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Modelling and reducing fuel flow pulsation of a fuel-metering system during pump mode switching in a turbofan engine

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Abstract. In this research, the transient characteristics of a new type fuel metering system for aircraft turbofan engines using two types of pumps, a gear pump and a centrifugal pump, were studied by modelling and simulation. This new fuel metering system is capable of hydraulically unloading the operating state of each pump by the pump mode switching mechanism within the permissible range of the pump rotation speed and the fuel flow rate and it is possible to reduce the fuel temperature rise due to the pressure loss in the circuit in the operation region of the entire fuel system. In addition to this, by using a small centrifugal pump that can realize high discharge flow rate, it is possible to reduce weight compared to the conventional fuel system. However, when the pump mode is switched, unexpected fluctuation in the fuel flow rate can occur when a large change in the inlet pressure of the system occurs, the pressure control unit cannot follow the change, and the transient phenomenon cannot be suppressed. At this time, the pressure waveform generated at the time of changing the pump mode includes a high frequency component such that the pressure control unit, which plays a role of keeping the metering valve differential pressure constant, cannot follow up. Therefore, in this paper, we focused on the frequency of the pressure change waveform when the pump mode was changed and studied the model of the fuel system inlet pressure in the gear pump mode. Based on that model, we propose a method to reduce the frequency characteristics of the pressure change occurring at the time of pump mode change.

1. Introduction

Recent aviation turbofan engines are controlled by Full Authority Digital Engine Control (FADEC). The fuel metering system (FMS) receives electrical signals from the electronic control unit in the FADEC and meters and supplies the fuel flow to the engine. Often, FMSs use gear pumps as fuel pumps. However, the centrifugal pump (CP) is also used as a fuel pump in a relatively large amount, and in many cases, CP is used as a fuel pump for afterburner which consumes a large amount of fuel. In the case of using CP as a fuel pump, there is an advantage that the pump mass can be greatly reduced as compared with the case where a gear pump (GP) is used if the rated fuel flow rate is the same. On the other hand, CP suffers from the disadvantage that the fuel can hardly be pressurized at a



speed lower than the rated rotation speed. In general, the discharge pressure of CP is proportional to the square of the rotational speed. At a rotation speed of 30% of the rated rotation speed, the discharge pressure drops to 1/10 or less of the rated discharge pressure. In many turbofan engines, the fuel pump is driven by the gear of the accessory gearbox directly connected to the high-pressure rotor, starting the fuel control system from about 10% of the rated rotational speed. At this time, a pressure of 20% to 25% of the rated discharge pressure is required, so at low rotational speed it is impossible to start the engine with only CP. One way to compensate for this disadvantage is a method of freely controlling the rotation speed of the fuel pump with an electric motor, etc. Several research results have been reported [1] - [2]. In view of this, in order to supply fuel at a low speed, it is convenient that a small GP and a relief valve (RV) are combined to prepare a pressure supply source for starting, and in the case where the rotation speed is high and the fuel flow rate is large, CP is used as a pressure supply source. In this new type system that is assumed to be used, either pump is selected and used by the pump mode switching mechanism. Since each pump supplies different pressures in FMS, there is an operating mode to operate each pump through the short time to operate in the transient state of the mode change. In this research, in order to improve the transient characteristics of a new FMS for aviation turbofan engines, the authors focused on the frequency of the pressure change waveform and consider the model of the fuel system inlet pressure in GP mode when the pump mode changes. For that purpose, we modelled the pressure fluctuations at various points in the system, including the system entrance and conducted a system dynamics simulation. From the results based on that model, we proposed a method of reducing the frequency characteristic of the pressure change occurring at the time of mode change

2. A new fuel metering system having two pump modes

Figure 1. shows a diagram of the new FMS through simplifying the concept. The proposed new fuel metering system is part of a new fuel control system and uses two types of pumps: GP and CP. In this FMS, GP supplies a constant pressure in combination with RV and is used as a pressure source like CP. In FMS considered, the pump operating mode switching mechanism has the function of unloading the fluid power during the pump operation of the two pumps. In addition, this FMS has an advantage that it is possible to reduce an increase in fuel temperature due to self-heating based on fluid resistance in a fuel system operation region where the pump rotation speed and fuel flow rate change. Furthermore, by using a compact CP capable of discharging fuel at a high flow rate, it is possible to greatly reduce the weight as compared with the conventional fuel control system. Here, a case where CP is used as a hydraulic pressure source is defined as CP operation mode, and a case where GP and RV are used as a pressure source is defined as GP operation mode. Figure 2. (a) shows CP operation mode and Figure 2. (b) shows GP operation mode. FMS has a fuel pressure source upstream and fuel metering valve and throttle valve in series. The differential pressure of the fuel metering valve (MV) is always kept constant by the differential pressure sensing valve and the throttle. In CP mode, the discharge flow rate of CP flows into GP inlet pipe and RV return pipe by the switching mechanism, so the differential pressure between the inlet and the outlet of GP becomes zero and the operation state of GP is unloaded. After that, the inlet pressure and outlet pressure of GP are the same as the discharge pressure of CP. Changing the pump operation mode from CP mode to GP mode or from GP mode to CP mode causes the pressure source pressure to change significantly in a stepwise fashion, causing pulsation of the metered fuel flow rate to occur more easily. Reducing this pulsation is one of important technical problems.

3. Fuel Metering System

3.1. System overview

The main components of FMS are shown in Figure 3. FMS weighs the fuel flow demanded by the FADEC based on the formula of the orifice flow rate and supplies it to the engine. In the orifice, when the differential pressure between the upstream side and the downstream side is constant, a flow rate

proportional to the opening area passes through the orifice. Since the differential pressure control unit can keep the differential pressure of MV constant, the metered fuel flow rate is supplied to the engine by controlling the opening area of MV by FADEC. When the differential pressure sensing valve detects a deviation from the set differential pressure, the differential pressure control unit including the differential pressure sensing valve and the throttle valve operates, and as a result, the fuel is supplied to the throttle valve on the downstream side. The differential pressure of MV is compensated to a certain extent by increase and decrease of the throttle valve opening degree on the downstream side.

Fuel flow metered by FMS varies greatly depending on engine speed, engine inlet temperature and operating altitude within a flow range lower than the rated flow rate of the pump. When the discharge

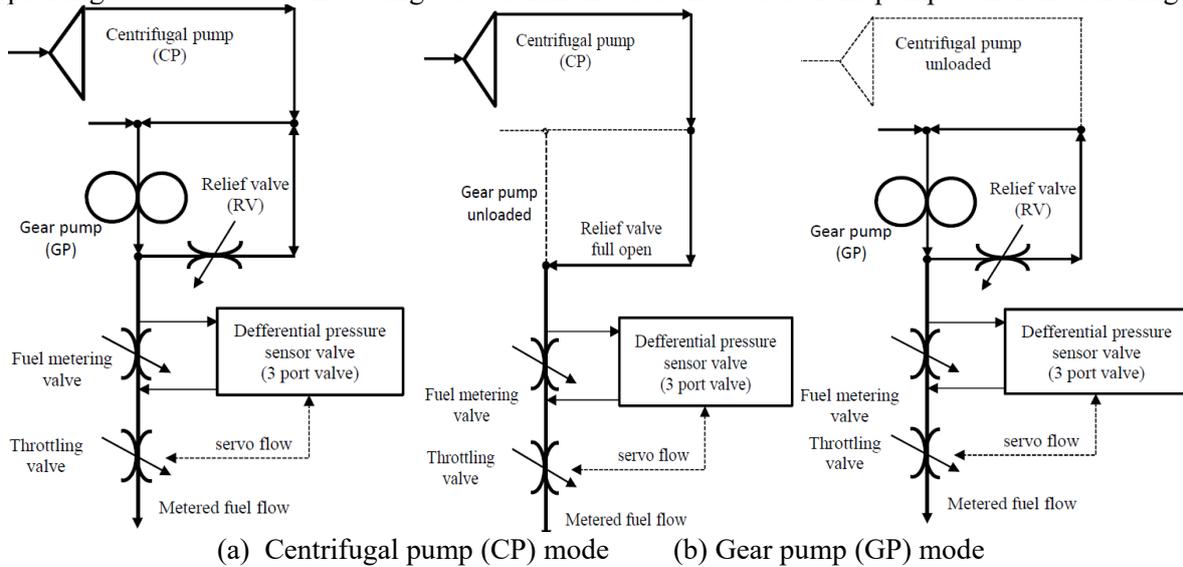


Figure 1. A simplified schematic diagram of a new FMS.

Figure 2. A simplified schematic diagrams of CP mode and GP mode based on Figure 1.

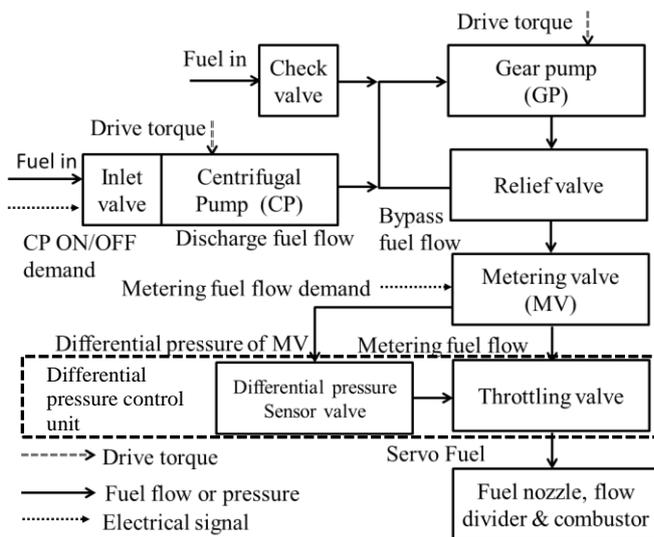


Figure 3. Fuel Metering System (FMS) configurations.

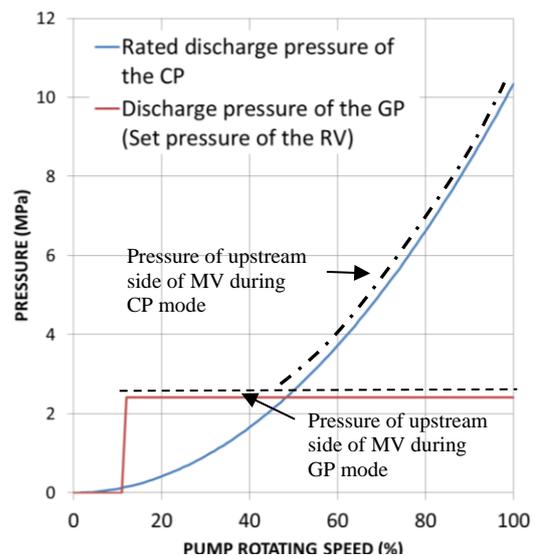


Figure 4. Pump speed V.S. discharge pressure of CP and GP.

flow rate is low, since the efficiency of CP is low, the temperature of the fuel easily rises. Especially when the flow rate is 1/20 or less of the rated fuel flow rate, excessive temperature rise of fuel occurs. Therefore, in the new fuel system proposed in this paper, when the fuel flow rate is low, fuel is supplied from GP so that the fuel temperature does not rise.

3.2. Causes and countermeasures of fuel pulsation generated at pump mode switching

Figure 4. shows the characteristics of the rated discharge pressure with respect to the rotational speeds of the two types of fuel pumps. The operating range of GP mode and CP mode is also shown. In CP mode, by setting the differential pressure between the upstream side and the downstream side of GP to zero, it is possible to set the fluid power of GP to almost zero. In GP mode, the fluid power of CP can be made substantially zero by closing CP inlet valve and rotating CP in steam or air without disk losses. This new FMS has the advantage that fluid power can be increased and decreased by mode switching, but since there is a large difference in the rated pressure in each mode, a large pressure fluctuation is liable to occur in the upstream pressure of the metered fuel at the time of mode switching, fuel pulsation is likely to occur in the system configuration. The pulsation of the metered fuel flow rate causes undesirable problems such as engine thrust pulsation, compressor surge and burnout of the combustor, so it is necessary to suppress the pulsation of the fuel flow rate within an allowable range.

As shown in Figure 5., we investigated a method of suppressing the disturbance of MV differential pressure, that is, the fluctuation of the metered fuel flow rate using the block diagram of the differential pressure control unit [9]. As a result, the following measures (1) - (3) are considered effective in reducing the sensitivity of the fluctuation of the metered fuel flow rate due to the pressure disturbance on the upstream side of MV.

- (1) Increase the responsiveness of the differential pressure control unit
- (2) Reduce the pressure fluctuation width on the upstream side of MV at the time of mode switching.
- (3) Eliminate high-frequency components that differential pressure control unit cannot follow from pressure fluctuation upstream of MV

In this research, we focused on measures (3) of these measures. In order to remove high frequency components from CP pressure, a model was created in consideration of CP discharge flow rate and RV outlet pressure. In order to remove the high frequency component from the large pressure change occurring when changing from GP mode to CP mode, it is necessary to accurately model the change of the discharge pressure of CP. Therefore, modeling was carried out focusing on the discharge flow rate of the pump. As a result, the model of the transient characteristic of discharge pressure of CP includes the following four characteristics (a) to (d). (a) The discharge flow rate of CP flows into GP inlet port, (b) after the pressure in the pipe rises, GP differential pressure becomes zero, and (c) GP is unloaded. (d) Thereafter, GP suction port pressure and the discharge pressure gradually rise to CP discharge pressure.

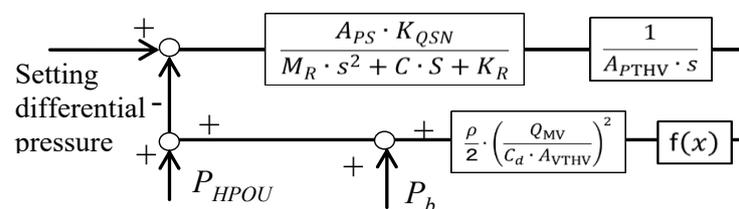


Figure 5. Block diagram of simplified the differential pressure control unit transfer function.

Figure 6. shows system diagrams of the fuel pressure circuit. In the schematic diagram of the fuel pressure circuit of Figure 6., the sum of these four fuel flow rates, that is, the sum of GP suction flow rate, the boost flow rate, RV return flow rate and CP discharge flow rate is time-integrated within the closed piping, through Equation (1). Therefore, by controlling the discharge flow rate of CP by the

flow rate control valve and sufficiently slowly changing the sum of the flow rate of the fuel in the closed pipe, it is possible to remove the high frequency component from the pressure change of GP suction port. As a result, it is possible to reduce the pulsation of the metered fuel flow rate by the differential pressure control unit. Specifically, high frequency components were removed from the flow control valve (FCV) displacement by removing the high frequency component from the input signal that controls FCV displacement using the first order lag filter in advance. As a result, high-frequency pressure components were removed from the displacement of FCV and the pressure of GP inlet. According to this, high frequency components can be removed from GP discharge pressure as well. From Equation (2), the bypass flow rate of RV can be predicted from GP discharge fuel flow rate, internal leakage and metering fuel. From Equation (3), it is also possible to estimate GP inlet flow rate from the metering fuel flow rate and the internal leak. Therefore, by controlling the flow rate of CP and slowly controlling the discharge pressure of GP within the range of inequality Equation (4), it is predicted that high frequency components can be reduced from the pressure fluctuation at pump unloading. After GP is unloaded, GP suction pressure changes according to Equation (1) until CP discharge pressure and GP suction port pressure are equal. The discharge flow rate of CP decreases as the differential pressure of GP approaches zero and gradually approaches the metered fuel flow rate.

$$P_{GPIN} = \int \frac{B}{V} \cdot (-Q_{GPIN} + Q_{BST} + Q_C + Q_{by})dt + P_{GPIN0} \quad (1)$$

$$Q_{GPOUT} = Q_{by} + Q_{MV} = Q_{HPIN} - Q_L \quad (2)$$

$$Q_C = Q_{MV} + Q_L - Q_{BST} \quad (3)$$

$$Q_C > Q_{GPIN} - Q_{BST} - Q_{By} \quad (4)$$

Figure 6. shows two system diagrams of the fuel pressure circuit. Figure 6. (a) shows a system circuit diagram which has been conventionally studied. Figure 6. (b) shows a system circuit diagram to be studied this time. In Figure 6. (b), a flow control valve is newly added downstream of CP. Relational expressions established in these schematic diagrams are the following expression (1) to (4). The sum of GP suction flow rate, the boost flow rate, RV return flow rate and CP discharge flow rate is time-integrated within the closed piping according to the equation (1), and becomes the discharge pressure of GP. Equation (2) shows the flow rate conservation law in the exit piping of GP and GP inside. From equation (2), the bypass flow rate of RV can be predicted from GP suction fuel flow rate, internal leakage and metered fuel. Equation (3) shows the relationship between boost flow rate, metered flow rate, CP discharge flow rate and internal leak. It is possible to estimate CP discharge flow rate using this equation. Inequality (4) shows the condition under which GP suction pressure rises. When this relation of inequality is satisfied, GP suction pressure rises according to equation (1). Therefore, in Figure 6. (b), by controlling the discharge flow rate of CP using the flow rate control valve (FCV), the suction pressure of GP can be gradually increased. Therefore, the differential pressure of GP gradually approaches zero, and the discharge flow rate of CP approaches the metered fuel flow rate. As a result, it is considered that the fluctuation of the metered fuel flow rate can be reduced by the differential pressure control unit. Specifically, if the high frequency component is removed from the input signal that controls FCV displacement using the first order lag filter, Specifically, if the high frequency component is removed from the input signal that controls FCV displacement using the first order lag filter, it is expected that high frequency components can also be removed from the response displacement of FCV and the discharge flow rate of CP, and consequently GP discharge pressure.

Figure 7. shows Amesim [10] model of FMS. In Amesim model, CP is modeled as an ideal pressure source which is defined as a function of the pump revolution speed and can supply the pressure regardless of the discharge flow rate, and GP discharge port is also modeled as an ideal flow rate supply source and GP suction port is a negative ideal fuel supply source which are defined as a function of the pump revolution speed and can supply the flow rate regardless of the discharge pressure. These pumps are modeled to output pressure or flow as a function of engine speed. Besides,

FCV is newly provided between CP discharge port and GP suction port. The boost pressure of GP and the fuel nozzle back pressure are modeled as an ideal pressure source. Dynamic characteristics are incorporated as models in the four kinds of valves: RV, differential pressure sensing valve, throttling valve, and fuel discharge valve. The displacement versus to opening area of these valve ports was given by the table. Fuel metering valves were modeled as simple variable orifices. Pipes connecting them is represented by a model considering only the volume. The simulation conditions are shown in Table 1 and Table 2.

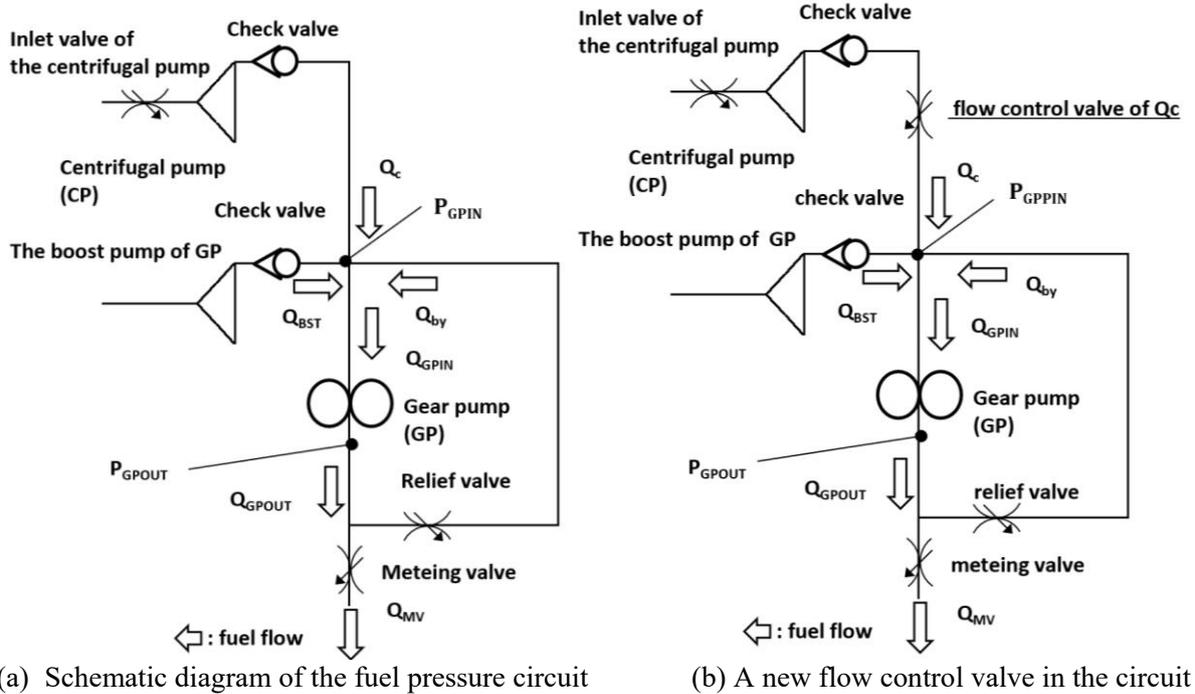


Figure 6. schematic diagram of the fuel pressure circuit.

Table 1. The conditions of the fuel system model.

Condition	Setting value
Rated fuel flow rate of the gear pump (GP)	100 (Litter/min)
Setting pressure of GP	2.758 (MPa)
Rated discharge pressure of the centrifugal pump (CP)	10.425 (MPa)
Rated fuel flow rate of CP	200 (Litter/min)
Differential Pressure of the metering valve	0.4137 (MPa)

Table 2. Model difference of a centrifugal discharge pressure source.

	Conventional model	New model
Model of a centrifugal discharge pressure source	Ideal pressure source which can supply RV downstream pressure regardless of the discharge flow with primary order time delay	Adding a flow control valve downstream of a centrifugal pump with primary order time delay to the conventional model

4. Simulation result and discussions

The simulation results are shown in Figure 8. (1), Figure 8. (2), Figure 9. and Figure 10.. In these simulations, at 4 seconds, the pump mode is switched from GP mode to CP mode, and at 6 seconds the mode is switched from CP mode to GP mode.

Figure 8. (1) shows the model switching simulation results of the original model. Figure 8. (1) (a) shows changes in MV upstream pressure, MV downstream pressure, CP discharge pressure, and GP outlet pressure. Figure 8. (1) (b) shows the fuel metering flow rate. The waveform of the discharge

pressure of CP at the time of mode switching passes through a low-pass filter of the first-order lag system with a time constant of 0.3 seconds, and high frequency components are removed. From this figure, it is understood that the pulsation of the fuel flow rate at the time of mode switching is very small like the conventional simulation result [9]. Since there is no flow rate through the control valve in the simulation of this model, the calculation result of the fuel flow rate passing through RV is shown as a reference in Figure 8. (1) (c) in order to compare with the flow rate passing. The simulation result of the new model is shown in Figure 8. (2). As can be seen from Figure 8. (2) (a), by controlling FCV to open loop, the pressure change at the time of mode switching is smoothly switched like the conventional model.

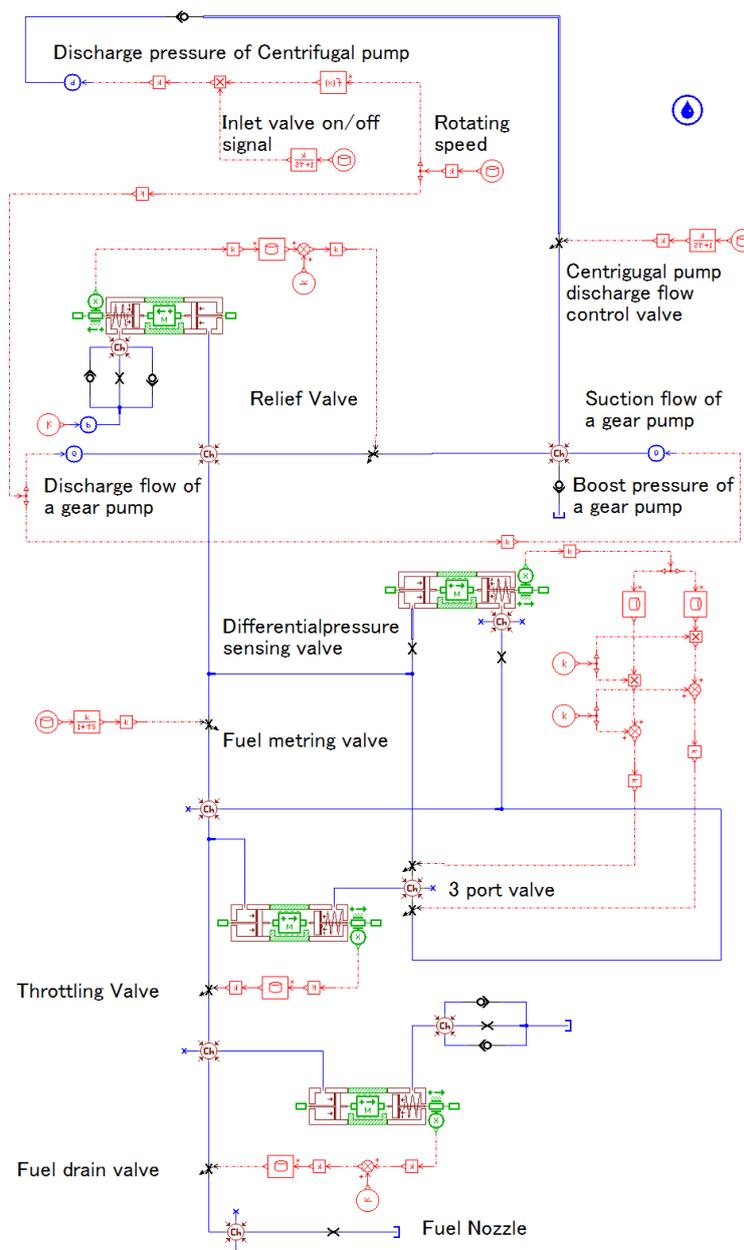


Figure 7. A New Amesim model of the fuel metering system with volume and Compressibility of fuel dynamics at a gear pump suction port.

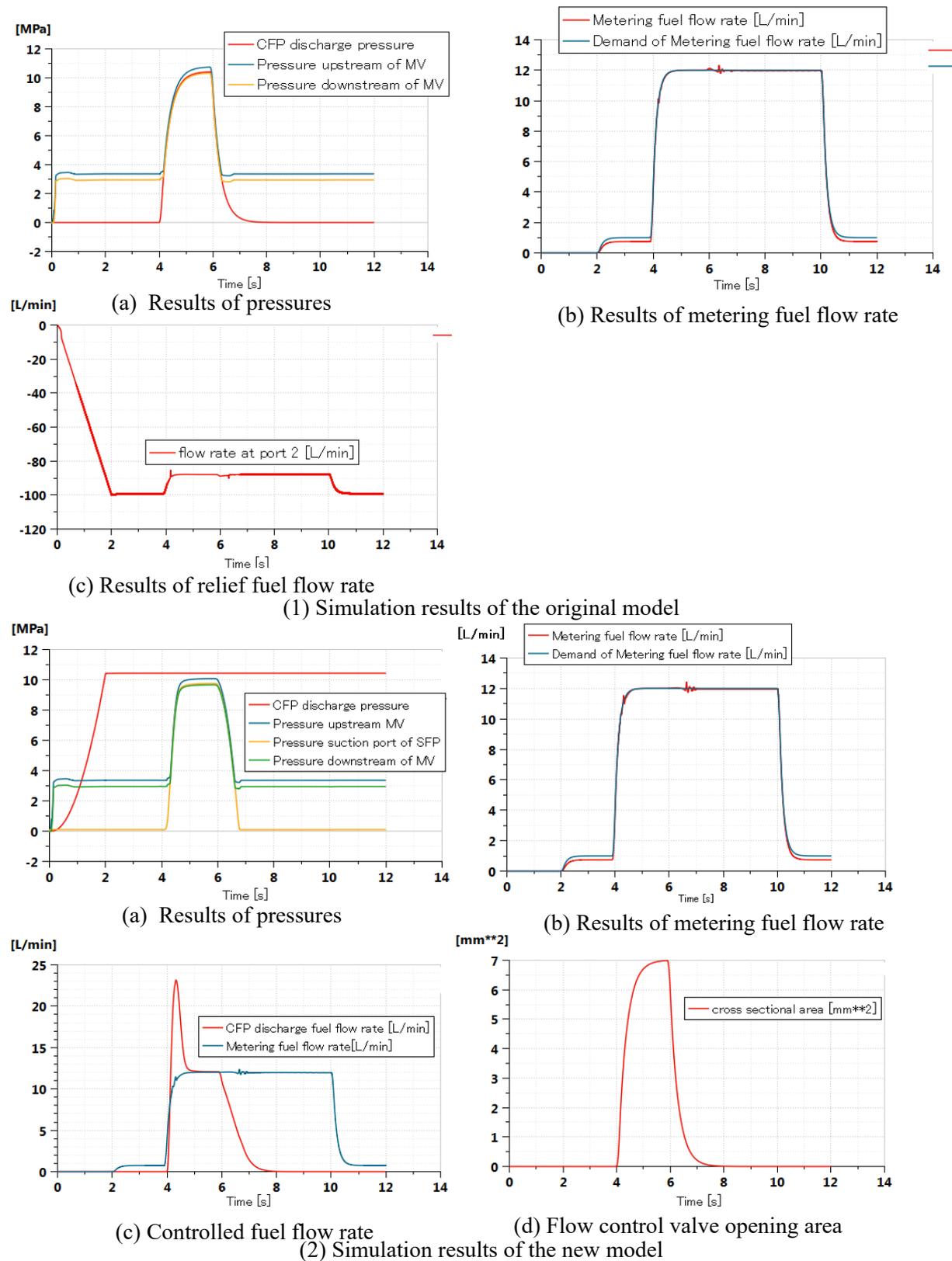


Figure 8. Simulation results of the pump mode switching.

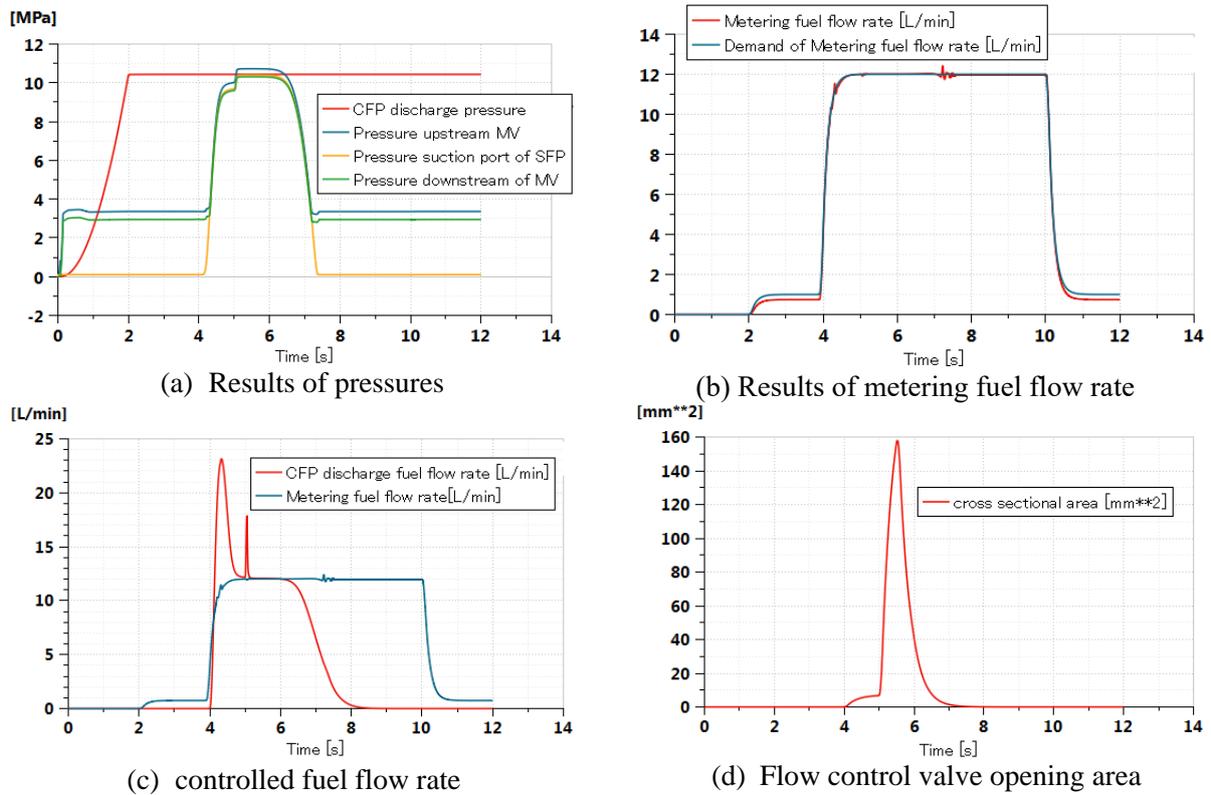


Figure 9. Simulation results of flow control valve full opening.

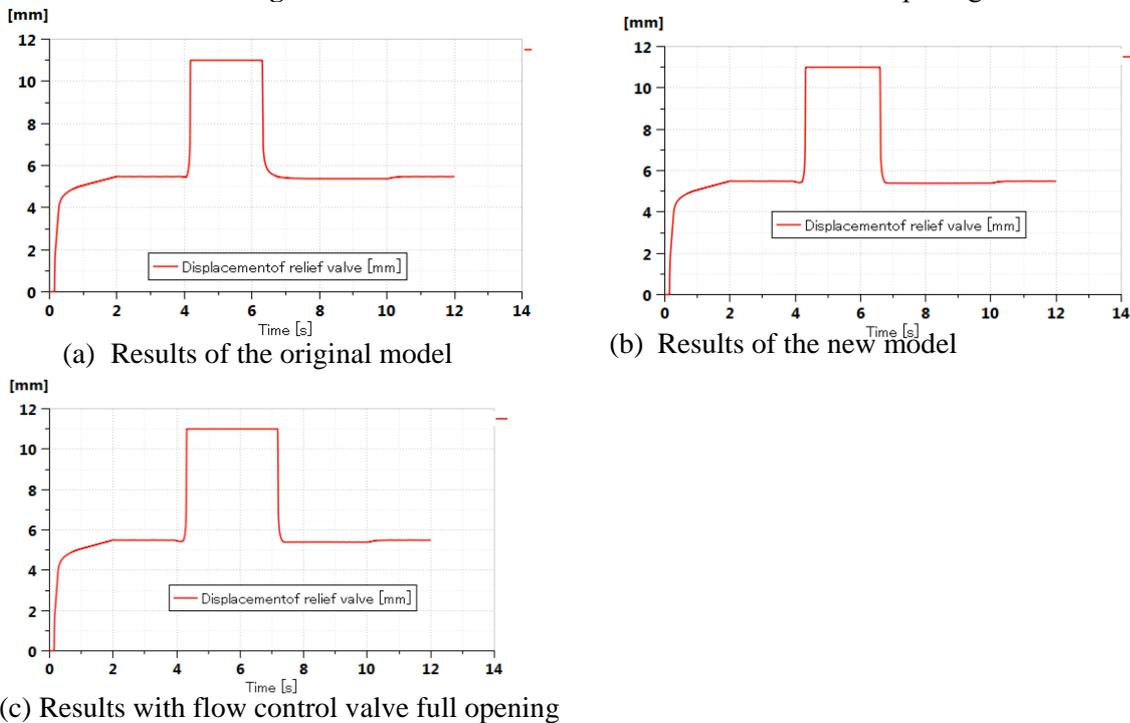


Figure 10. Simulation results of the relief valve displacement.

It can be confirmed from Figure 8. (2) (b) that the pulsation of the metering fuel flow rate at the time of switching mode is also small as in the conventional model. It can be seen from (c) and (d) that the discharge flow rate of CP has almost the same inflow rate as that of the metering fuel, so that the flow rate of the inflowing fuel is small and the valve opening area required for the flow rate control valve can be made very small. It is also confirmed from Figure 8. (2) (d) that moving the flow rate control valve opening area with the input valve displacement after removing high frequency by the low pass filter is effective for reducing the pulsation of the metering fuel flow rate.

Figure 9. shows a simulation result in which the opening area of FCV is greatly changed in order to correspond to the increase in the flow rate of the metering fuel after the changeover of the pump mode. From Figure 9. (a) and Figure 9. (d), it can be seen that FCV of the discharge flow rate of CP moves greatly at about 5 seconds, and the pressure changes on the step. Figure 9. (b) shows that the change in fuel flow rate is small even if the pressure is switched over to the step. Figure 9. (c) shows that the flow rate change when the flow rate control valve is widely opened is a spike-like flow rate pulsation. From this result, it is considered that change in metering flow rate was absorbed by the pipes.

Figure 10. shows the change in RV displacement in three simulation results. At nearly the same time as switching to CP mode, the valve displacement rapidly changes to Max. stop. This is because the sensitivity of the pressure displacement characteristics of RV spring is high, it is considered that the valve displacement instantaneously changes greatly when RV loses the regulating ability and the differential pressure applied to the valve increases. After RV is fixed to the Max. stop, the deflection of the metering fuel is relatively small with respect to the change of the pressure. In either simulation, there is no significant difference in displacement of RV. From these results, we found that fluctuation in the fuel flow rate at the time of mode switching can be reduced by controlling the flow rate flowing from CP to GP suction port when switching to CP mode.

In this simulation, only the switching transient characteristic is focused on. However, when a larger fuel flow rate is required, as shown in Figure 9., it is possible to supply a larger flow rate by fully opening FCV after switching. When the flow rate control valve is fully closed, even when CP is turned on and off, no fluctuation in the fuel flow rate occurs. Therefore, it is desirable that CP is turned on and off when the flow rate control valve is fully closed. Moreover, since the flow rate at the time of switching is relatively small, it is necessary to set the servo valve and the large on/off valve as a set instead of CP flow control valve, so that it is easy enough to perform the above control using the practical valve. However, the inlet valve of CP is indispensable for the unloading of CP. Adding a flow control unit causes a problem that the system becomes redundant. For practical use, it is considered necessary to sufficiently examine the trade-off between controllability improvement and redundancy.

5. Conclusion

In FMS with two kinds of pump modes, we devised a model to estimate the fluctuation of MV differential pressure against large pressure fluctuation generated during mode switching, taking into consideration the dynamic characteristics of piping model. Based on this model, a method to remove high frequency fluctuation from a pump was studied and its effectiveness was verified by simulation. Specifically, FCV was provided at the discharge port of CP to control the flow rate. As a result, it was possible to gradually change GP outlet pressure at the time of mode switching, and it was possible to eliminate the waveform of the high frequency component which cannot suppress the fluctuation by the differential pressure control unit. By combining the method proposed in this study with the improvement of the responsiveness of the differential pressure control part, it is expected that the pulsation of the fuel flow rate can be further reduced.

NOMENCLATURE

A_{PTHV}	pressure receiving area of the throttling valve
A_{PS}	pressure receiving area of the sensor valve
A_{VTHV}	opening area of the throttling valve port

B	bulk modulus of the jet fuel
C	coefficient of viscosity on relief valve piston
C_d	flow coefficient
$f(X)$	Throttling valve Opening area function of the valve displacement
K_{OSN}	flow gain of the differential pressure sensing valve
K_R	spring constant of the differential pressure sensing valve
M_S	Mass of the sensor valve piston
P_b	back pressure of the metering valve
P_{GPIN}	inlet pressure of GP
P_{GPIN0}	inlet pressure of GP (initial condition)
Q_{GPIN}	suction fuel flow rate of GP
Q_c	discharge fuel flow rate of CP
Q_{by}	RV bypass flow rate
Q_{BST}	the boost fuel flow rate
Q_{MV}	the metering fuel flow rate
Q_L	the internal leakage of GP
Q_{GPOUT}	the discharge fuel flow rate of GP
S	Laplace operator
V	Volume of the piping between MV and throttling valve
X	displacement of the throttling valve
ρ	density of the jet engine fuel
Abbreviation	
CP	centrifugal pump
GP	gear pump
FCV	flow control valve
FMS	fuel metering system
MV	metering valve
RV	relief valve

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