Measurement and Simulation of Fuel Injection Pipe Pressure and Study of Its Effect on the Heat Release in a Direct Injection Diesel Engine

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Fuel injection pipe pressures are measured and simulated to study the effect of fuel injection system characteristics on the heat release in a direct injection diesel engine. The fuel injection simulation is based on a linear model. The governing equations are solved by the finite difference method. The measured fuel pipe pressures and the simulated fuel pipe pressures matched well to each other except for the interval when the nozzle is closing. The effects of the fuel pipe length and the nozzle opening pressure are tested. The longer fuel pipe length causes proportional retardation of the fuel injection time. The higher nozzle opening pressure results in increase of the maximum fuel pipe pressure and shorter combustion duration.

Key Words: Diesel Engine, Fuel Injection, Finite Difference Method, Pressure Simulation, Heat Release Analysis

Nomenclature

A_p : Plunger cross section area
A_{pipe} : Area of fuel injection pipe
A_{hole} : Nozzle hole area
A_{no} : Flow area from nozzle chamber to sac volume
A_v : Delivery valve area
A_{sc} : Sac volume cross section area
A_{nseff} : Effective flow area from nozzle chamber to cylinder
C_d : Discharge coefficient
D : Fuel injection pipe inner diameter
f_d : Delivery valve initial spring force
f_n : Needle valve initial spring force
k_d : Delivery valve spring constant
k_n : Nozzle spring constant
M_d : Delivery valve moving mass
M_n : Nozzle valve moving mass
P : Pressure
P_p : Plunger chamber pressure
P_d : Delivery chamber pressure
P_n : Nozzle chamber pressure
P_s : Sac volume pressure
P_c : Cylinder pressure
Q_{pd} : Flow between plunger chamber and delivery chamber
Q_{ps} : Flow between plunger chamber and spill port
V_d : Delivery chamber volume
V_p : Plunger chamber volume
W : Cumulative flow
W_1 : Cumulative flow of first pipe node
W_L : Cumulative flow of last pipe node
X_n : Needle lift
X_d : Delivery valve movement
β_o : Bulk modulus at vapour pressure
ρ_o : Fuel density
ν : Kinematic viscosity
ΔX : Length step
Δt : Time step

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1. Introduction

Diesel engine is one of the power generation devices which convert chemical energy to mechanical energy. Due to its high thermal efficiency, diesel engines have been applied more and more to the vehicles and power plant. Due to the pollution of the environment and the shortage of energy sources, many diesel engine researches are carried on to improve the performance and to reduce the toxic emissions such as NOx and Particulates. (Pierpont & Retiz, 1995 ; Sugiyama et al., 1994) Many of these researches are focused on the optimization of the fuel injection system. This is because the diesel combustion processes are characterized by non-homogeneous diffusion flame and auto-ignition. The fuel injection system affects the diesel engine combustion process through the control of injection time, injection quantity and injection rate. (Heywood, 1988)

Many simulations of the diesel engine fuel injection system are carried out to understand the fuel injection system characteristics. (Lee, 1994 ; Arcoumanis & Fairbrother, 1992 ; Kumar et al, 1983) The method of characteristics and the finite difference method have been used to solve the governing equations. Recent studies show that finite difference method is more advantageous than the method of characteristics concerning computation time and nonlinearity. (Lee, 1994 ; Kumar et al, 1983) A simple linear model solved by the finite difference method is developed in this study which is programmed in C language. An one-zone heat release calculation is carried out using the measured cylinder pressure to analyze the combustion process.

2. Simulation

2.1 Fuel injection system modelling

The task of the fuel injection system is to meter the appropriate quantity of fuel for given engine speed and load, and to inject that fuel at an appropriate time. A fuel injection system is divided into pump, pipe and nozzle component to model the entire system. Following assumptions are made to develop a model. (Lee, 1994 ; Arcoumanis & Fairbrother, 1992 ; Kumar et al, 1983)

1) The variation of fuel bulk modulus and density is negligible.

2) The flow in the fuel injection pipe is one dimensional laminar flow. The convective term in the momentum equation can be neglected. The steady flow friction coefficient is used for the transient flow friction coefficient.

3) The discharge coefficient is constant everywhere in the fuel injection system.

As a result of above assumptions, a linear equation of state for the pressure can be formulated as Eqs. (1) and (2).

$$P=\beta_o S \tag{1}$$

$$S\equiv\frac{\rho-\rho_o}{\rho_o} \tag{2}$$

2.2 Fuel injection pump

The volume of the pump component can be divided into 3 control volumes such as plunger chamber, delivery chamber, and spill port as shown in Fig. 1. The continuity of flow in 3 control volumes and the equilibrium of forces on the delivery valve give three governing equations.

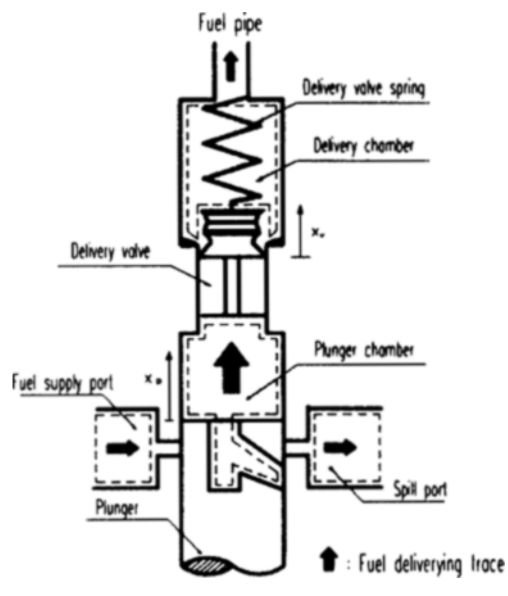

Fig. 1 Schematic diagram of a fuel pump modeling

An orifice model is used as a flow model for the interfacing control volume.

$$A_p\frac{dX_p}{dt}=\frac{V_p}{\beta_o}\frac{dP_p}{dt}+A_v\frac{dX_d}{dt}+A_v\frac{dX_d}{dt} + Q_{pd}+Q_{ps} \tag{3}$$

$$A_v\frac{dX_d}{dt}+Q_{pd}=\frac{V_d}{\beta_o}\frac{dP_d}{dt}+\frac{dW_1}{dt} \tag{4}$$

$$M_d\frac{d^2X_d}{dt^2}+k_dX_d=A_v(P_p-P_d)-f_d \tag{5}$$

2.3 Fuel injection pipe

The continuity equation, momentum equation and the equation of state (Eq. (1)) are taken into account to calculate the flow and the pressure in the fuel injection pipe. The momentum equation can be expressed by Eq. (6) in the form of a cumulative flow equation.

$$\frac{\partial^2 W}{\partial t^2}=-\frac{A_{pipe}}{\rho_o}\frac{\partial P}{\partial x}-\frac{32\nu}{D^2}\frac{\partial W}{\partial t} \tag{6}$$

2.4 Fuel injection nozzle

The fuel injection nozzle volume can be separated into two control volumes, i. e. nozzle chamber volume and sac volume as in Fig. 2. It is assumed in the process of constructing governing equations that the injection rate into the combustion chamber is the same as the fuel flow rate from the nozzle chamber to the sac volume. By considering of the continuity equation and the equilibrium of force on the needle valve, Eqs. (7) and (8) can be obtained.

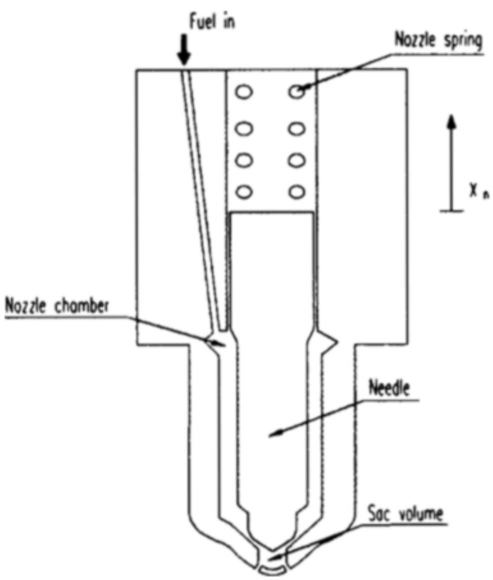

Fig. 2 Schematic diagram of a nozzle

$$\frac{dW_L}{dt}=\frac{V_n}{\beta_o}\frac{dp_n}{dt}+A_n\frac{dX_n}{dt} + C_dA_{nseff}\sqrt{\frac{2(P_n-P_c)}{\rho}} \tag{7}$$

$$M_n\frac{d^2X_n}{dt^2}+k_nX_n=P_n(A_n-A_{sc}) + P_sA_{sc}-f_n \tag{8}$$

where $A_{nseff}=\dfrac{A_{hole}A_{no}}{\sqrt{A_{hole}^2+A_{no}^2}}$

2.5 Boundary conditions

Fuel is delivered into the cylinder through the injection pump, the injection pipe, and the injection nozzle. Fuel is compressed by the plunger lift in the pump and injected into the cylinder. For that reason, the plunger lift and the cylinder pressure are used as the boundary condition. The plunger lift is calculated from the cam profile and roller's diameter in the in-line pump. The cylinder pressure is measured by the pressure transducer mounted in the cylinder head. As mentioned above, the entire fuel injection system is modeled as a three-component model i. e. pump, pipe, and nozzle. In order to calculate the whole system, it is assumed that the delivery chamber pressure is identical to the first node pressure of the injection pipe. Similar assumption is also applied to another boundary section where the last node of the fuel pipe meets the nozzle chamber.

2.6 Calculation method

When Eqs. (3), (4), and (7) are discretized by the finite difference method, they are nonlinear equations. To find solutions for these nonlinear simultaneous equations, Newton-Raphson method is employed. The fuel injection pipe governing equations are turned into a finite difference form by the Leap-Frog scheme. This explicit scheme converges when Eq. (9) is satisfied. (Anderson et al, 1984)

$$\frac{\Delta X}{\Delta t}\geq C_o=\sqrt{\frac{\beta_o}{\rho_o}} \tag{9}$$

3. Experiment and Analysis

3.1 Experimental setup

The experiment is carried out to measure the cylinder pressure and the fuel injection pipe pressure. The experimental set-up includes a 4 stroke cycle 6 cylinder diesel engine, a dynamometer, data acquisition system, pressure transducers, and etc. Table 1 shows the specifications of the test engine. The specification of the fuel injection system is listed in Table 2.

3.2 Heat release analysis

Heat release is widely used to obtain informations in the progress of combustion. This method starts with the first law of thermodynamics for an open system which is quasi-static. In this study, an one-zone heat release which is simple but reliable, is introduced. An one-zone heat release has been developed to apply more appropriate specific heat ratio in gasoline engines (Chun & Heywood, 1987 ; Cheung & Heywood, 1993) and a diesel engine. (Cheng & Heywood, 1986) We tested this one-zone analysis and showed its accuracy. (Shin et al, 1996) The burning duration is calculated on the basis of the integrated heat release.

Table 1 Specification of the test engine

Type	Direct injection, 6 cylinder watercooling
Induction type	Natural aspiration
Displacement volume	7545 cc
Cylinder Bore × Stroke	118×115 mm
Compression ratio	17.5
Ignition order	1-5-3-6-2-4
Maximum Torque	495 N · m at 1400 rpm
Maximum Power	123 Kw at 2200 rpm

Table 2 Specification of fuel injection system

Pump type	Bosch in-line pump
Plunger diameter	10.5 mm
Prestroke	3.2 mm
Fuel pipe diameter	2 mm
Nozzle type	Hole type
Nozzle hole number	5
Nozzle hole diameter	0.31 mm
Nozzle hole angle	160°
Opening pressure	21.6 MPa

4. Results and Discussion

Figures 3 and 4 show the measured and simulated fuel pipe pressures for the engine speeds of 1000 rpm and 1500 rpm at full load. The maximum fuel pipe pressure of the simulation is similar to that of the measurement with slight

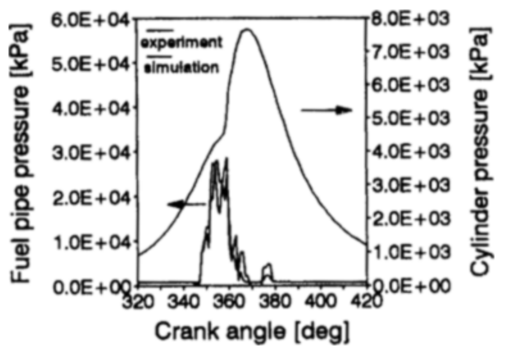

Fig. 3 Measured and simulated fuel pipe pressure and cylinder pressure at 1000 rpm, full load (opening pressure : 21.6 MPa, pipe length : 420 mm)

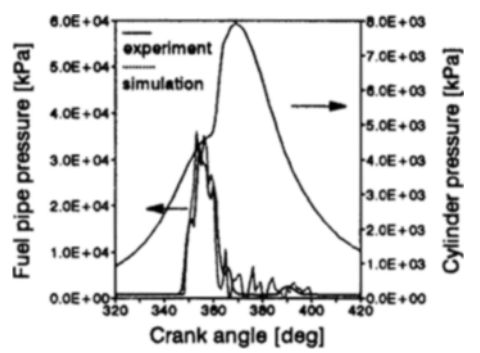

Fig. 4 Measured and simulated fuel pipe pressure and cylinder pressure at 1500 rpm, full load (opening pressure : 21.6 MPa, pipe length : 420 mm)

difference in the phase. The difference between the simulated and the measured pressures becomes clear as the delivery valve starts to close. This can be explained by the followings. The computation model is a linear model which is not adequate for the cavitation phenomena. The cavitation may occur in the later period of the injection. Just before the termination of the injection, the delivery valve closes rapidly to ensure the rapid closure of the needle. The high closing velocity generates a steep expansion wave which might result in cavitation. When cavitation occurs, pressure wave has high frequency and negative values. These characteristics make the measurement difficult (Lee, 1994). Figures 5 and 6 show the needle lift and injection rate which are calculated.

Variations of the fuel injection pipe pressure in different fuel injection pipe lengths are shown in Figs. 7 and 8. The engine is operated at constant speed with full load. The pipe length is the distance from the pump outlet to the nozzle holder inlet. 420 mm, 490 mm, 545 mm and 650 mm are chosen as the test fuel pipe length. The fuel pipe length of the commercial engine is 420 mm. As the fuel pipe length increases, fuel injection pressure arrives at the nozzle later.

The cylinder pressures are affected by the fuel pipe length as shown in Fig. 9. Heat release rate calculated by the one-zone analysis (Shin et al, 1996) in Fig. 10 shows that combustion starts later with a longer pipe length.

Variations of the fuel injection pipe pressure in different nozzle opening pressures are shown in

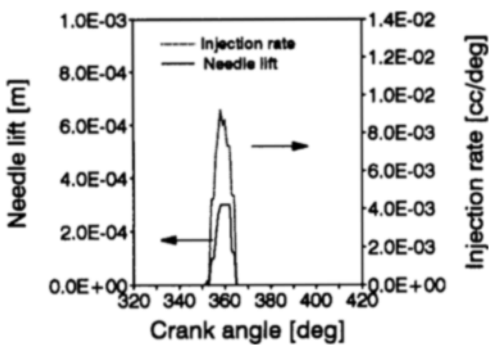

Fig. 5 Needle lift and injection rate at 1000 rpm, full load from fuel injection system simulation (opening pressure : 21.6 MPa, pipe length : 420 mm)

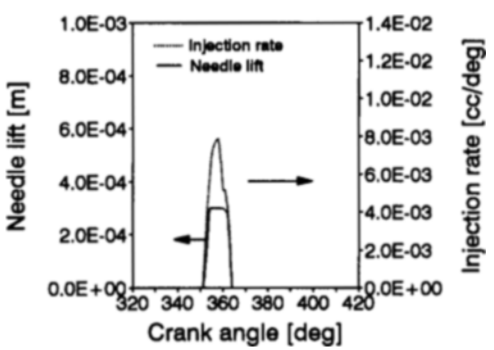

Fig. 6 Needle lift and injection rate at 1500 rpm, full load from fuel injection system simulation (opening pressure : 21.6 MPa, pipe length : 420 mm)

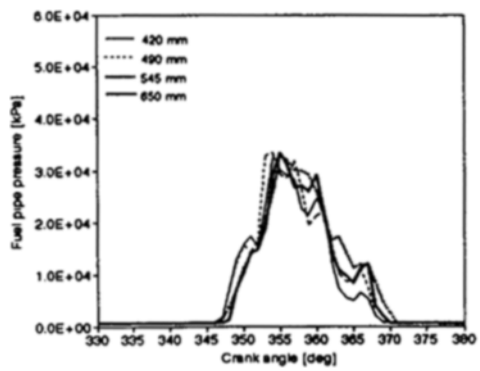

Fig. 7 Measured fuel pipe pressure for various fuel pipe lengths at 1500 rpm, full load (opening pressure : 21.6 MPa)

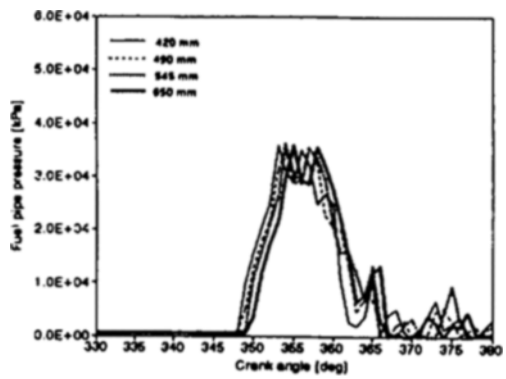

Fig. 8 Simulated fuel pipe pressure for various fuel pipe lengths at 1500 rpm, full load (opening pressure : 21.6 MPa)

Figs. 11 and 12. The nozzle opening pressures can be adjusted by changing the initial nozzle spring tension. Selected test opening pressures are 19.0 MPa, 21.6 MPa, 24.5 MPa and 28.4 MPa. The results of simulation and experiment are compared with engine speed and pipe length fixed. The

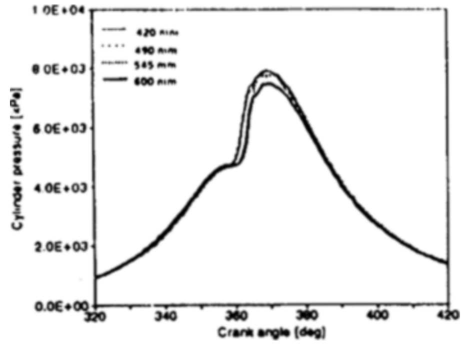

Fig. 9 Cylinder pressure for various fuel pipe lengths at 1500 rpm, full load (opening pressure : 21.6 MPa)

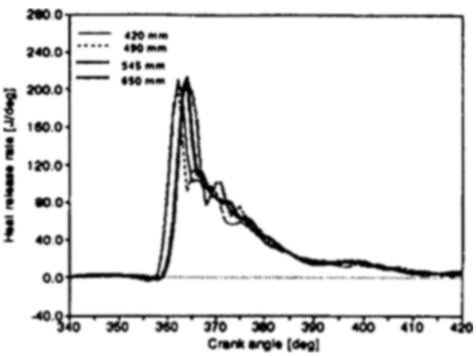

Fig. 10 Heat release rate for various fuel pipe lengths at 1500 rpm, full load (opening pressure : 21.6 MPa)

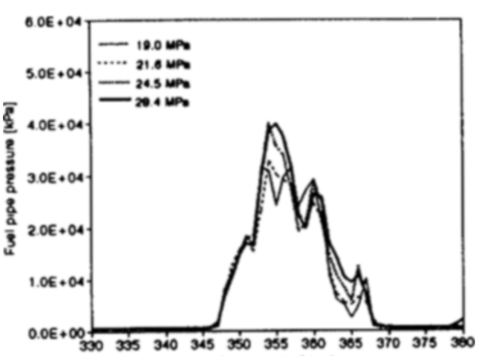

Fig. 11 Measured fuel pipe pressure for various opening pressures at 1500 rpm, full load (pipe length : 420 mm)

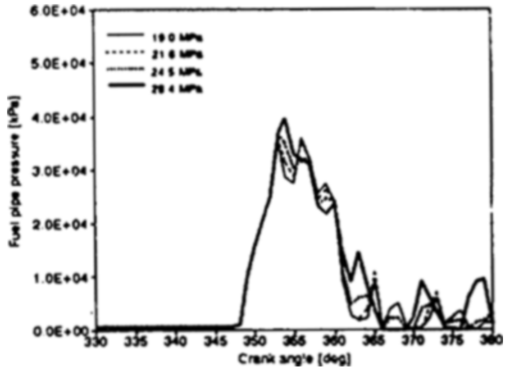

Fig. 12 Simulated fuel pipe pressure for various opening pressures at 1500 rpm, full load (pipe length : 420 mm)

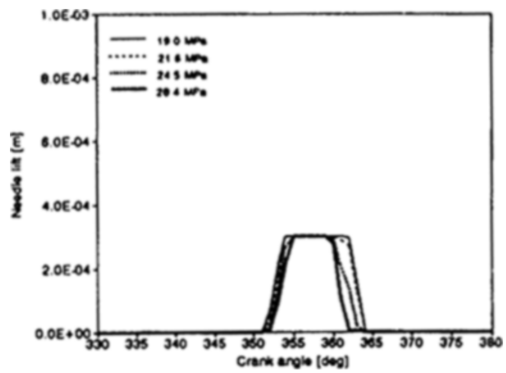

Fig. 13 Simulated needle lift for various opening pressures at 1500 rpm, full load (pipe length : 420 mm)

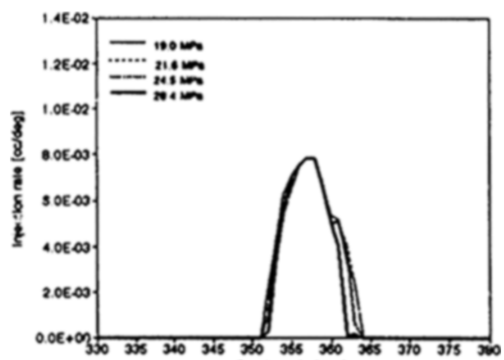

Fig. 14 Simulated injection rate for various opening pressures at 1500 rpm, full load (pipe length : 420 mm)

higher injection nozzle opening pressure results in maximum fuel pipe pressure increase and injected fuel quantity decrease.

As injection nozzle opening pressure increases, the injection starting time is retarded and the injection duration becomes shorter as shown in Figs. 13 and 14. Cylinder pressure variations are plotted in Fig. 15. Heat release results are illustrated by Figs. 16 and 17. The overall burning duration is reduced with higher nozzle opening pressure while the initial burning duration is almost the same. It may be caused by the fact that fuel evaporates faster and mixes with air better with the injection pressure increase.

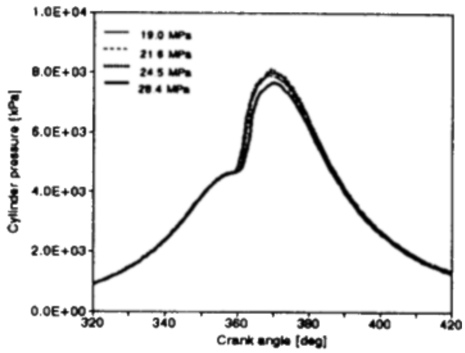

Fig. 15 Cylinder pressure for various opening pressures at 1500 rpm, full load (pipe length : 420 mm)

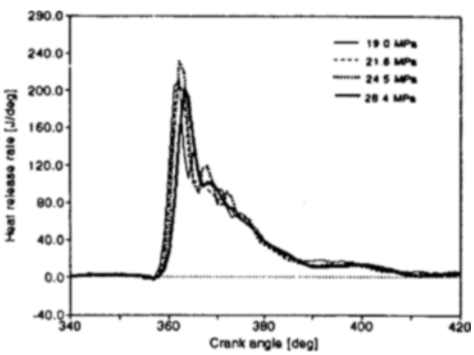

Fig. 16 Heat release rate for various opening pressures at 1500 rpm, full load (pipe length : 420 mm)

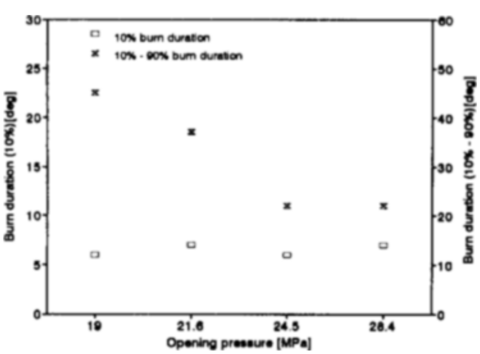

Fig. 17 Burn duration as a function of opening pressure at 1500 rpm, full load (pipe length : 420 mm)

5. Summary and Conclusion

A study is carried out to investigate the effects of the fuel injection system pressure variations on the combustion of a direct injection diesel engine. The fuel injection pressure and cylinder pressure are measured at the same time. Fuel injection system flow is simulated by the finite difference method. The cylinder pressures are analysed by an one-zone heat release analysis to find out the combustion characteristics. The following conclusions are derived from this research.

(1) The measured fuel pipe pressures and the simulated fuel pipe pressures matched well to each other except for the later stage of the injection when cavitation might be occurring.

(2) The longer fuel pipe length causes the retarded fuel injection timing. These variations affect the injected fuel quantity and the injection pipe pressure profile after the injection.

(3) The higher nozzle opening pressure results in maximum fuel pipe pressure increase, injection duration decrease and retardation of fuel injection time. The overall burning duration becomes shorter as the nozzle opening pressure becomes higher.

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