

# Dynamic Behavior Analysis of Hydraulic Power Steering Systems

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Research regarding dynamic modeling of hydraulic power steering systems was conducted. Unexpected phenomena, which worsen driving feeling in hydraulic power steering systems, were analyzed. By using the proposed modeling for power steering, these unexpected phenomena were logically simulated, and optimum parameters for steering designs could be obtained.

## 1. Introduction

Power steering enables the driver to comfortably steer the vehicle with minimal effort.

In countries such as Japan where almost all vehicles are equipped with power steering systems, drivers will not be satisfied by merely enhancing steering power. In recent years, there have been demands for improved performance, such as for quieter, smoother steering feeling. However, while many technical improvements have been achieved through experience with prototypes, analytical approaches to resolving performance problems have been insufficient.

This paper concerns hydraulic power steering, the type of system most widely used at the present time, and presents the results of dynamic behavior analysis of noise and kick-back phenomena.

## 2. Hydraulic Steering System Analysis

### 2.1 Creation of a Theoretical Model

As many mechanical elements of the power steering system as possible were taken into account in formulating the theoretical model. The theoretical model for the power steering system used in the analysis is shown in Fig. 1. The power steering system is a rack and pinion type, with the steering wheel and rack shaft being mechanically linked. The control valve operates via a torsion bar.

Power assist is supplied when the steering wheel is turned (or when external load is applied on the tires).

Here inertia is defined as delay when the components of the power steering system are accelerated, and the proportional constant as the pressure characteristics of the valve produced by twisting of the torsion bar. In particular, pressure lag due to hose expansion when pressure is high was replaced by an equivalent spring/viscous element, and external load on the tires (contact force with the road) by an equivalent friction/spring element. Dynamic behavior analysis of the power steering system was conducted using this model.

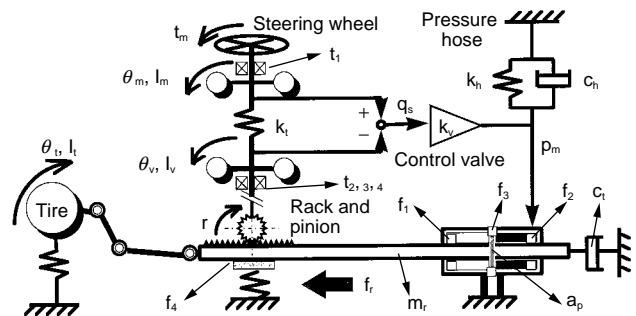


Fig. 1 Steering system model

- $t_m$  : Steering torque
- $I_m$  : Moment of inertia of steering wheel
- $I_v$  : Moment of inertia of pinion shaft
- $I_t$  : Moment of tire inertia of kingpin shaft
- $\theta_m$  : Steering angle
- $\theta_v$  : Pinion angle
- $\theta_t$  : Tire angle
- $k_t$  : Spring constant of torsion bar
- $k_h$  : Spring constant of hose
- $a_p$  : Pressured area of piston
- $P_m$  : Discharge oil pressure
- $q_s$  : Pump discharge oil flow
- $k_v$  : Control valve gain
- $m_r$  : Mass of rack shaft
- $r$  : Pitch circle radius of pinion gear
- $c_t$  : Viscous constant with rate of rack shaft displacement
- $c_h$  : Hose expansion viscous constant
- $f_r$  : Rack shaft output
- $t_1$  : Oil seal friction torque (steering wheel side)
- $t_2$  : Control valve oil seal friction torque (high-pressure side)
- $t_3$  : Control valve oil seal friction torque (low-pressure side)
- $t_4$  : Oil seal friction torque (pinion side)
- $f_1$  : Cylinder seal frictional force (low-pressure side)
- $f_2$  : Cylinder seal frictional force (high-pressure side)
- $f_3$  : Piston seal frictional force
- $f_4$  : Friction component frictional force

**2. 2 Force Balance in Theoretical Model**

We determined the force equilibrium equations for each part of the power steering system from the simplified servo model power steering operation described in Fig. 1.

- ① Force equilibrium equation between steering wheel and torsion bar

When the driver applies steering torque, response torque is produced from accelerated motion of the steering wheel, which has a moment of inertia, torsional torque is produced from the phase difference of the twisted torsion bar, and frictional torque is produced from the oil seal. Thus the following equation is established:

$$t_m = I_m \ddot{\theta}_m + t_1 + k_t (\theta_m - \theta_v) \tag{1}$$

- ② Force equilibrium equation between torsion bar and rack shaft

The piston receiving hydraulic drive force from the pump due to the steering of the driver moves the rack shaft horizontally by one degree of freedom. At this time, the torsional torque of the torsion bar affects operation of the rack shaft in the same way as the manual force of the driver. On the other hand, the inertia due to the turning of the pinion shaft, rack shaft weight, and oil seal friction act as resistance to rack shaft operation. Thus, the following equation is given:

$$f_r = a_p p_m - (f_1 + f_2 + f_3 + f_4) - m_r \ddot{\theta}_v - c_r \dot{\theta}_v + \{k_t (\theta_m - \theta_v) - (t_2 + t_3 + t_4) - I_v \ddot{\theta}_v\} / r \tag{2}$$

- ③ Pressure transmission equation between the control valve and hose

When the control valve is opened or closed by the driver's steering torque, statically, valve pressure is produced according to the steering torque. With an actual system, however, hose volume varies according to the pump pressure ripple, and pressure lag of the control valve can easily be anticipated. Assuming that hydraulic oil pressurized by the pump is consumed by the expansion of the hose when the hose expands, control valve pressure  $p_m$  is defined by a function having the following variables:

$$p_m = P (q_s, k_v, k_h, c_h, \theta_m, \theta_v) \tag{3}$$

- ④ Required rack shaft driving force due to friction between tires and the road

It has been reported<sup>1)</sup> that the kingpin drive torque needed for steering angle at the time of steering during parking is calculated from the caster angle of the kingpin and the external load distributed on the tires, assuming that the contact shape of tires and the road is not distorted according to steering angle.

With an actual system, however, the contact shape of the tires and the road is probably distorted according to the steering angle. We determined the modification factor by experimentation and treated torque needed for driving the rack shaft due to friction between the tires and road as a function having the following variables:

$$f_r = F (\tau, R, N, f_s, n, r, \theta_v, \theta_r, I_r) \tag{4}$$

- $\tau$  : Caster angle
- $R$  : Tire radius
- $N$  : External load on front tire
- $f_s$  : Friction coefficient between tires and the road
- $n$  : Steering wheel/tire steering angle ratio

Phenomena peculiar to power steering were analyzed using the above equations of equilibrium (1) ~ (4).

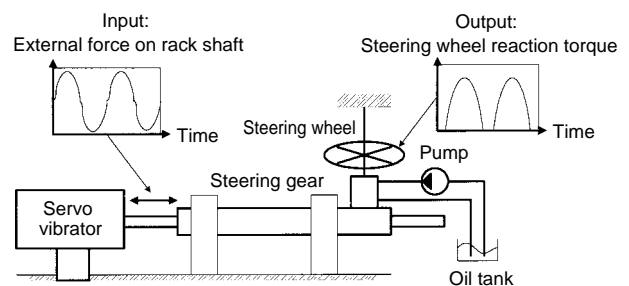
**3. Application to Kick-Back Phenomenon**

**3. 1 Kick-Back Phenomenon**

Steering receives external vibration from the road when the vehicle is traveling. The driver recognizes this as external vibration by steering wheel reaction force torque at this time. This reaction force torque is defined as "kick-back." If the amount of kick-back produced by external vibration is abnormally large, the driving feeling is unpleasant. In this report, kick-back phenomenon is discussed by comparison of experimental results with simulation.

**3. 2 Kick-Back Phenomenon Test on the Bench**

Using a power steering system evaluated as having an extremely large amount of kick-back during driving tests as a sample, we performed testing by the method shown in Fig. 2 and Table 1.



**Fig. 2** Bench test for kick-back

**Table 1** Test conditions

Input force wave form	Sine wave
Input frequency	1 ~ 20 Hz
Max. input force	4 000 N
Oil temperature	80°C
Discharge flow rate of pump	167 × 10 <sup>-6</sup> m <sup>3</sup> /sec
Steering angle	0 deg (fixed)
Steering number	3 ℓ class

### 3.3 Comparison of Test and Simulation Results

A comparison of the measurement results obtained from the bench test shown in Fig. 2 and calculation results is shown in Fig. 3. It was found that these results agree fairly well. Particularly concerning control valve pressure, the characteristic cyclical occurrence of small pressure peaks and large pressure peaks can be periodically observed.

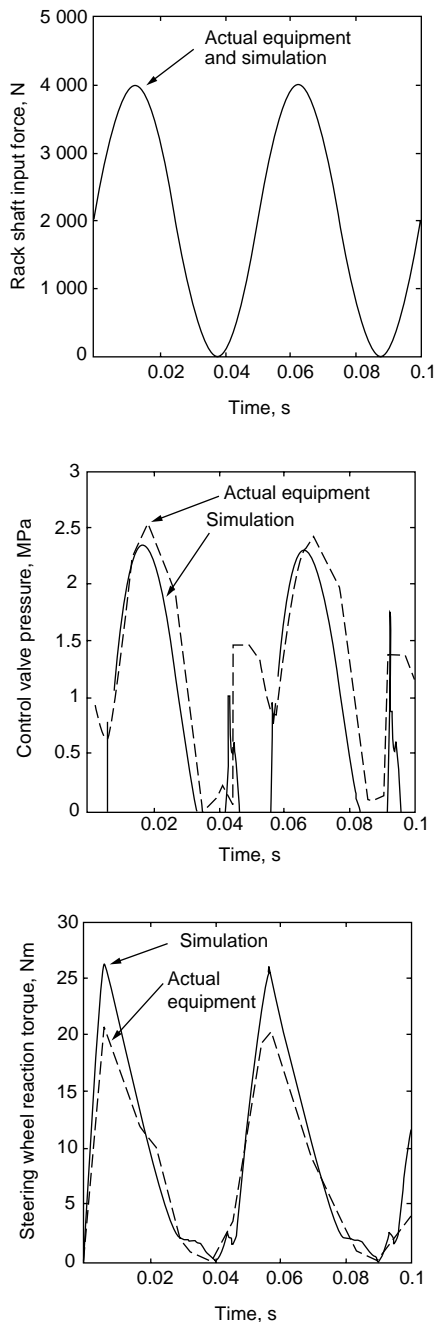


Fig. 3 Comparison of experiment and simulation results

### 3.4 Consideration of Kick-Back

The calculation results of system frequency response characteristics with design parameters of prototype samples for which kick-back occurred and those for which it did not occur are shown in Fig. 4. The figure shows gain and phase of

torque needed to keep the steering wheel angle when external vibration is applied to the rack.

In the power steering type in which kick-back occurred, when external noise frequency is low (approx. 20 Hz), the gain of response torque has already risen, and it can be seen that it is a servo system that is extremely sensitive to external vibration. To improve the kick-back phenomenon, we must design a system which has stable characteristics against the noise frequency band in which kick-back is a problem.

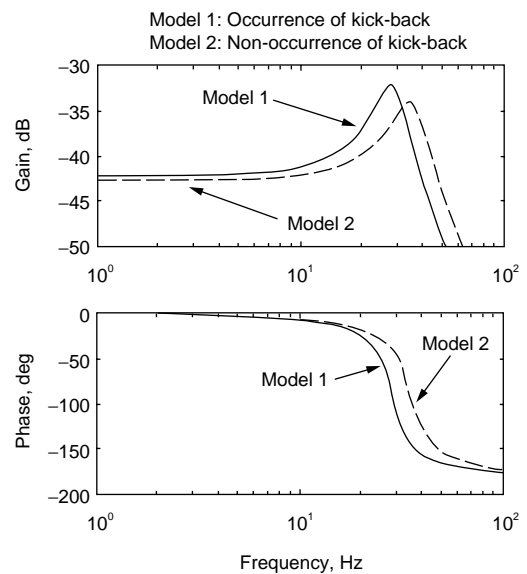


Fig. 4 Frequency response of kick-back

## 4. Application to Noise

### 4.1 Steering Noise during Parking

Power steering greatly improves driving feeling, but it sometimes causes the driver to feel unpleasant vibrations or noise.

We analyzed one abnormal vibration phenomenon seen when steering with the vehicle in a parked state. As the steering wheel is turned, a comparatively loud "goo" noise (hereinafter referred to as "goo noise") is created as a typical abnormal vibration noise occurring during parked turning. The primary cause has already been determined by experimentation, but this time we were able to recognize it by simulation.

### 4.2 Goo Noise Occurrence Based on Experience

Even if goo noise occurs in a power steering system mounted in a vehicle, the goo noise may not always be able to be heard during a bench test. This was one reason the analytical approach was difficult. The results<sup>2)</sup> of investigating goo noise occurrence by the driver's sense of hearing with the power steering system mounted in a vehicle are shown in Fig. 5. There is clearly a correlation between goo noise and design data of the power steering system. As for goo noise, there is an intimate correlation between control valve gain and the hose pulsation damping rate. It has been found that goo noise does not occur on most vehicles if the control valve gain

does not exceed the value indicated by the arrow. We can therefore easily surmise that goo noise may be caused by the hunting effect in the hydraulic servo system.

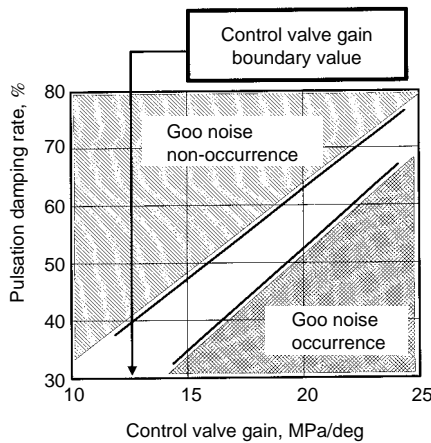


Fig. 5 Boundary value of control valve gain for goo noise

### 4.3 Simulation Results

As was previously stated, goo noise does not always occur on a bench, so load values equivalent to vehicle load conditions (distribution of vehicle weight on tires, friction between tires and road, suspension, etc.) at the time of parked steering was calculated and simulation carried out. We also calculated control valve pressure variation, which is the most likely cause of goo noise, by simulation. Simulation results are shown in Fig. 6.

Fig. 6 a) shows the results of simulating control valve pressure during parked steering based on design data of power steering systems which did not produce goo noise. Pressure instantaneously reaches target pressure (8 MPa), and control valve pressure is stable. On the other hand, Fig. 6 b) shows the results of similar simulation based on design data for power steering systems which produced goo noise. We were able to calculate the appearance of control valve pressure hunting. Pressure hunting of this sort vibrates the pressure hose and creates noise. The driver recognizes noise thus produced as goo noise.

### 4.4 Consideration of Goo Noise

We tried predicting the correlation of various types of power steering design data and goo noise occurrence by simulation. The recommended design data values that can avoid the occurrence of goo noise were clarified by comparison with an actual steering system.

Fig. 7 shows the results of mapping goo noise occurrence and design data of power steering systems contributing largely to goo noise occurrence as determined by simulation. Steering wheel moment of inertia, control valve gain and hose elastic constant were used as design parameters selected on x, y and z axes in Fig. 7. Simulation showed that goo noise would not be produced as long as control valve gain did not exceed the plane values indicated by the arrow in Fig. 7 for design data of all sorts of power steering systems. The results of this simulation are shown in Fig. 5 and agree with the numerical

values of goo noise avoidance learned through experience. Therefore, the reliability of the simulation model has been confirmed.

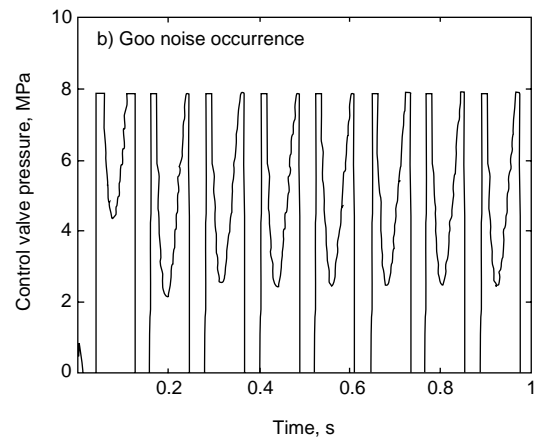
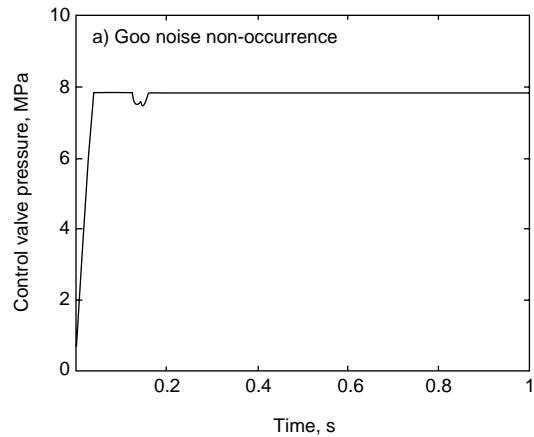


Fig. 6 Simulation results of control valve pressure

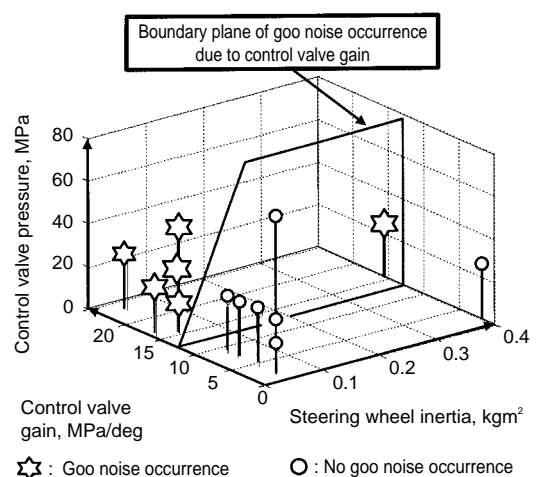


Fig. 7 Occurrence distribution of goo noise for steering design parameters

## 5. Conclusion

The phenomena (kick-back, goo noise during parked steering) particular to power steering systems that give the driver an unpleasant steering feeling were analyzed. Using the power steering system analysis model in this report, a simulation system that could reproduce these phenomena theoretically or propose design parameters to avoid them has been built.

This report deals only with phenomena that cause the driver to experience an unpleasant feeling. In the future, on the other hand, there is a need to convert the pleasant features experienced by the driver into numerical data and build a simulation system that can study the analysis of those power steering systems.

## References

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- 2) M. Kobayashi : Giya-gu on kaiseki the 2<sup>nd</sup> report, TD-3558-02X (1992).