Development of Simulator for Evaluation of Steering Systems

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Steering reaction torque is one important type of information for drivers since it has significant influence on vehicle maneuverability. Even with today's advanced simulation technology, however, it is very difficult to accurately simulate steering feeling. The purpose of this study is to develop a steering Hardware-in-the-Loop (HIL) simulator that can quantitatively evaluate steering systems. This simulator can control the force on the tie rod by means of an AC servomotor. Validity of this HIL simulator has been ascertained by comparing the simulation results with those obtained during actual vehicle testing.

Key Words: steering system, evaluation, HIL, reaction torque

1. Introduction

The origin of steering systems is said to be a direction-changing system named the "tiller method" installed on a steam engine car that was manipulated by a steering stick. From that time on, steering evolved along with the progress of technology. In the 1930s, the modern steering system was developed, which was basically the same as the current design, i.e. a system comprised of a steering wheel at the manipulating end, ball screw or rack & pinion gears as the speed reduction mechanism, and a steering rod for connection to the tire. Furthermore, with the spread of power steering systems, increasing demands for comfortable steering feeling and more safety brought about the development of a speed-sensitive hydraulic reaction mechanism, and the assist force is electronically controlled by an electric motor, has been commercialized.

In evaluating steering systems, there still are many aspects such as steering feeling regarding which definite evaluation cannot be made without vehicle tests. If more accurate bench testing is available prior to vehicle tests, however, it will be instrumental for earlier debugging and faster feedback for development of new steering systems. Since evaluation of a steering system involves not only mechanical performance factors such as output force but also very important sensory performance factors such as steering feeling, it is imperative that bench testing includes reproduction of the human-vehicle reactive loop in order to improve its evaluation accuracy. However, the steering system involves factors such as the intermediate shaft and other mechanical elements as well as the dynamic characteristics of electrical circuits, and for these factors computation models generally have not been developed. Steering feeling, on the other hand, is influenced by the various characteristics of the control unit and assist motor in addition to the mechanical elements involved in transmission of force from the road surface to the steering wheel. Due to these factors, it becomes difficult for computer simulations generally used for vehicle motion systems to achieve the evaluation accuracy of vehicle tests.

In that regard, the aim of this study has been to construct a steering evaluation simulator that enables bench testing to satisfy such requirements. This paper presents an outline of this simulator for steering evaluation as well as a comparison between the test results with this simulator and those of vehicle tests.

2. Simulator for Steering Evaluation

During vehicle tests for evaluation of steering feeling, etc., it is presumed that the test driver feels the torque transmitted from the road surface and the vehicle motion in response to his input force on the steering wheel. Therefore, in order to evaluate steering on a test bench, it is necessary to transmit torque and vehicle behavior to the driver like in a vehicle test. However, at present, there is no simulator that can accurately transmit all such torque and vehicle behavior that the driver feels in a vehicle test. That is why various attempts have been made to create bench testing that can replace a portion of vehicle testing by incorporating transmission of a part of such torque and vehicle motion.

Koyo has developed various simulators for the purpose of evaluating steering systems, which are divided into the following two categories: One is the driving simulator shown in Fig. 1, which is mainly intended for evaluation of the driver's human factors, while the other is the actual-steering-mounted simulator (steering simulator) designed for evaluation of steering system characteristics. While the driving simulator is effective for verification of steering control logic, it cannot easily evaluate the steering system. In this project, a hardware-in-the-loop (HIL) simulator has been developed for evaluation of steering systems.
3. Steering HIL Simulator

3.1 Outline of Simulator

The performance of a steering system is influenced by various factors, such as the electrical characteristics of the control logic and motor as well as mechanical characteristics like friction and viscosity, typically represented by the assist characteristics. As a result, steering system evaluation must take into account all comprising elements, including the steering electrical control unit (steering ECU), steering column and steering gear. Therefore, the developed simulator incorporates the steering ECU in addition to such mechanical elements as the steering column and steering gear. As shown in Fig. 2, when the driver manipulates the steering wheel, the input end is the steering wheel and the output end is the steering tie rod. In order to simulate the steering reaction torque in this simulator, a force must be applied to the tie rod (rack force) from which the force must be transmitted to the steering wheel via the steering gear. In this project, therefore, we tried to develop a simulator that can evaluate steering on a system level by combining an accurate reproduction of the force on the tie rod and incorporation of actual mechanical and electrical components of the steering system. To generate the torque that is felt by the driver through the steering wheel during driving, the PC for simulator control incorporates a vehicle model wherein the self-aligning torque is calculated from the side slip angle of the front tires on a real-time basis, while the servomotor is signaled to reproduce the axial force thus computed.

In order to reproduce the actual driving environment in the simulator, simulated vehicle speed information is given to the steering ECU.

In this study, the steering subjected to simulation was a column type EPS, that is, an EPS incorporating an assist motor on the steering column shaft. Also, this report did not include the stationary steering operation, which has a different axial force generation mechanism than that during driving.

3.2 Installed Models

In this study, based on consideration of the steering evaluation simulator, a vehicle model with a low degree of freedom was adopted so that the simulation in the PC would not be too complicated. Specifically, an equivalent two-wheel model taking into account three degrees of freedom, i.e. lateral, yaw and roll directions, as shown in Fig. 3, was used.

The equation of motion around the vehicle center of gravity and the roll axis are shown below, provided that the products of inertia with respect to x and z directions are considered to be 0.

\[
m\dot{V}(\dot{\beta} + \gamma) - m_h \dot{h} \dot{\phi} = 2F_f + 2F_r \tag{1}
\]

\[
I_\gamma \ddot{\gamma} = 2l_i F_f - 2l_r F_r \tag{2}
\]

\[
I_c \ddot{\phi} - m_h h \dot{V}(\dot{\beta} + \gamma) = (-K_t + mgh_v) \dot{\phi} - C_t \dot{\phi} \tag{3}
\]

In this study, the side slip angle of the tires is considered to be very small, and a model characterized by linear tire lateral force with respect to the side slip angle of the tires was used. Taking into account the dynamic characteristics of the tire lateral displacement during the time until the tire lateral force is generated from the side slip angle of the tires, the following equations were employed:

\[
\frac{C_i}{kV} \dot{F}_l + F_l = C_i \beta_l \tag{4}
\]

\[
\frac{C_i}{kV} \dot{F}_r + F_r = C_i \beta_r \tag{5}
\]
Also, the following equations represent respectively the side slip angles of the front and rear tires based on the geometrical relationship.

\[
\beta_f = \frac{1}{V} \gamma - \beta_i - a_i
\]

\[
\beta_r = \frac{1}{V} \gamma - a_i
\]  

(6)  
(7)

Where, \(a_i, a_r\) are respectively the roll steer of the front and rear wheels as expressed in the following equations:

\[
a_i = a_i \phi
\]

\[
a_r = a_r \phi
\]  

(8)  
(9)

Where, \(a_i, a_r\) represent the roll steer magnitudes per the unit roll angle, i.e. coefficients of roll steer. Here, these values are considered to be constants assuming that roll steer magnitude varies linearly with respect to roll angle.

The self-aligning torque felt by the driver as steering reaction torque is generated due to the difference between the center of rotation of the tire in the tire-road surface contact area and the application point of the lateral force generated on the tire. To calculate the force acting on the tie rod, the value obtained by dividing the self-aligning torque by the length of the knuckle arm is the rack force in the model. The self-aligning torque and the rack force are expressed in the following equations, assuming that, in this study, the forces on the right and left tie rods were even and the steering system in the vehicle model installed in the simulator was a rigid body.

\[
T_{sa} = 2\xi F_i
\]

\[
F = \frac{2\xi}{L} F_i
\]  

(10)  
(11)

4. Verification of Vehicle Model by Vehicle Test

An attempt was made to verify that the model described in the previous section could reproduce the rack force to be simulated in the simulator. For this purpose, an actual vehicle incorporating the same steering system as that used in the simulator was employed to measure the rack forces and frequency responses of vehicle behavior against varying steering angular input, with which the outputs of the model were compared for verification.

The vehicle test was conducted on a dry asphalt road by steering a car in a sine-wave pattern of 60 degrees amplitude while driving the vehicle at speeds of 20 km/h, 40 km/h, 60 km/h and 80 km/h, and each vehicle information were measured. The rack force acting on the actual vehicle was measured as the sum of the forces measured by load cells attached to the right and left steering tie rods.

From the measurement data, the gain at each speed as well as the phases against the steering wheel angles were calculated by means of FFT analysis. Those values were plotted for each frequency of steering in the test run for comparison with the values obtained by the models described in the preceding section. Figures 4, 5 and 6 show respectively the frequency responses in the yaw rate, roll angle and rack force.
In order to transmit the target force to the AC servomotor, a control logic that superimposes feed forward control to give the target force with a feedback compensation for friction, etc. in the torque transmission system was employed. For feedback compensation, a PI control was employed. Figure 8 shows the control logic for steering reaction force.

Figure 8: Control system of reaction force generator

Use of this control logic to control the reaction force generator resulted in the change of rack force on the HIL simulator shown in Fig. 9 compared with the target force, and it was confirmed that the target force could be reproduced on the simulator.

Figure 9: Experimental result of force on tie rod in HIL simulator (V=50 km/h)

Furthermore, to verify the rack force response to steering wheel angle, steering wheel angle input having the same sinewave pattern as the vehicle test was given to the HIL simulator. The rack force measured on the HIL simulator was compared with the output from the simulation model corresponding to the target value of rack force along the continuum of steering wheel angles. The simulation was done at the two speeds 40 km/h and 60 km/h. The result of this simulation is shown in Fig. 10.

5. Simulation System for Reaction Force

5. 1 Design of Control Unit

For the purpose of applying a force on the steering tie rod on the simulator, a device combining an AC servomotor and a ball screw as shown in Fig. 7 was made because of its easy control and high efficiency as a reaction force generator. The reaction force simulation control unit could give torque signals to the AC servomotor in accordance with the target force calculated for a particular vehicle model from the steering wheel angle.

Figure 7: Appearance of reaction force actuator
6. Conclusion

The purpose of this project was to develop a simulator for efficient evaluation of steering systems. As this simulator is intended to evaluate the steering system itself, those mechanical and electrical elements considered to affect steering performance were installed in the simulator.

In order to obtain simulation results closer to vehicle test results, the simulator integrated simulation of steering wheel reaction force, for which the rack force acting on the steering tie rod was reproduced. In this study we incorporated tire lateral force dynamic characteristics and roll moment into the equivalent two-wheel model, and in regard to the steering frequency range of 1 Hz at vehicle speeds of 40~60 km/h, we constructed a model capable of reproducing rack force characteristics in response to steering input by the driver. This simulation model was combined with an actuator using an AC servomotor to construct an HIL simulator that can reproduce the rack force comparable to that on an actual vehicle. A series of comparative simulator and vehicle tests gave results verifying reproducibility of steering torque including the influence of the mechanical and the control elements of the steering system.
In the future, Koyo will make use of the features of this simulator for evaluation and analysis of steering feeling while resolving the problems revealed in this study.

Explanation of symbols:

- \( V \) Vehicle speed
- \( \beta \) Side skid angle at center of gravity of vehicle
- \( \gamma \) Yaw rate
- \( \phi \) Roll angle
- \( m \) Gross vehicle mass
- \( m_s \) Sprung mass
- \( I\) Yaw inertia moment
- \( I_r \) Roll inertia moment
- \( C_u \) Roll equivalent viscosity coefficient
- \( K_u \) Roll stiffness
- \( l_f \) Distance from center of gravity to front wheel axis
- \( l_r \) Distance from center of gravity to rear wheel axis
- \( h_s \) Distance from roll center to center of gravity
- \( F_f \) Lateral force on front wheel
- \( F_r \) Lateral force on rear wheel
- \( g \) Gravitational acceleration
- \( C_r \) Cornering stiffness of front wheel (for one wheel)
- \( C_r \) Cornering stiffness of rear wheel (for one wheel)
- \( k \) Lateral stiffness of tire
- \( \beta_f \) Angle of side skid of front wheel
- \( \beta_r \) Angle of side skid of rear wheel
- \( \delta \) Effective steered angle of front wheel
- \( \alpha_f \) Roll steer of front wheel
- \( \alpha_r \) Roll steer of rear wheel
- \( T_{sa} \) Self-aligning torque
- \( F \) Rack force
- \( \xi \) Trail (\( \xi = \xi_c + \xi_n \))
- \( \xi_c \) Caster trail
- \( \xi_n \) Pneumatic trail

References