Technology Explanation

Characteristic analysis of high-efficiency hydraulic system

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Abstract

Exhaust gas regulations for passenger cars have recently become stricter year by year as many countries promote carbon-neutral activities from the viewpoint of the environment and energy saving.

In response, manufacturers have sought the development of transmissions with higher fuel efficiency as one solution as they accelerate their efforts to improve the overall efficiency of their vehicles.

KYB has recently improved the efficiency of the pump used as the power source for the transmission, reaching a level approaching its maximum.

In this study, as a means of confirming the effectiveness and feasibility of this hydraulic system, we modeled the entire system and built simulation technology that enables performance prediction. In addition, by utilizing our simulator we clarified the design parameters that affect the performance and set the prospects for improving the performance of the hydraulic system by coordinating the control of each component.

1 Introduction

Recently, many countries have promoted carbon-neutral activities from the viewpoint of the environment and energy saving. Exhaust gas regulations for passenger cars have also become stricter year by year. In response to this trend, the automobile industry has accelerated its efforts to improve the overall efficiency of vehicles. This means that automobile transmissions are required to deliver higher efficiency.

The transmission uses a pump as its hydraulic power source. Reducing the output from the pump section (specifically, reducing any excessive flow discharged from the pump) will help improve the fuel efficiency. That is why KYB has conventionally promoted efforts to improve pump efficiency. However, there is limited scope for further improvements in efficiency of the pumps alone. Then, to achieve further improvement of efficiency in the pump section, this study focused on a hydraulic system that combines two or more pumps and selector valves and tried to confirm the feasibility.

As a means of confirming the effectiveness and feasibility of this hydraulic system, we established an 1D simulation [Note 1)] technology that enables performance prediction. This paper explains the effort and the results of its evaluation.

Note 1) A method of simulation that expresses phenomena of products and components mathematically and calculates them spanning two or more physical phenomena.

2 Overview of KYB Hydraulic System

As shown in Fig. 1, the hydraulic system that we developed in this study (hereinafter "the KYB hydraulic system") consists of a hydraulic pump driven by an engine (hereinafter "the mechanical pump"), another hydraulic pump driven by a separate electric motor (hereinafter "the electric pump"), and a change-over mechanism consisting of a solenoid and a selector valve. The mechanical pump has two ports: main and sub. These ports can be switched over by the change-over mechanism to deliver two-stage variable discharge.

The following describes the principle of operation of the KYB hydraulic system. Normally, the discharge rate of the mechanical pump changes in proportion to the speed of the engine. In other words, the pump may

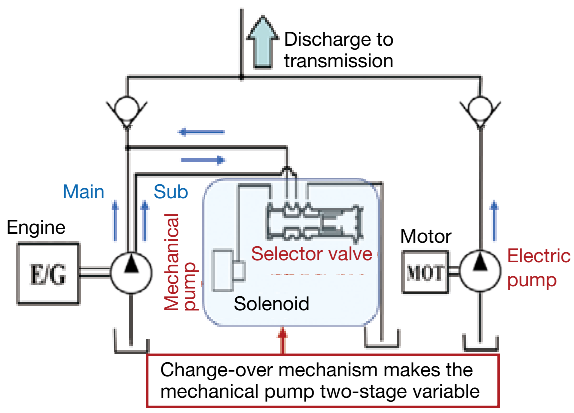


Fig. 1 KYB hydraulic system

discharge more flow than required depending on the operation mode of the vehicle. This increased pump output leads to lower fuel efficiency. The KYB hydraulic system is designed to solve this problem. In an operation mode where the transmission requires a low flow rate, the system diverts the discharge from the mechanical pump (via the sub port) (hereinafter "the sub discharge") to a tank via the selector valve and lets the discharge from the mechanical pump (via the main port) (hereinafter "the main discharge") alone flow to the transmission section, thereby reducing the output of the mechanical pump. In an operation mode where the transmission requires a high flow rate, the solenoid is activated to operate the selector valve to merge the sub discharge with the main discharge, ensuring normal operation of the transmission. Moreover, some operation modes of the vehicle, such as kick down, may quickly require a high flow rate. In this case, the electric pump is activated in addition to the mechanical pump, to compensate for the insufficient flow rate. In this way the KYB hydraulic system can supply the exact amount of flow required to the transmission with fine control. Compared with the system consisting of a single mechanical pump (hereinafter "the single mechanical pump system"), the KYB hydraulic system can be expected to substantially reduce the pump output, namely, improve the fuel efficiency of the vehicle.

3 Feasibility Study

We checked in advance the effectiveness of the KYB hydraulic system by conducting a system performance evaluation using a simplified 1D simulation model. This chapter describes the details of this effort. The 1D simulation model we used was commercially available Simcenter™ Amesim™ software (SIEMENS, Germany).

3.1 Overview of Simulation Model

The simulation model basically consists of the following three major sections as shown in Fig. 2:

[1] Flow control section (hydraulic system): Uses the engine speed as input and controls the pump flow rate through electronic control of the solenoid and motor.

[2] Pressure control section: Uses two or more voltage/electromagnetic valves to control the discharge pressure from the hydraulic system (hereinafter "the control pressure").

[3] Transmission control section: Controls the transmission gear ratio according to the control pressure.

To determine the effect of pump output reduction in this simulation, driving data (engine speed and control pressure) measured in the automobile's fuel efficiency evaluation mode was taken as input conditions. Then, the pump discharge with these input conditions was calculated. In this study, we modeled two systems: the KYB hydraulic system and the conventional single mechanical pump system. By comparing the pump output

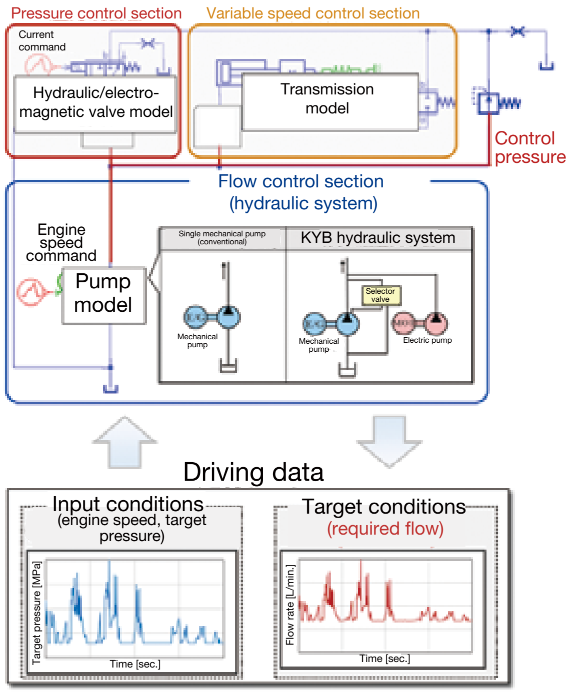
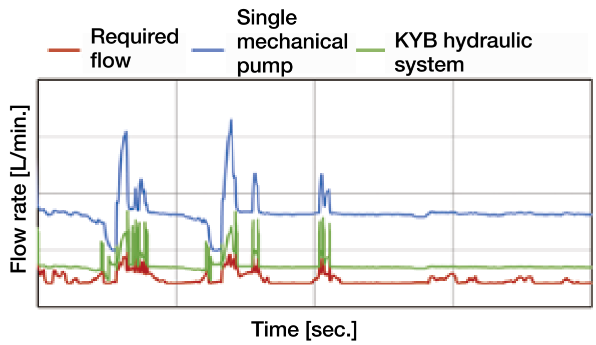


Fig. 2 Overview of simulation model

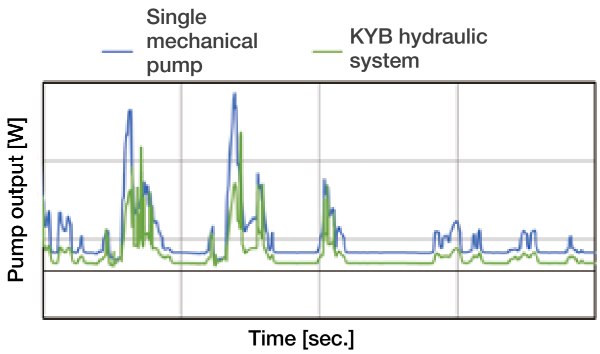
calculated in the same operation pattern between the two systems, we checked the effectiveness of the KYB hydraulic system.

3.2 Confirmation of Effectiveness of KYB Hydraulic System

Fig. 3 shows a simulation comparison between the single mechanical pump system model and the KYB hydraulic system model (flow rate and pump output). According to Fig. 3(a), both models meet the flow rate required for the transmission. The Figure also indicates that the KYB hydraulic system successfully curbs the flow rate over a wider range compared with the single mechanical pump system. The KYB hydraulic system achieves this by reducing the discharge from the mechanical pump by directly connecting the sub pump to the tank via the selector valve in the time zones where the single mechanical pump system has a highly excessive flow. Accordingly, the KYB hydraulic system is superior to the single mechanical pump system in reducing the pump output over the entire time zone as shown in Fig. 3(b), showing a pump output reduction rate of 48.3%. Thus, the effectiveness of the KYB hydraulic system was confirmed.



(a) Flow rate



(b) Pump output

Fig. 3 Simulation comparison results

4 Challenges and Countermeasures of KYB Hydraulic System

The KYB hydraulic system is susceptible to a surge pressure occurring when the selector valve is changed over, as shown in Fig. 4. Upon the change-over, the sub discharge will flow into the main line to merge with the main discharge, causing an abrupt change in pressure (surge pressure) in the circuit. This surge pressure appears as a shock-like kick back in the vehicle. It is desirable to reduce the surge pressure as far as possible. We then conducted, against a hydraulic circuit with a selector valve that may cause a surge pressure, a comparative evaluation of simulation and actual vehicles, a parameter study [Note 2)] using the simulation, and optimization [Note 3)] of the valve form. This chapter describes the details.

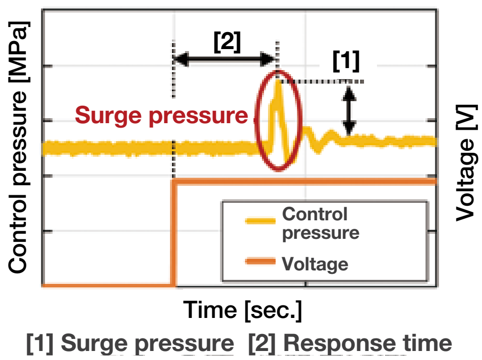


Fig. 4 Surge pressure caused by valve change-over

Note 2) To set analytical models and conditions as parameters and repeatedly conduct analysis while changing the parameter values.

Note 3) An approach to find a best parameter value by minimizing or maximizing a design target (an objective function) of a system.

4. 1 Comparative Study of Surge Pressure

Photo 1 shows a valve change-over test machine and Fig 5 shows its hydraulic circuit diagram. The hydraulic circuit for this test has a check valve on the sub discharge line in parallel with the mechanical pump and a selector valve on the sub discharge line at a further position. The sub discharge flows into the tank when the selector valve is open (hereinafter "the selector off") or flows through the check valve to merge with the main discharge when the selector valve is closed (hereinafter "the selector on"). In this test, a surge pressure was measured when the selector valve was operated with the pump running at a certain speed to deliver a certain discharge pressure. We also created a simulation model of the same configuration as that of the actual test machine to compare the simulation results with the test results. In this comparative evaluation of the surge pressure, the magnitude of surge pressure and the response time after the change-over (the time from the issuance of a current command to the occurrence of a surge pressure) were selected as evaluation items.

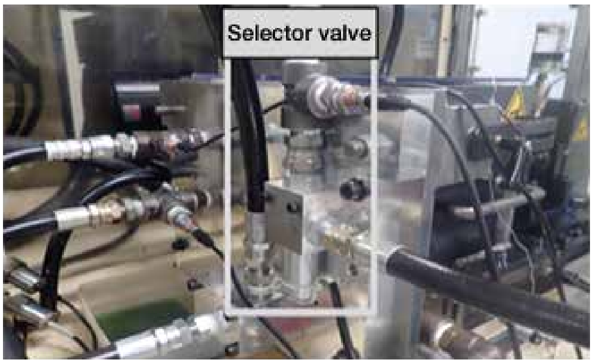


Photo 1 Valve change-over test machine

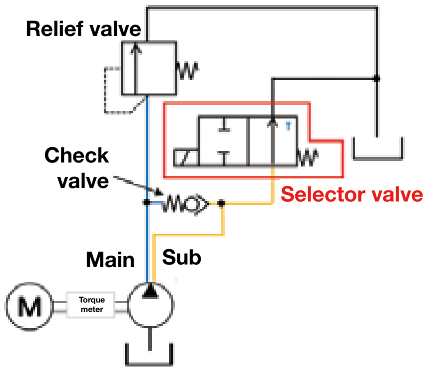


Fig. 5 Hydraulic circuit of valve change-over test machine

Fig. 6 shows a comparison of pressure at the time of change-over between the simulation and the actual machine test. According to Fig. 6, the simulation results closely matched those of the actual machine test with errors within just ±3% for all the evaluation items except response time (the time from the selector on to the selector off) whose error was 8.8%. It was confirmed that the simulation can reproduce the behavior of the actual machine with an accuracy within ±10% for all the evaluation items.

Thus, the simulation model we built in this study proved its high validity, making it possible to predict the surge pressure upon change-over of the selector valve.

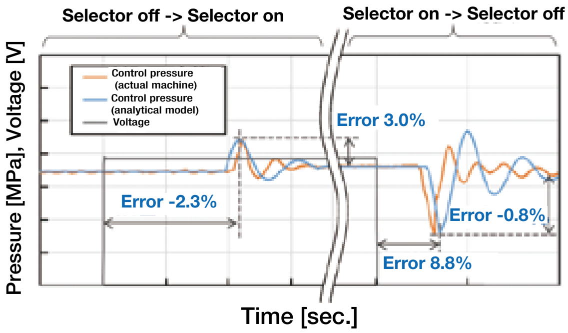


Fig. 6 Comparison of selector valve between analytical model and actual machine

4. 2 Implementation of Parameter Study

To identify design parameters that may affect the surge pressure and/or system response time, we carried out a parameter study using the simulation model created in section 4.1.

Parameter values were adjusted by halving or doubling the design median value. Otherwise, parameter values were changed to an extent that would not hamper the change-over function. We eventually conducted simulation in several tens of patterns. Table 1 shows an example of sensitivity trends of design parameters that affect the surge pressure and/or system response time. The positive (+) and negative (-) signs in Table 1 indicate an increase and a decrease in the relevant performance/characteristics respectively.

Table 1 Parameter sensitivity for various evaluation items (example)

Target model	Parameter change	Evaluation item			
		Surge pressure performance (Off -> On)	Responsivity (Off -> On)	Surge pressure performance (On -> Off)	Responsivity (On -> Off)
Piping	Diameter up	(+) sensitivity: High	(-) sensitivity: Middle	(+) sensitivity: High	(-) sensitivity: Low
	Length up	(+) sensitivity: Middle	(-) sensitivity: Middle	(+) sensitivity: Middle	(-) sensitivity: Low
	Rigidity up	(-) sensitivity: High	(+) sensitivity: Middle	(-) sensitivity: High	(+) sensitivity: Low
Selector valve	Opening characteristics	Sensitivity: High	Sensitivity: Low	Sensitivity: Middle	Sensitivity: Low
	Spool friction up	(+) sensitivity: Middle	(-) sensitivity: Middle	(+) sensitivity: Low	(-) sensitivity: Low
	Spool diameter up	(+) sensitivity: High	(-) sensitivity: High	(+) sensitivity: Middle	(-) sensitivity: Middle
	Spring force up	(-) sensitivity: High	(+) sensitivity: High	(+) sensitivity: Low	(-) sensitivity: Low
Check valve	Spool diameter down	No effect	No effect	(+) sensitivity: High	No effect
	Lap length down	No effect	No effect	(+) sensitivity: Middle	No effect

From the results shown in Table 1, we obtained insights about the overall sensitivity trend. The surge pressure performance is substantially affected by the piping and check valve while the response time is substantially affected by the selector valve and piping.

In the piping model, the surge pressure performance is higher with lower rigidity, a larger diameter, or a longer length. This is probably because the piping has an increased accumulator effect to be more capable of absorbing surge pressures. In the check valve model, the surge pressure performance is higher with a smaller spool diameter or a shorter lap length. On the other hand, a back flow is more likely to occur when the check valve is closed, which may degrade the robustness [Note 4)] as a system. This issue should be taken into account during design. Description of the selector valve model is omitted here and will be included in the next section.

From the results of the parameter study, we also confirmed that there is basically a trade-off relationship between the surge pressure performance and the responsivity.

Note 4) A property of a system or structure that it is unlikely to be affected by various disturbances.

4. 3 Study on Valve Design Optimization

From the results of the parameter study in section 4.2, we determined that the opening characteristics of the selector valve and the spring specifications (including spring constant and initial load) substantially affect the surge pressure performance and responsivity (see Table 1). In light of this, we optimized the form of the selector valve to enhance the valve change-over performance. The following describes the details of the optimization. Please note that, prior to optimization, we reviewed the valve configuration by simulation and then modified the configuration so as to hopefully deliver even higher change-over performance.

Table 2 shows predictor variables, objective functions, and limiting conditions for optimization. Since there is basically a trade-off relationship between surge pressure performance and responsivity as indicated by the results in section 4.2, the objective function was set so as to optimize the sum of the weighted surge pressure and response time, thereby allowing the valve to have both good surge pressure performance and good responsivity. In terms of limiting conditions, some parameters that cannot help having errors for design reasons, including stroke and spring constant, were given variations. Then, it was judged whether the sub discharge was diverted properly when the valve was changed over. This made the selector valve, which had already been optimized, more robust.

Table 2 Parameters used to optimize the valve design

Predictor variable	Objective function	Limiting condition
- Underlap - Stroke - Spool diameter - Orifice - Spring constant - Initial load - Opening notch	- Evaluation function expressed in surge pressure and response time	- Judgement of the valve's change-over operation with variations in design values

Fig. 7 shows a comparison of surge pressure/response time (the selector off -> the selector on) before and after optimization. It can be confirmed in Fig. 7 that the optimized model shows substantially lower surge pressures and shorter response time than the valve change-over model created in section 4.1. This analysis method was found to be effective for finding optimal design parameters that can improve both surge pressure and response time at the same time.

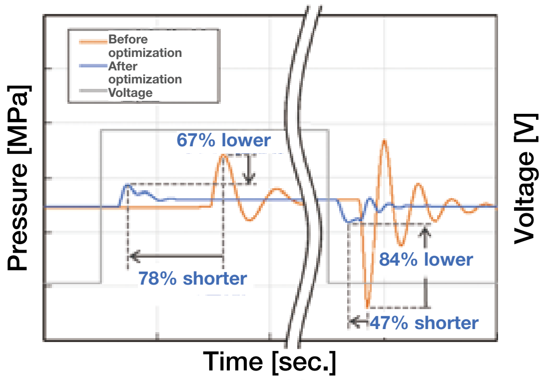


Fig. 7 Evaluation of surge pressure characteristics of optimized model

4.4 Performance Evaluation of Optimized Selector Valve on Actual Machine

To determine the effect of the optimized selector valve obtained through simulation, we conducted a confirmation test on the actual machine. Fig. 8 shows the surge pressure and response time of the original selector valve in section 4.1 and of the optimized selector valve. According to Fig. 8, the optimized selector valve shows about 80% lower surge pressure and about 40% shorter response time than the original counterpart.

This clarified that the optimized selector valve can also substantially improve both properties in the actual machine in the same manner as for the simulation. We thus confirmed the validity of the simulation and the effectiveness of the optimization program.

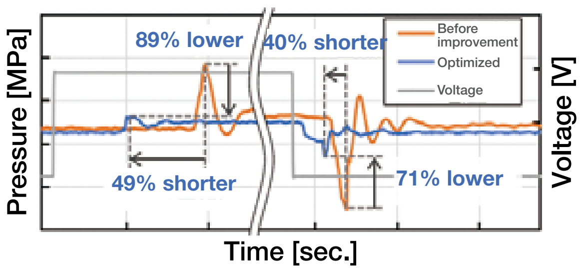


Fig. 8 Comparison of results of surge pressure tests

5 Building of an Electric Pump Model

The KYB hydraulic system model includes an electric pump in addition to the mechanical pump and the selector valve. The electric pump model used in the feasibility study in Chapter 3 was specified to operate without delay according to speed commands. Efficiency and responsivity were not factored into the model design. This study in turn requires a model for the KYB hydraulic system that can determine feasibility and efficiency of the system operation. Therefore, the electric pump, which is one of the system components, also needs to reproduce its efficiency and response time.

Then, we modeled the electric pump, which is one of the components to be installed in the KYB hydraulic system, based on data collected on the actual machine. This chapter describes the details of this effort. The target model accuracy for efficiency and response time was set to within ±10%.

5.1 Evaluation with Initial Electric Pump Model

Fig. 9 shows the configuration of the actual electric pump. The pump section includes a PMSM [Note 5)] and a vane pump and uses torque maximizing control [Note 6)] and flux weakening control [Note 7)] strategies to control the motor current. The electric motor was modeled using simulation software products MATLAB® and Simulink® [Note 8)], which are often used to build control models, instead of the Simcenter Amesim used to model the hydraulic equipment. By coupling the motor and pump models we had created, we conducted an analysis of the electric pump. In terms of pump operation, the actual speed of the pump motor was fed back and controlled to minimize deviations from the speed commands.

Note 5) Acronym for Permanent Magnet Synchronous Motor. PMSM is driven through synchronization between the motor rotation and the frequency of the AC current flowing into the coil.

Note 6) A motor control method that selects from the same effective current to maximize the output torque.

Note 7) A motor control method that sends a current in a direction that cancels the magnetic flux of the permanent magnet, enabling high-speed operation while reducing the torque.

Note 8) MATLAB® and Simulink® are registered trademarks of The MathWorks, Inc.

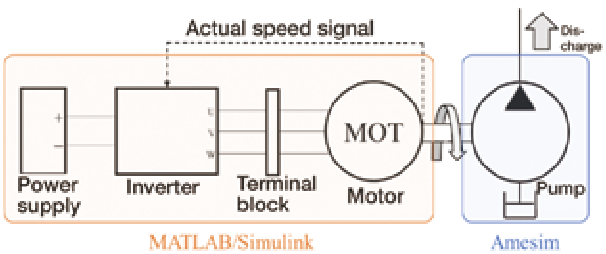


Fig. 9 Configuration of electric pump

Fig. 10 shows equivalent circuit diagrams of the PMSM circuit converted through coordinate transformation to the d-axis (in the direction of the main flux) and the q-axis (shifted from the d-axis by 90˚). In the coordinate transformation shown in Fig. 10, the motor current can be divided into a component I_q that generates a motor torque and a component I_d that causes the rotor to produce a magnetic flux. These current components are individually controlled. Modeling of the electric motor in this study was based on the electric circuit diagrams shown in Fig. 10. Parameter settings for the model used measurements of the actual machine, which is the comparison target, and calculated values from the specifications.

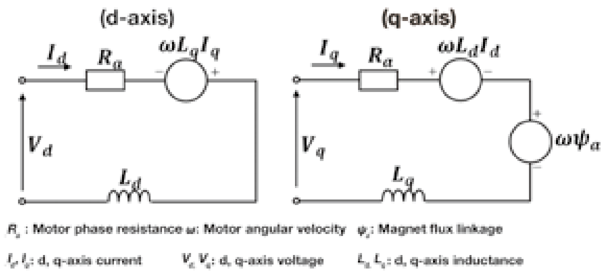


Fig. 10 Equivalent circuit diagrams of motor

Table 3 shows the results of comparative tests of electric pump efficiency between the model and the actual machine. Table 3 indicates that the simulation gives higher efficiency than the actual machine under all the test conditions. This is probably because the motor model was created by considering electrical and mechanical losses, but not considering magnetic hysteresis and eddy currents. The efficiency errors may be attributable to the fact that the dependency of motor inductance on current and temperature was not taken into account. The magnitude of efficiency error is not less than 10% on the whole and over +100% under the high discharge pressure and low speed conditions in particular. Thus, the efficiency of the pump model was found to differ largely from that of the actual machine.

Table 3 Efficiency evaluation of initial electric pump model

	Test conditions (speed and discharge pressure)		
	Low speed, High pressure	Mid speed, Mid pressure	High speed, Low pressure
Model efficiency error	+113%	+12%	+14%

Fig. 11 shows the results of comparative tests of electric pump responsivity between the model and the actual machine. In this study, the pump responsivity was evaluated by the time from the step time of the speed command signal (start time) until the discharge flow reaches 90% of the normal flow rate.

According to Fig. 11, the response time of the analytical model was about twice (about +100% error) that of the actual machine. With such a big difference, it was considered that the model must be improved.

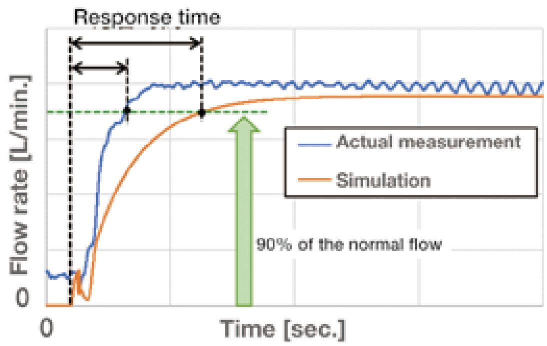


Fig. 11 Evaluation of responsivity of initial electric pump model

5. 2 Improvement of Motor Model

Based on the results in section 5.1, we reviewed the motor model and parameters with consideration given to iron loss. Fig. 12 shows d, q-axis equivalent circuit diagrams with iron loss taken into account [2)]. The equivalent circuit diagrams in Fig. 12 express the iron loss caused by rotor rotation in equivalent iron loss resistance R_{cp}. Also, L_{dq} and L_{qd} are used to reflect the d, q-axis mutual inductance. We then created another motor model based on these equivalent circuit diagrams. Since the equivalent iron loss resistance R_{cp} shown in Fig. 12 is a variable parameter relative to I_d, I_q, or the motor speed N, the iron loss obtained in the actual machine test was used to formulate an approximate expression, which was applied to the model. The inductances were set in table data as I_d/I_q-dependent parameters and then subjected to temperature correction before insertion in the equations. The motor temperature was set to a value predicted from the actual machine data.

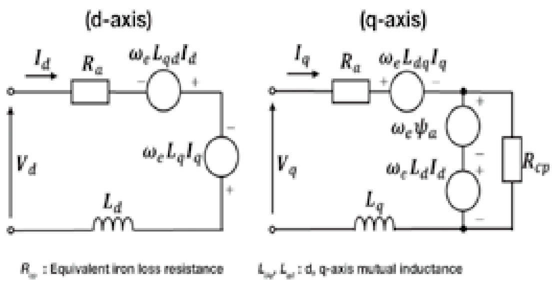


Fig. 12 Equivalent circuit diagrams of motor with iron loss taken into account

Fig. 13 shows the T-N (torque and number of revolutions) characteristics of the actual motor, the motor model in section 5.1 (hereinafter "the initial motor model"), and the motor model in this section (hereinafter "the improved motor model"). In Fig. 13, the horizontal axis indicates motor speed and the vertical axis represents motor torque. The torque-speed (T-N) characteristics are plotted for each of the effective values I_e of the three-phase current.

According to Fig. 13, there are two distinctive sections: a constant torque section and a section where the torque decreases as the speed increases. In this chapter, these sections are called by their current control method: the constant torque section is called the torque maximizing control zone and the slanted graph section is called the flux weakening control zone. When the results of the actual machine are compared with the results of the initial motor model for the various common current values in Fig. 13, the analytical model closely reproduces the graph gradient, namely, changes in torque, but the analysis values are higher than the measurements by about +5% to +10% over the entire torque range. On the other hand, the improved motor model shows motor characteristics with an accuracy as high as within ±3% error for all the current values in the torque maximizing control zone. Also in the flux weakening control zone, the error is limited to within about ±10%. Thus, we successfully built, through improvements including the addition of iron loss, an improved motor model that can reproduce the characteristics of the actual motor with a higher accuracy than the initial motor model.

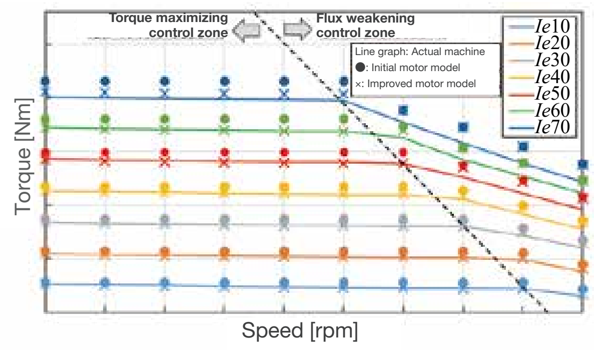


Fig. 13 Evaluation of T-N characteristics of motor models

5.3 Improvement of Pump Model

Fig. 14 shows an example of measurements of motor speed and discharge flow rate of the actual electric pump. When the actual motor speed is compared with the discharge flow rate in Fig. 14, there is a delay between the start-up of the motor and the start of oil discharge. From this finding, we assumed the following pump behaviors:

[1] After the electric pump is started, the pump vanes remain retracted delivering no flow or no discharge pressure until the pump speed reaches a certain level.

[2] When a certain speed level is reached, the pump vanes extend to produce a flow and build a discharge pressure accordingly.

[3] When the running pump (motor) is stopped, the vanes retract to depressurize the pump.

We designed a pump model that can reproduce behaviors [1] to [3] above for modeling with an operation delay taken into account. On the assumption that the timing for the vanes to extend depends on the angular velocity or angular acceleration of the pump, we determined an equation to obtain the extension timing of the vanes based on the actual machine's data, and then incorporated a mathematical model for calculating the start-up and response times of the pump model. Of course, we could reproduce the response of the pump model by incorporating components including vanes and cam rings. In this study, however, we did not select this approach because:

[1] the calculation load is very high, particularly it takes enormous time to analyze the fuel efficiency;

[2] it is necessary to identify unknown parameters such as the coefficient of friction of the vanes, and;

[3] the contribution of the pump section to the delay of the entire electric pump is low.

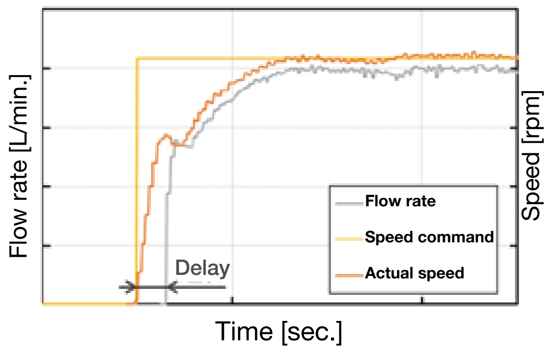


Fig. 14 Actual measurements of pump operation

Fig. 15 shows the pump model we created. By incorporating models for determining the vane state (retracted or extended), we built a pump model that can simulate a delay from the motor start-up until the shift to the hydraulic circuit with the extended vanes, and then incorporated the model into the electric pump model.

5.4 Evaluation of Improved Electric Pump Model

The combined electric pump model created by incorporating the improved motor model in section 5.2 and the pump model in section 5.3 was evaluated for efficiency and response time.

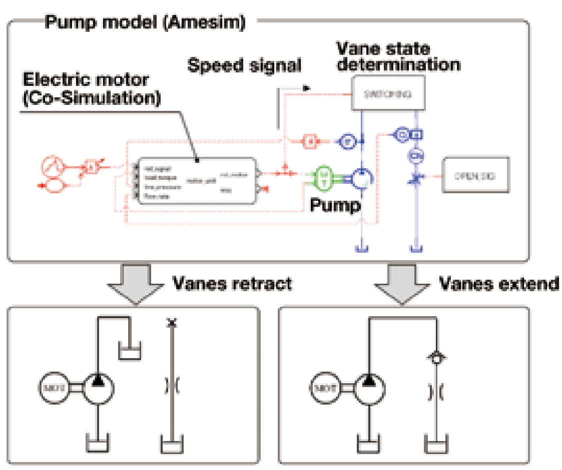


Fig. 15 Pump model with vane behavior taken into account

Table 4 shows a comparison of efficiencies during operation of the electric pump. It should be noted in Table 4 that the efficiencies of the overall assembly (electric pump) are actual measurements calculated from the actual pump output, while the efficiencies of various components are estimates calculated from theoretical equations.

The efficiency error of the entire assembly is within ±10% under any of the conditions. In particular, the error for the mid-pressure and mid-speed operation or the low-pressure and high-speed operation is within ±3% not only for the entire assembly but also for all components, proving a very high model accuracy. For the high-pressure and low-speed operation, the entire assembly has -5.5% error, but some components show around 10% error. These quite big errors are probably because the model accuracy is low for rotary motion of components "between the motor and the pump" downstream of the electric pump system.

Thus, the model still needs to be improved, but was confirmed to reproduce the pump with high accuracy particularly for the entire assembly with an efficiency error of about ±5%.

Table 4 Evaluation of efficiency of improved electric pump model

		Operating conditions		
		High pressure, Low speed	Mid pressure, Mid speed	Low pressure, High speed
Model efficiency error	Between the power and the inverter	−3.0%	−1.2%	−0.1%
	Inside the inverter	−9.4%	−2.5%	0.5%
	Between the inverter and the motor	0.2%	1.0%	1.2%
	Inside the motor	−4.1%	−1.7%	−2.1%
	Between the motor and the pump	12.8%	2.0%	2.6%
	Inside the pump	−0.8%	−0.2%	0.0%
	Entire assembly (electric pump)	−5.5%	−2.7%	2.0%

Next, we evaluated the response time of the electric pump model. The response time of the pump is affected not only by the discharge pressure and pump speed but also by the P control gain [Note 9)] setting for motor speed, which was added to the test conditions. In the testing, the P control gain setting for speed control was adjusted at three levels: low, middle, and high.

Fig. 16 shows a comparison between the actual machine and simulation regarding the response time of the electric pump under high-speed, low-pressure operation with the high P control gain setting. Fig. 16 reveals that the simulation almost reflects the response time as well as the flow rate waveform including the timing of the rising edge of the flow rate and the process to reach the steady state.

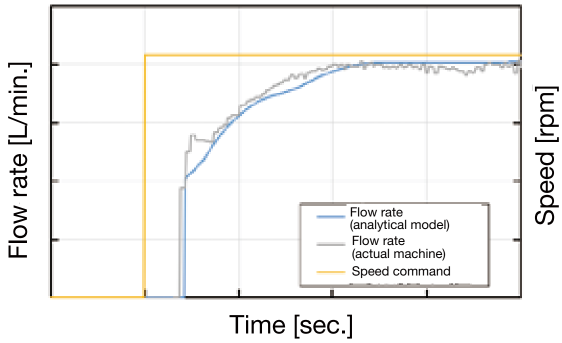


Fig. 16 Electric pump discharge flow rate over time

Note 9) A control gain for adjusting the control input in proportion to the difference between the target and current values of the controlled object

Table 5 shows the response time of the analytical model under various test conditions compared with the actual machine. According to Table 5, the response time error is within ±10% with the P control gain set to high. On the other hand, it is between about 10% and 30% with the P control gain set to low. This is probably because a lower P control gain leads to a smaller difference between the motor torque and the load torque in the transient state, resulting in a relatively stronger impact of the motor torque error of the model on the rotary action.

Table 5 Evaluation of responsivity of improved electric pump model

Operating conditions	P control gain	Response time error
Low pressure, High speed	Low	12.8%
	Mid	-12.0%
	High	2.0%
High pressure, Low speed	Low	21.6%
Mid pressure, Mid speed	Low	7.1%
	Mid	4.3%
	High	-7.9%

In this way, we successfully built an electric pump model that can reproduce efficiency and response time with the target accuracy by setting the P control gain for pump speed to a high value. Note that the KYB hydraulic system described in Chapter 6 uses a high P control gain setting for performance evaluation.

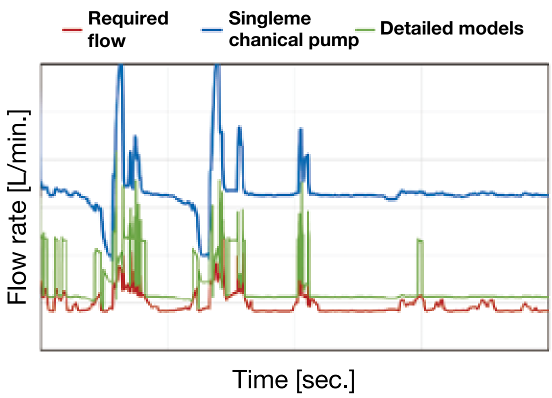
6 Performance Evaluation of Hydraulic System

We combined the models built in the previous chapter (hereinafter "the detailed models") into a whole hydraulic system and evaluated the performance through simulation in order to determine whether the system is effective.

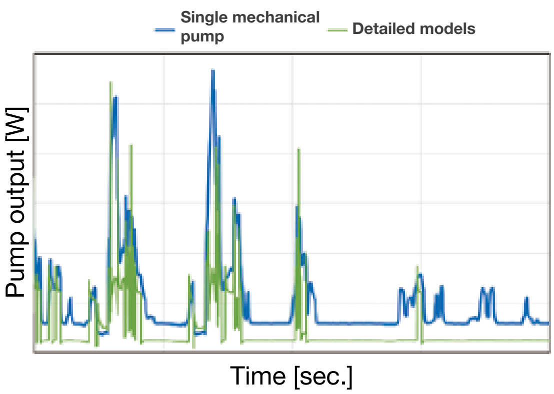
6.1 Performance Evaluation of Hydraulic System

This section describes the results of system performance evaluation of the detailed models which was conducted in the same manner as for Chapter 3. Fig. 17 shows the results of the simulation. Fig. 17(a) shows the pump discharge flow rate, indicating that the pump meets the required flow rate. This graph also reveals that the detailed models discharge at a lower flow rate as a whole than the single mechanical pump system, implying that any excessive flow has been reduced.

The pump output of the detailed models is lower than that of the single mechanical pump system model over the entire range as shown in Fig. 17(b). The output reduction rate is 48.3%, proving the effectiveness of this system.



(a) Flow rate



(b) Pump output

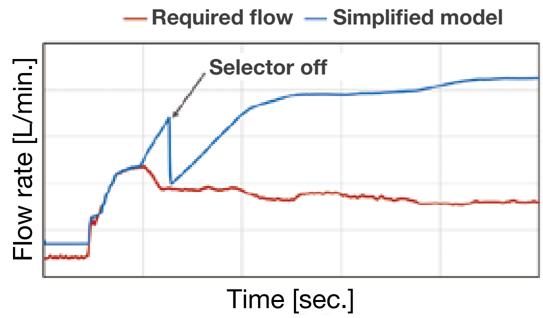
Fig. 17 Results of simulation

6.2 Stability Evaluation of Hydraulic System

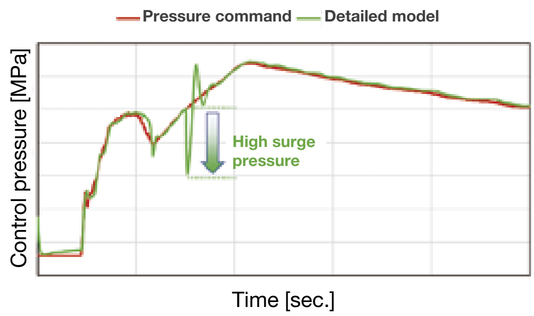
This section describes a simulation of an operation pattern that needs instantaneous high flow rate and control pressure, such as kick down (hereinafter "KD"), to determine the follow-up performance.

Fig. 18 shows the results of analysis of flow rate and control pressure. Fig. 18(a) indicates that the required flow rate is mostly satisfied over the entire operating range. Fig. 18(b) indicates that the control pressure follows the pressure command, except for high surge pressures with the selector off.

As described above, the surge pressure causes a shock to the entire vehicle, leading to lower ride comfort. This issue might be resolved by cooperative control of the selector valve and the electric pump. Then, we took a simplified measure for reducing the surge pressure, which is described in this section.



(a) Flow rate



(b) Pump output

Fig. 18 Follow-up performance during KD

Fig. 19 shows an overview of the cooperative control logic. The KYB hydraulic system is conventionally controlled to merge the main discharge with the sub discharge if the main discharge alone is not sufficient to provide the required flow. If this is not enough to meet the required flow, the electric pump is started up to compensate for the flow. This is a control logic only intended to satisfy the required flow. In this case, a surge pressure is likely to occur when the sub discharge is turned on/off with the selector valve, as shown in the upper section of Fig. 19. Then, we decided to add surge pressure reduction control to the cooperative control logic as shown in the lower section of Fig. 19. This control system is intended to prevent abrupt changes in flow in the whole hydraulic system by decreasing/increasing the discharge of the electric pump in response to an increase/decrease in sub discharge upon change-over of the valve.

Fig. 20 shows the results of the analysis of control pressure in the KD evaluation using the model including the surge pressure reduction control. It was confirmed that the addition of this control can reduce the surge pressure that occurs upon change-over of the valve by about 50%.

In addition to the stability improvement verified in this section, a further study on cooperative control will enable us to reduce the frequency of change-overs, implement a robust operation design, and improve the effect of pump output reduction.

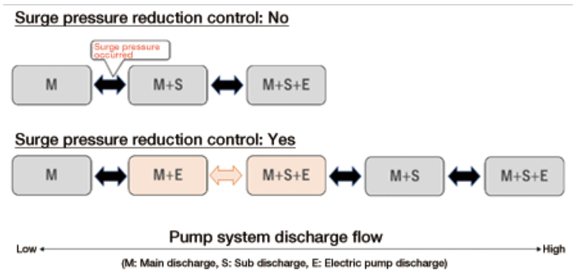


Fig. 19 Cooperative control logic

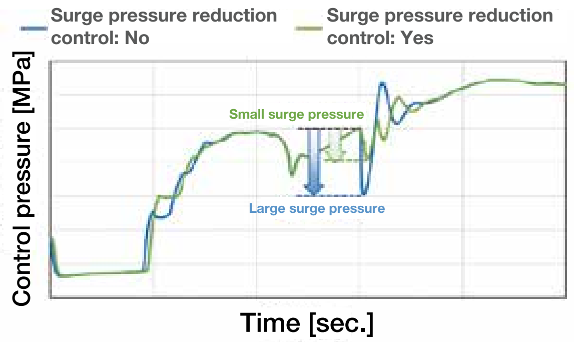


Fig. 20 Effect of surge pressure reduction by cooperative control

7 Concluding Remarks

We have established a simulation technology that enables performance prediction as a means of confirming the feasibility of a high-efficiency hydraulic system and obtained the following findings:

(1) We modeled a KYB hydraulic system consisting of a mechanical pump, an electric pump, and a selector valve to enable prediction of the system performance with high accuracy. We then confirmed the effectiveness of this hydraulic system.

(2) We carried out a parameter study using the model built to identify parameters affecting the stability, clarifying the design indices with which surge pressure and responsivity can be controlled.

(3) We successfully set the prospects for improving the performance of the hydraulic system by coordinating the control of each component according to operation mode.

The results obtained make it possible to accelerate the development with less frequent prototyping of actual machines and design a hydraulic system with priority placed on the main machine.

From now on, we will study optimal equipment specifications and/or cooperative control suited to operation modes required by customer specifications, with the aim of maximizing the system performance/quality. We will also apply the technology to the prediction of the lubrication/cooling system for eAxle (an integrated structure of the motor, reducer, and inverter) that is growing in demand in the automobile industry.

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