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Experimental validation of steering torque feedback simulator through vehicle running test †

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Abstract

This paper describes a force feedback system based on real-time multibody dynamic analysis. This system can provide the analyzed reactive force to the operator through the operational device. In this study, this system is used as a steering torque feedback simulator of an automobile. This simulator can provide the haptic sensation of the steering wheel to the operator. For the purpose of evaluating the validity of the developed simulator, we conducted some vehicle running tests with an experimental electric vehicle. The results of these tests were compared with the results simulated on the steering torque feedback simulator. It was shown that the developed simulator can provide a suitable steering torque to the operator.

Keywords: Steering maneuverability; Steering torque; Real-time analysis; Multibody dynamics

1. Introduction

Haptic sensation is important information for the operator of a mechanical system such as an automobile. The operator can recognize the condition of the mechanical system by the reactive force from the operational device, and it influences the judgment of next operation. The characteristic of reactive force depends on the stiffness property of mechanical components, layout of mechanical links, and so on. Multibody analysis is an effective tool for the accurate analysis of this reactive force characteristic when developing a mechanical system operated by a human.

We developed a force feedback system based on real-time multibody dynamic analysis [1]. This system realizes an intuitive evaluation of operability of mechanical systems before prototyping. In this study, this system is used as a steering torque feedback simulator. The haptic sensation of a steering wheel of an automobile is provided to the operator with this simulator. In this system, the operation of the steering wheel is measured with an rotary encoder and is used as the input data for multibody dynamic analysis. The steering torque is calculated with the inverse dynamic analysis, and the target torque value is given to the DC servomotor. Finally, the reactive torque is provided to the operator. In this process, the real-time analysis of multibody system is indispensable. For this purpose, the authors had developed a hand-coded multibody analysis program running on the RTLinux/Free [2] environment.

To evaluate the validity of the developed steering torque feedback simulator, we did some running tests with an experimental electric vehicle developed in our laboratory. The comparison of running test results with developed steering torque simulator is discussed in this paper.

2. Steering torque feedback system

2.1 Force generator

The steering torque feedback simulator used in this

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study is shown in Fig. 1. A brushless DC servomotor equipped with a rotary encoder is mounted on the steering shaft as a force generator. The maneuvering of an operator is measured by the rotary encoder, and is used as the input data of the multibody dynamic analysis. The inverse dynamic analysis is executed to obtain the reactive torque on the steering wheel. Finally, the DC servomotor is controlled to achieve the desired torque. The specification of the force generator is shown in Table 1.

Real-time analysis is required to calculate the desired steering torque within a constant time period. In our simulator, the calculation task is managed by RTLinux/Free. RTLinux/Free is one of the real-time OS's, and its kernel is designed to act as a plug-in into the regular Linux operating system kernel. Therefore, a real-time process is assured at the kernel level. The effectiveness of the RTLinux is discussed in the following section.

2.2 Assurance of real-time process

The computational speed is an important factor for real-time analysis. From the aspect of the performance of matrix operations, the matrix libraries such as LINPACK [3], LAPACK [4], and CPPLAPACK [5] are effective for speed up of matrix calculations. However, it is difficult to run these libraries on an RTLinux environment because the available commands are restricted on RTLinux. Therefore, we cannot use these matrix libraries for the multibody analysis in the developed simulator. The commonly-used LU-factorization algorithm is used to solve the linear equation in the developed system, whose calculation performance is less powerful in comparison with those matrix libraries mentioned above. The computational time of the LU-factorization algorithm on RTLinux was compared with that of CPPLAPACK on the Linux environment by solving the following linear equation:

$$\begin{bmatrix} a_{11} & \cdots & a_{1n} \\ \vdots & \ddots & \vdots \\ a_{n1} & \cdots & a_{nn} \end{bmatrix} \begin{bmatrix} x_1 \\ \vdots \\ x_n \end{bmatrix} = \begin{bmatrix} b_1 \\ \vdots \\ b_n \end{bmatrix} \tag{1}$$

Matrix size n was set to 200 in this evaluation. These calculations were executed on the same PC. The specifications of each environment are shown in Table 2. The calculation was executed for 50000 steps, and the computational time was recorded in each step.

The result is shown in Table 3 and Fig. 2. Fig. 2 shows the distribution of the computational time in each environment. Although the average value of the computational time on Linux with CPPLAPACK is smaller, the computational times are widely distributed. In contrast, the variation of the computational times on RTLinux is very small. The stability of the computational time is the most significant factor for a real-time system like ours, and this result shows that the use of RTLinux is suitable for the developed simulator.

Table 1. Specification of force generator.

Parameter	Value
Rated output	250 W
Stall torque	3.25 Nm
Reduction gear ratio	21 : 1
Encoder resolution	500 counts/rev

Fig. 1. Steering torque feedback simulator.

Table 2. Computational environment.

	RTLinux	Linux
CPU	Intel Pentium4 2.8 GHz	
OS version	RTLinux 3.1	Vine Linux 2.6
Program code	C	C++

Table 3. Computational time on each environment.

	RTLinux	Linux
Average [ms]	9.204	7.246
Maximum [ms]	9.994	129.1
Minimum [ms]	9.169	7.172

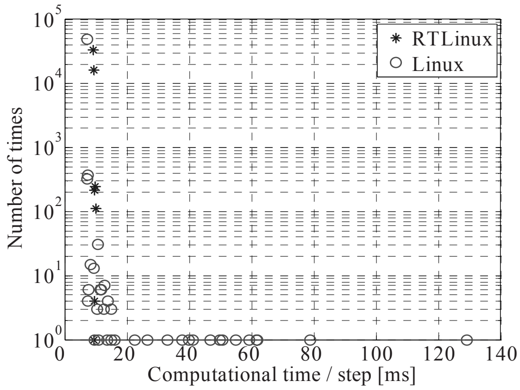


Fig. 2. The distribution of computational times.

3. Multibody vehicle model

The experimental vehicle shown in Fig. 3 was a target of the multibody vehicle model applied in the developed steering torque feedback simulator. The specification of this vehicle is shown in Table 4. This vehicle contains many mechanical linkages. In multibody analysis, these mechanical linkages are described as the constraint equations. In addition, the springs and dampers mounted on the vehicle can be considered as the external force elements. The equation of motion of this vehicle can be described in differential algebraic equation [6]:

$$\begin{bmatrix} \mathbf{M} & \mathbf{\Phi}^T \\ \mathbf{\Phi} & \mathbf{0} \end{bmatrix} \begin{bmatrix} \ddot{\mathbf{q}} \\ \boldsymbol{\lambda} \end{bmatrix} = \begin{bmatrix} \mathbf{Q} \\ \boldsymbol{\gamma} \end{bmatrix} \tag{2}$$

where,

$\mathbf{M}$: Mass matrix
$\mathbf{\Phi}$: Jacobian of constraints
$\mathbf{q}$: Generalized coordinates
$\boldsymbol{\lambda}$: Lagrange multipliers
$\mathbf{Q}$: Generalized forces
$\boldsymbol{\gamma}$: Acceleration equations of constraints

Based on the Lagrange multiplier calculated from this equation, the reactive force **F** can be calculated by the inverse dynamic analysis as a constraint force.

$$\mathbf{F} = \mathbf{\Phi}_{\mathbf{q}}^{\mathbf{T}} \boldsymbol{\lambda} \tag{3}$$

In the multibody dynamics analysis, the number of body components greatly influences the computational time. To reduce the computational time, we introduce the concept of "massless link model [7]," in which the suspension arms are treated as constraint equations and the effect of the mass of suspension arms is neglected. In our vehicle model, the upper arm of the front suspension was described as two distance constraints, and the tie-rod arm was treated as a distance constraint (Fig. 4). The performance of this "massless link model" was compared with the "rigid-body link model," in which the suspension arms are described as independent body elements. The numbers of bodies and constraint equations in each body are compared in Table 5.

Fig. 5 shows the distribution of the computational times with both models. The multibody analysis was executed for 1000 steps. The average of computational times with the rigid-body link model was 6.30 milliseconds, and that of the massless link model was 1.78 milliseconds. The analysis results of yaw rate and steering reactive torque with sinusoidal steering input are compared in Fig. 6, and it can be observed that the results are similar. These results indicate that the use of a massless link model is reasonable for our purpose and effective to enhance the computational performance.

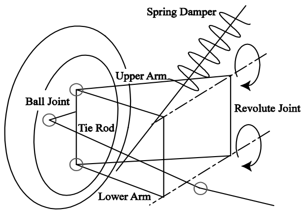
Table 4. Specifications of experimental electric vehicle.

Parameter	Value
Weight distribution	41.7 : 58.2
Moment inertia	123.6 kgm^2
Total weight	240 kg
Overall length	2205 mm
Wheel base	1865 mm
Wheel track	1165 mm

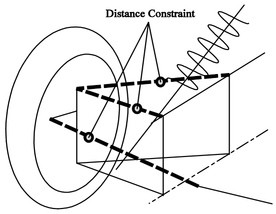
Fig. 3. The experimental electric vehicle.

Table 5. Specification of model.

	Rigid-body link model	Massless link model
Number of bodies	11	7
Number of constraints	58	36
DAE matrix size	146	92



(a) Rigid-body link model



(b) Massless link model

Fig. 4. Comparison of vehicle models.

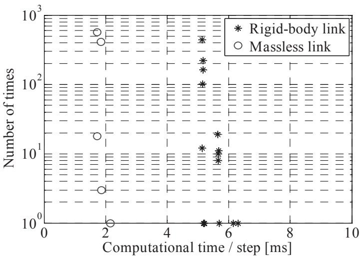
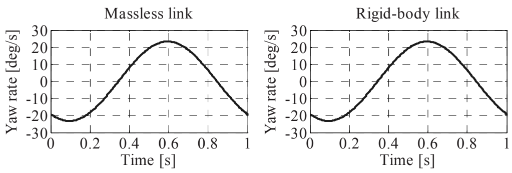
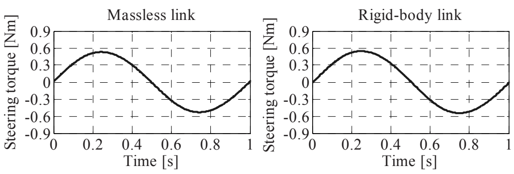


Fig. 5. Difference of computational times for each model.



(a) Yaw rate of the vehicle



(b) Steering reactive torque

Fig. 6. Analysis result with sinusoidal steering input.

4. Evaluation of steering torque simulator

4.1 Static test

For a driver of an automobile, the steering torque is one of the most important pieces of information as well as the visibility. Therefore, providing the suitable steering torque to the driver is very important when developing a driving simulator for the research of the human-vehicle system. We conducted a static test with the experimental electric vehicle to evaluate the validity of the developed steering torque simulator. In this test, the front tires of the experimental vehicle were mounted on the radius gauges in order to moderate the influences of the lateral force and the friction of the tire, as is shown in Fig. 7. The static test enabled us to investigate the fundamental characteristics of the steering torque. In this test, the steering wheel was slowly operated in the range from –360 to 360 degrees. The steering torque, which the operator felt, was measured by the torque transducer equipped between the steering wheel and the steering shaft. As the next step, the same operation was given to the developed steering torque feedback simulator. The step size of the numerical integration was set to 3 milliseconds.

Fig. 8 shows the measurement results of the steering torque with the actual experimental vehicle and developed simulator. The effects of the mechanical friction can be observed in both lissajous curves, and the generated steering torque had good agreement with the actual vehicle. In Fig. 8(b), the calculated torque by real-time multibody analysis is also indicated. Because of the mechanical friction of the developed simulator, the measurement result of steering torque was different from the calculation result. These results indicate that the mechanical friction level of the developed simulator under slow steering operation was similar to the actual vehicle.

Fig. 7. Static test with experimental vehicle.

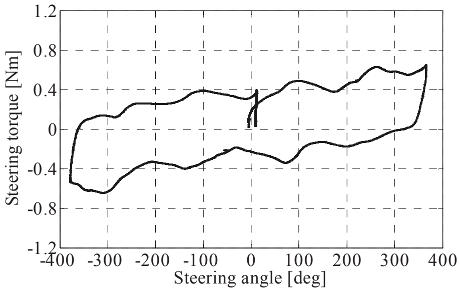
4.2 Running test

We tried to realize the steering torque under the running condition with the developed simulator as the next step. A slalom running test was carried out with the experimental vehicle, and the same situation was emulated with the developed simulator. In this test, the steering wheel was operated in the range from –100 to 100 degree. The characteristics of the tire force should be considered when simulating the running condition, for the tire force affects the steering torque. Although it is well known that the tire force characteristics are very complex and have strong nonlinearity [8], the slip angle of the tires in this test was rather small; hence the lateral force of the tire was simply described by using a linear equation with the cornering power in the developed simulator as follows:

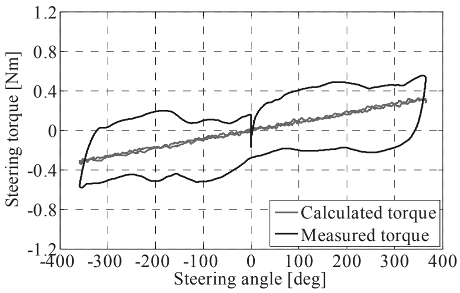
$$F_{lat} = K\alpha \tag{4}$$

where α is the slip angle of the tire. The cornering power K was estimated by some steady state circular tests. The front cornering power K_f was set to 16000 N/rad and the rear cornering power K_r was set to 22000 N/rad in the simulator.

The measurement results of steering angle, yaw rate and steering torque are shown in Fig. 9. The yaw rate calculated in the simulator corresponded with the result in the actual vehicle test, and it indicates that the multibody dynamic analysis can simulate the behavior of the vehicle adequately. It was observed that although the tendencies of the steering torque were similar, the generated torque in the developed simulator was bigger than in the actual vehicle. The results of the steering angle show that the operational speeds

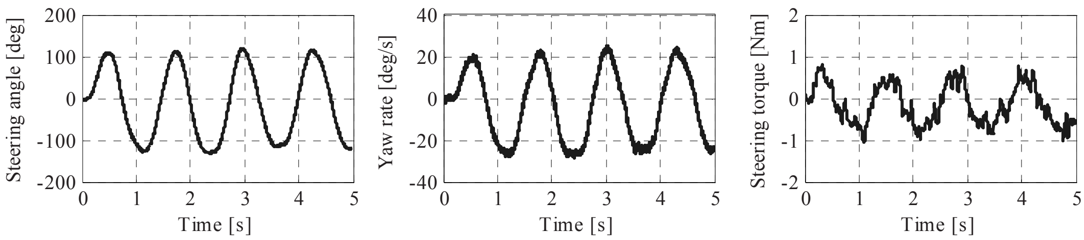


(a) Experimental electric vehicle

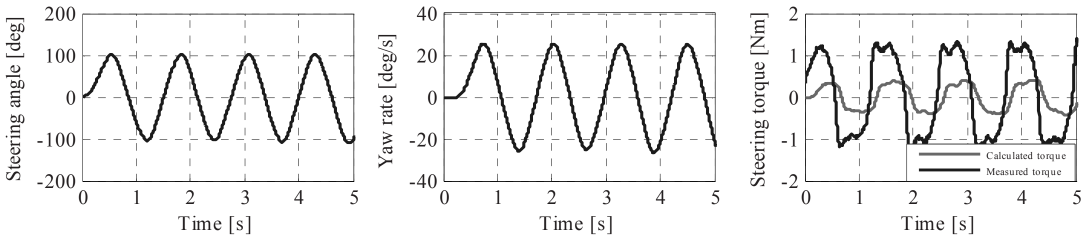


(b) Developed steering torque feedback simulator

Fig. 8. Comparison of steering reactive torque.



(a) Running test results with experimental vehicle



(b) Experimental results with developed steering torque simulator

Fig. 9. Evaluation of the steering torque through slalom running test.

measured in these tests were quicker than the ones in the static test. This quick operation induces the difference of the mechanical friction in both systems, and is thought to bring about the difference in the steering torque results. In addition, the results are considered to be influenced by the difference of the moment of inertia in both steering systems. The enhancement of the reality of the steering torque would be expected by changing the motor and gear system in the developed steering torque feedback system.

5. Conclusion

A torque feedback simulator based on real-time multibody simulation has been developed. The computational speed of the analysis and the assurance of real-time process were evaluated by executing the matrix operation, and the effectiveness of RTLinux was verified. To evaluate the steering torque produce by the developed simulator, the vehicle model of an experimental electric vehicle was used in the multibody simulation. Real-time analysis with the step size of 3 milliseconds was achieved in the simulation by introducing the concept of the massless link model. We performed the evaluation tests in order to validate the developed simulator. In the static test, it was shown that the developed simulator can provide the torque generated on the steering wheel to the operator under the condition that the tire characteristics such as a lateral force and a friction do not affect the steering torque. The results of the running test indicated that the multibody dynamic analysis in the simulator can evaluate the vehicle behavior suitably. A difference of the steering torque was observed, which appeared to be caused by the mechanical friction and the moment of the developed steering torque simulator. By improving the motor and gear system, an enhancement of the reality of the steering torque would be expected.

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