × number)	0.20×6	0.22×6	0.20×8	0.22×8
Effective Diameter	0.489	0.539	0.566	0.622
Effective Diameter : $D_{eff}=\sqrt{\frac{4}{\pi}(n\times A_N)}$				

4.2 Analysis methods of calculation results

The analytical calculation result is compared with the experimental output (Lee et al, 1997) to examine the reliability of the analytical algorithm developed in this study. Due to the limitation of the experimental condition, only two performance characteristics of the UI system, that is the maximum plunger pressure (bar) and the total volume (mm^3) of the fuel injected into atmospheric condition per cycle, were measured at each operational condition for comparison with the results to the calculation outputs.

For the evaluation of the calculation result, the mean value of the error rate (%) is incorporated as given below,

- Error rate (%) of maximum plunge pressure ;

$$\Delta P_{err}=\frac{|(\Delta P_{\exp}-\Delta P_{th})|}{\Delta P_{\exp}}\times 100(\%) \tag{14}$$

- Error rate (%) of total mass flow rate of injected fuel ;

$$\dot{m}_{f_{err}}=\frac{|(\dot{m}_{f_{\exp}}-\dot{m}_{f_{th}})|}{\dot{m}_{f_{\exp}}}\times 100(\%) \tag{15}$$

5. Results and Discussion

For details of the performance of a newly designed UI system, the transient outputs of the important performance parameters of the UI system should be obtained from the analytical calculation. The following parameters are the calculated outputs of the analytical simulation algorithm developed in this study. However, only the first two factors have been compared with the experimental results.

- Injected Fuel Volume ($\dot{Q}_f$)
- Plunger Chamber Pressure (P_p)
- Injection Nozzle Pressure (P_N)
- Lift of Plunger Piston (L_{pp})
- Velocity of Plunger Piston (V_{pp})
- Lift of Solenoid Valve (L_s)

5.1 Analytical simulation results of the model UI system

The principal parameters for the general performance analysis of the model injector are the maximum injection pressure ($\boldsymbol{P_{inj}}$) and the injected fuel volume ($\boldsymbol{Q_f}$) per cycle at each operational condition. However, only the maximum plunger pressure ($\boldsymbol{P_p}$) was measured in the experiment due to the difficulties in measuring the injection pressure ($\boldsymbol{P_{inj}}$). The plunger pressure was compared with the simulation result.

Figure 3 shows the effect of the effective diameter of the injection nozzle on the maximum plunger pressure and the injected fuel volume at 10 degree of the duration of injection.

The plunger pressure increases linearly with respect to the rotational speed of the plunger cam. When the effective diameter of the injection nozzle decreases, the plunger chamber pressure increases over all the cam speed ranges.

The injected fuel volume increases inversely as the cam speed increases. The smaller the size of the injection nozzle, the less the amount of the fuel injected into cylinder. This is mainly due to the kinetic energy loss of the fuel flow in the injection system, that is, the surface friction loss in the fuel line and the surging effect of the fuel flow in the orifice-type injection nozzle. As shown in Fig. 4, the injected fuel volume increases with respect to the cam rotational speed at 2.5 degree of the duration of injection. This means that the increased rate of the plunger pressure in

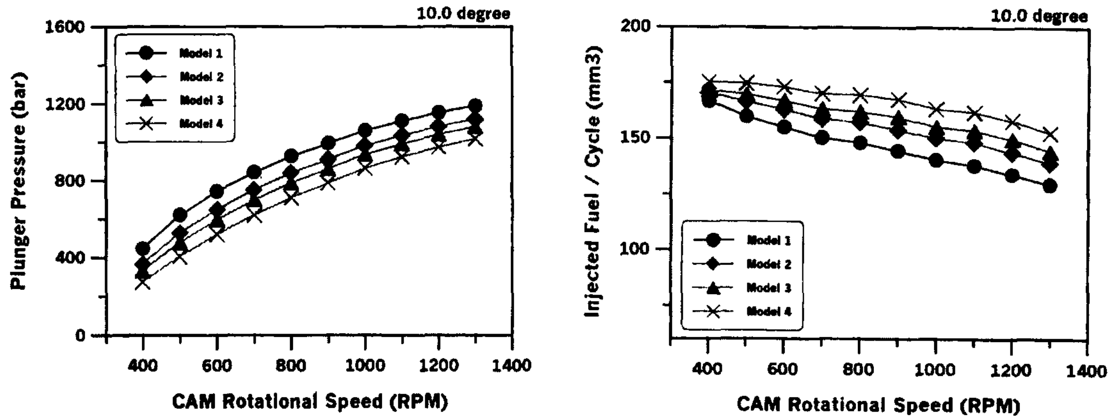

Fig. 3 Effect of the effective diameters of the model injectors on the injection characteristics at 10.0 degree of the duration of injection ; (Effective diameter in Model 1~Model 4: 0.489, 0.539, 0.566, 0.622 respectively)

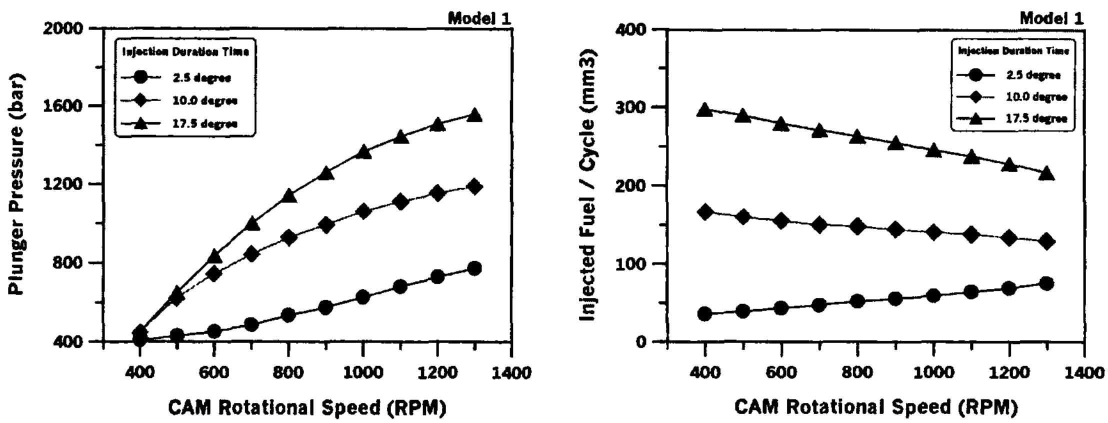

Fig. 4 Effect of the duration of injection (t_{inj}) on the injection characteristics at Model 1

the system is higher than that of the kinetic energy loss of fuel flow.

The duration of injection (t_{inj}) is another important design parameter of the UI system. Figure 4 shows the effect of the duration (t_{inj}) on the performance of Model 1 injector. When the duration of the injection is longer, a higher plunger pressure (P_p) is achieved in the system. This is due to the longer duration of injection in the injection process, the bigger volume of the plunger chamber which is contracted at a given cam rotational speed.

Because of this, the plunger pressure increases as the duration of injection increases. The maximum plunger pressure reached at 17.5 degree of the duration of injection is 1559.4bar at 1300 rpm.

The injected fuel volume increases as the duration of injection increases over all the cam speed ranges. The injected volume decreases with respect to the cam speed at the first two durations of injection ; 10.0, 17.5 degree ; however, the injected volume increases at 2.5 degree with respect to the cam speed. This means that the kinetic energy loss, and the surging effect of the fuel flow due to the extreme plunger pressure in the fuel line, is less for 2.5 degree than for 10.0 degree and 17.5 degree.

With the above analytical simulation results, the configuration of a newly designed UI injector ; hole diameter, number of holes and the length of high pressure line, and its operation conditions ; cam speed and the duration of injection, can all be decided in order to provide an optimised UI system for a newly designed CI engine.

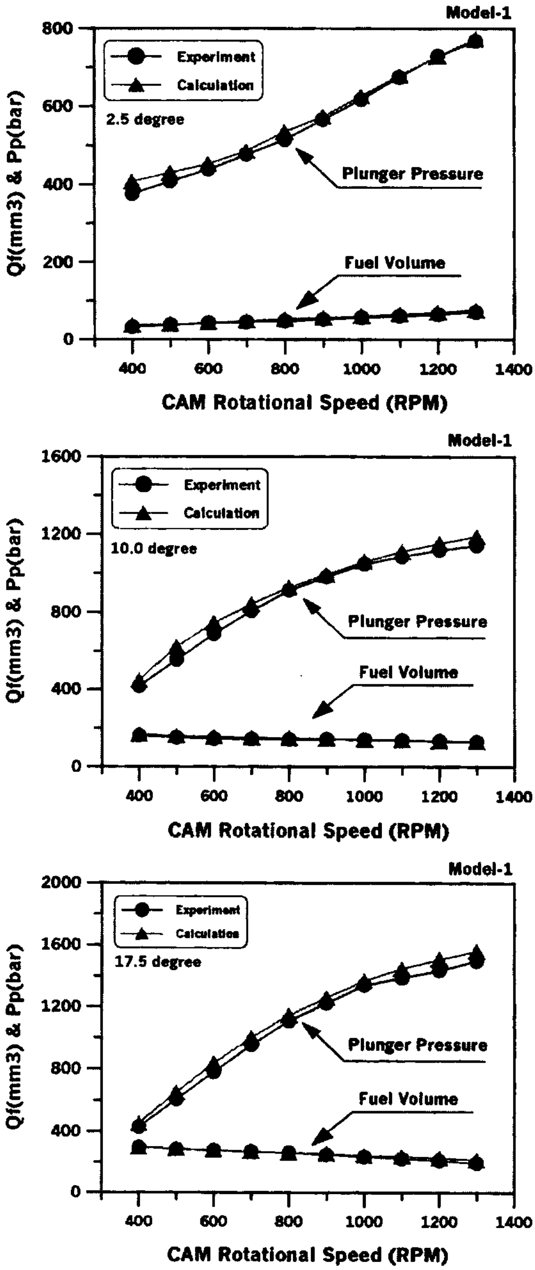

Fig. 5 Comparison of the analytical result to the experimental output in the maximum plunger pressure (P_p) and injected fuel volume (Q_f)

5.2 Comparison of the simulation results with the experimental outputs

In this section, the analytical simulation results of the newly designed UI system are compared with its experimental outputs. This provides a feasibility test of the analytical algorithm developed in this study.

Figure 5 shows the difference between the simulation results and the experimental output of the principal performance parameters ; the maximum plunger pressure (P_p) and the injected fuel volume (Q_f) of the newly designed UI system at different duration injection steps (t_{inj}).

In the case of the maximum plunger pressure (P_p), the simulation result is slightly higher over the cam speed range at each step of the duration of injection (t_{inj}); however, reasonable simulation results of the new designed UI system are obtained in all operational conditions. The same trends are shown for the three different models.

5.3 Analysis of error rate between the analytical simulation results and experimental outputs

The averaged error rate (%) of the maximum plunge pressure and injected fuel volume between the analytical simulation results and the experimental outputs are calculated by Eq. (16) and Eq. (17). The mean value was determined again among the error rates (%) for the same model of injector.

$$\overline{\Delta P_{err}}=\frac{1}{n}\sum_{i=1}^{n}\Delta P_{err_i} \quad (16)$$

$$\overline{\dot{m}_{f_{err}}}=\frac{1}{n}\sum_{i=1}^{n}\dot{m}_{f_{err_i}} \quad (17)$$

where i is the number of the step of the duration of injection ; 2.5, 10.0 and 17.5 degree.

As shown in Fig. 6, the error rate (%) of each model is less than 5% except in Model 4. The mean value of the error rate (%) of the maximum plunger pressure (P_p) is 2.87% and 4.94% for the injected fuel volume (Q_f).

The error rate (%) of the calculated results of the UI system is analyzed in a different way. The error rate of each case is averaged according to

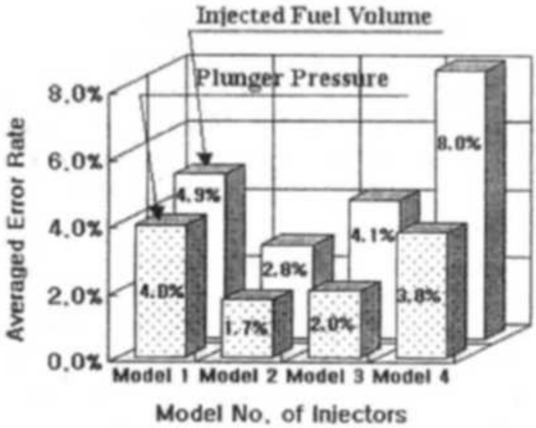

Fig. 6 Variation of error rate (%) of the maximum plunge pressure (P_p) and injected fuel volume (Q_f) at each model

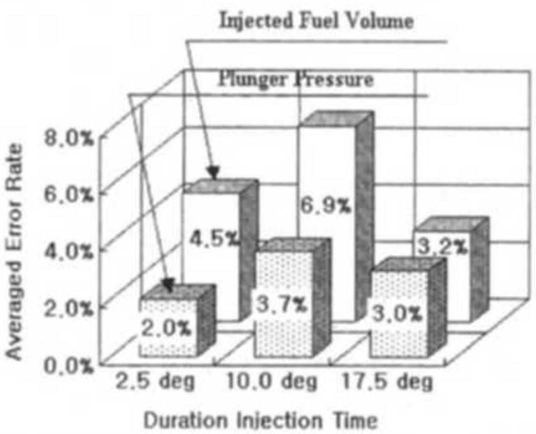

Fig. 7 Variation of error rate (%) of the maximum plunge pressure (P_p) and injected fuel volume (Q_f) at each injection duration

the duration of injection. Thus, the mean value of the error rate (%) of the maximum plunger pressure (P_p) and the injected fuel volume (Q_f) are 2.90% and 4.87%, respectively.

6. Conclusions

An analytical simulation algorithm was developed to predict the performance of a newly designed Unit-injector System of a diesel engine. This study is a primary part of the research work to develop a design software of Unit-Injector system for the application to small and medium size diesel engines.

The fundamental theory of the algorithm is based upon the continuity equation of fluid dynamics. It is inevitable that there would be several reasons for the loss of mass and pressure in the system during operation. Therefore, the physical phenomena that cause the loss of mass and pressure in the system should be carefully studied and analyzed quantitatively to take into account the proper sink terms in the continuity equation.

To be able to find the proper amount of the loss of fuel mass and pressure during injection in the UI system, three representative physical phenomena that occur repeatedly in the system during operation have been analyzed. These are,

- Compressibility effect of the liquid fuel in the dead volume of the system
- Wall friction and kinetic energy loss of fuel whilst it runs through the high-pressure lines in the system
- Leakage of the fuel from the system

6.1 Reliability of the analytical algorithm

From the error rate analysis of the analytical simulation results in section 5.3, it was found that the average error rate of the maximum plunger pressure (P_p) and the injected fuel volume (Q_f) is less than 3.0% and 5.0% respectively. Although the error rate of the simulation results in some cases are slightly over the average values, the analytical algorithm developed in this study is quite reasonable for the prediction of the performance of a newly designed UI system.

Some assumptions are given for the estimation of the mass and pressure loss of the fuel in the UI system. For example, the wall friction loss in the high-pressure fuel line was calculated with the assumption that the inner surface of the pipeline is smooth ; the spray hole of the injector nozzle was assumed as a circular orifice nozzle to estimate the discharge coefficient (C_N); the compressibility factor (β) of the fuel is mathematically fitted to estimate the value in the higher-pressure range with the given experimental data.

These assumptions are the major sources that can be the cause of the error in the simulation results of this analytical simulation algorithm. Therefore, a further study is required on the previous assumptions to reduce the errors in the prediction of the performance of a newly designed UI system.

6.2 Injection characteristics of a UI system

In this analytical study, the performance of the newly designed UI system was predicted with changes in the operational conditions ; the rotational speed of the plunger cam, the injection duration and the effective diameter of the injection nozzle.

(1) Effect of the rotational speed of the plunger cam

The plunger pressure (P_p) increases linearly with respect to the cam speed ; however, the increase rate of the pressure was reduced as the cam speed was increased. It is believed that the main reason for this phenomenon is due to the change in the bulk modulus of the fuel, which is sensitive to its pressure and temperature.

The injected fuel volume (Q_f) was decreased linearly with respect to the cam speed at 10.0 and 17.5 degree duration of injection ; however, it showed the reverse trend at 2.5 degrees of the injection time. It is assumed that the plunger pressure in the UI system was not developed sufficiently to decrease the discharge coefficient of the injector nozzle at 2.5 degrees of injection time and the surface friction loss of the fuel flow is not higher as much as at 10.0 and 17.5 degree.

(2) Effect of the duration of injection (t_{inj})

A higher plunger pressure (P_p) can be attained, as the injected time (t_{inj}) is lengthened over all the ranges of cam rotational speed. This is due to the size of the compressed fuel volume in the plunger chamber. If the duration of injection is longer at a given cam speed, more fuel volume is compressed until the solenoid valve is opened.

In the case of the injected fuel volume, the longer the injection duration sustained, the bigger the volume of the fuel injected into the engine cylinder.

(3) Effective diameter of the injector

The size of the injector nozzle hole is directly related to the value of the discharge coefficient of the hole.

At a fixed rotational speed of the plunger cam, the plunger pressure can reach high and the injected fuel volume is decreased as the hole size is increased.

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