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# Model recognition and validation for an off-road vehicle electrohydraulic steering controller

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**Abstract**

Automated steering control is essential for an autonomous off-road vehicle. Many off-road vehicles use an electrohydraulic (E/H) actuator to implement the steering control. This paper reports the design and validation of an electrohydraulic steering controller through a combination of system identification, model simulation, and field tests. A kinematic model of the steering linkage geometry provided the gain between the hydraulic actuator and the front wheels. The system model was used to close the steering control loop based on the feedback signal from the hydraulic steering actuator rather than from the front wheels. Test results were used to identify the non-linear and dynamic characteristics of the original electrohydraulic steering system. The system identification model was used to develop a preliminary controller, which was simulated under Matlab before beginning full-scale vehicle testing. The simulation and vehicle test results indicated that the steering controller developed was capable of efficiently handling the non-linearity and dynamic asymmetry of the electrohydraulic steering system. © 2002 Elsevier Science Ltd. All rights reserved.

*Keywords:* Off-road vehicle; Steering control; Electrohydraulics; Dynamic system model; System identification; Simulation validation

## 1. Introduction

Automated off-road vehicles offer the promise of higher efficiency, better and more consistent performance, and reduced labor costs. Advances in application technology of off-road vehicles, such as precision farming, reinforce the need for automated off-road vehicles. Automated steering control is one of the fundamental

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**Nomenclature**

| | |
|---|---|
| $\varphi$ | angle of tractor kingpin arm, deg. The angle is determined from $\varphi_1$ and $\varphi_2$ |
| $\theta$ | structural angle between steering linkage bars, deg |
| $b, d$ | steering arm length, m |
| $c$ | steering tie rod length, m |
| $f$ | steering system linkage length, m |
| $a, e$ | length of imaginary bars in the steering linkage, m |
| $y$ | displacement of the steering actuator, m |
| $h(\beta)$ | length of an imaginary bar in the steering linkage, m |
| $\beta(y)$ | angle in the steering linkage determined by cylinder displacement, deg |
| $I$ | inertia on spindle axle, kg m$^2$ |
| $\eta$ | viscous damping of the front wheel corresponding to the steering axle, kg m$^2$/s |
| $T_s$ | aligning torque of the wheel, kg m |
| $T_f$ | Coulomb friction between the ground and wheel, kg m |
| $T$ | steering torque on the front wheel, kg m |
| $P_1, P_2$ | pressures in two sides of the cylinder actuator, Pa |
| $A_1, A_2$ | area of the head-end and rod-end piston of the steering actuator, m$^2$ |
| $\kappa$ | coefficient related to the linkage size, dimensionless |
| $\alpha_1, \alpha_2$ | angles determined by the linkage triangle, deg |
| $K_{NL}$ | non-linear gain factor |
| $G(s)$ | transfer function |
| $\tau$ | time constant of a dynamic system |
| $s$ | Laplace operator |
| $\omega_c$ | cylinder natural frequency |
| $\omega_s$ | electrohydraulic valve natural frequency |
| $\omega_{pw}$ | cylinder chamber pressure wave frequency |
| $\xi_c$ | cylinder damping ratio |

functions for automated off-road vehicles. In addition, automated steering control will improve human and equipment safety, and release the operator from driving to concentrate on implementing other functions.

Steering controller design for off-road vehicles differs from that for on-highway vehicles due to the differences in operation conditions. Off-road vehicles often operate on unprepared, changing, and unpredictable terrain, ranging from paved highways during transportation to spongy topsoil in field operations. Therefore, the steering controller for off-road vehicles should provide an appropriate steering response and account for variations in operating state, travel speed, tire stiffness, ground condition, and other vehicle dynamics parameters.

Designing an electrohydraulic (E/H) steering controller, Laine [1] analyzed the control characteristics of a parasitic steering valve, and found that the performance of a steering controller depended on both the characteristics of the steering control valve and the steering dynamics of the vehicle. Erbach et al. [2] pointed out that non-negligible and inconstant friction could produce significant and unpredictable side-slip, which would significantly affect steering performance of a vehicle. In an early analysis on steering dynamics of off-road vehicles, Grovum and Zoerb [3] developed a dynamic model suitable for simulation on an analog computer. Julian [4] developed a second-order empirical model for the turning (yaw) rate of an off-road vehicle, which could reasonably predict the steering behaviors of the vehicle under various controlled circumstances. O'Connor et al. [5] designed an automatic steering controller for an off-road vehicle based on a set of linearized motion equations. This controller ignored the effect of mass and inertia, and limited the vehicle motion to slow forward speeds. To improve the steering controller design technology, Lee [6] used a "model-following" method to validate both steady-state and transient lateral response of a variable dynamic testbed vehicle. Krishnaswami and Riozzoni [7] have reported a practical method of estimating vehicle steering dynamics for a steer-by-wire system.

Previous research indicates that off-road vehicles require a fast, but stable, steering control. The high degree of non-linearity and changing terrain make it difficult to design an appropriate steering controller for off-road vehicles using traditional design methods. This paper describes a steering controller design method for off-road vehicles using identified steering system parameters and reports the validation results of using the designed steering controller in field tests.

## 2. Model recognition of E/H steering system

E/H steering systems are commonly used on off-road vehicles. To design a steering controller capable of performing prompt and accurate steering control for an off-road vehicle, it is important to recognize the characteristics of the E/H steering system and to identify the key parameters affecting steering performance. The goal of this research was to develop a practical method for designing an off-road vehicle E/H steering controller using a model recognition approach.

In the experimental approach, both time domain and frequency domain analyses were used to determine the parameters of the E/H steering system model. Time domain parameters included the rise time, the settling time, and the percentage overshoot. The frequency domain parameters included both the gain and the phase margins. System non-linearities, including deadband, saturation, asymmetry, and hysteresis, were determined experimentally.

The research was based on a CaseIH [1] Magnum 7220 2-WD agricultural tractor platform (Fig. 1). Fig. 2 shows the hydraulic steering actuator (the hydraulic

[1] CaseIH is a trademark of Case Corporation. Mention of trade name, proprietary product or specific equipment does not constitute a guarantee or warranty by the University of Illinois, and does not imply the approval of the named product to the exclusion of other products that may be suitable.

Fig. 1. CaseIH Magnum 7220 2-WD agricultural tractor.

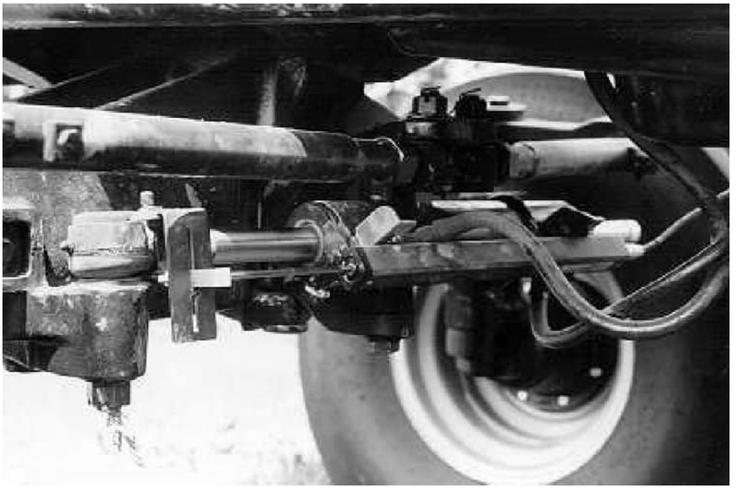

Fig. 2. Hydraulic steering actuating system on the test vehicle.

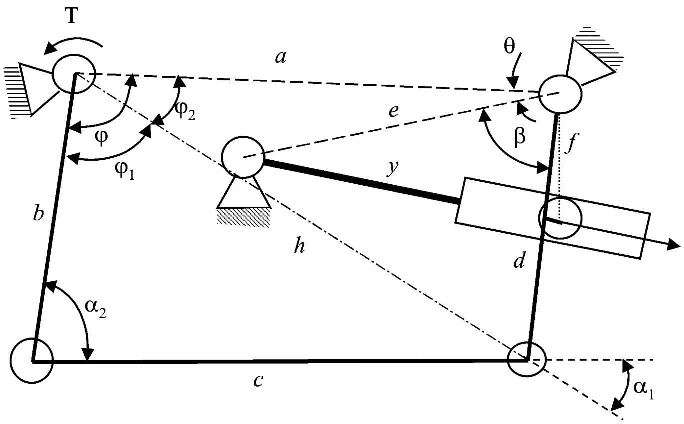


Fig. 3. Sketch of the four-bar steering linkage used in the agricultural tractor. Bar $a$ is the frame of the tractor supporting the rest of the bars. The hydraulic cylinder $y$ actuates bar $d$ in performing steering operations. When bar $c$ transmits the motion from bar $d$ to bar $b$, bar $b$ actually turns the front wheel of the tractor to complete the steering operation.

cylinder) and the attached linear potentiometer (used to measure the displacement of the hydraulic cylinder). The hydraulic cylinder drives a steering linkage, which is a four-bar system (Fig. 3), to turn the front wheels of the tractor to complete the steering. The motion of the hydraulic steering actuator was controlled using an E/H steering valve. Fig. 4 depicts a hydraulic schematic of the E/H steering system. An on-board computer was used for real-time control and data analysis.

### 2.1. Model of steering linkage gain

As discussed earlier, this model was developed as the basis for designing an automatic controller for the E/H steering system. Because of the high non-linearity of the E/H steering system, it is important to consider non-linearities in the model development. One source of non-linearity came from the steering linkage gain between the displacement of the steering actuator and the rotating angle of the front wheels. This linkage gain could be determined from the geometric relationship between the steering cylinder displacement, $y$, and the kingpin arm angle, $\varphi$ (as shown in Fig. 3),

$$\varphi = \varphi_1 + \varphi_2 = \cos^{-1}\left(\frac{b^2 + h^2(\beta) - c^2}{2bh(\beta)}\right) + \cos^{-1}\left(\frac{a^2 + h^2(\beta) - d^2}{2ah(\beta)}\right), \tag{1}$$

where

$$h(\beta) = \sqrt{a^2 + d^2 - 2ad\cos(\beta(y) + \theta)} \tag{2}$$

and

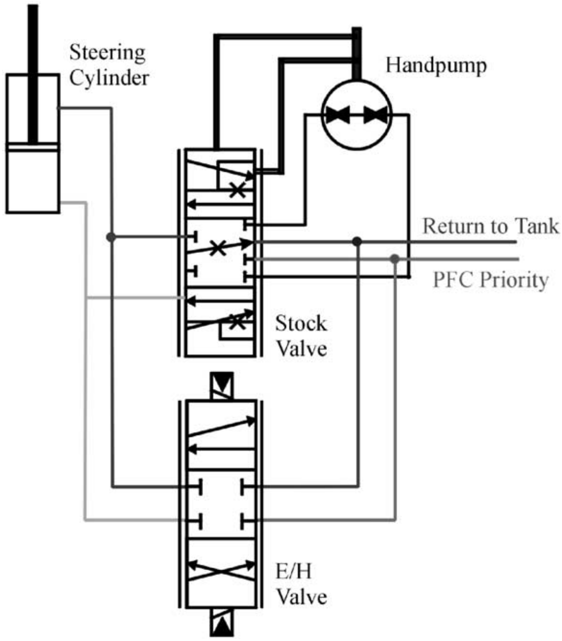


Fig. 4. Schematic of the electrohydraulic steering system on the research platform. It consists of a stock valve which is the standard design for CaseIH Magnum 7220 2-WD tractor, and an E/H valve which is the steering control valve added to the research platform for implementing automated steering control.

$$\beta(y) = \cos^{-1}\left(\frac{e^2 + f^2 - y^2}{2ef}\right). \tag{3}$$

The dynamic model of the steering system was derived from the steering linkage model,

$$I\ddot{\theta} = T - \eta\dot{\theta} - T_{\mathrm{s}} - T_{\mathrm{f}}, \tag{4}$$

where

$$T = k\cos\alpha_1 \sin\alpha_2(P_1A_1 - P_2A_2). \tag{5}$$

A simulation model was developed using LabVIEW and these previous equations to evaluate the linearity of the steering linkage gain. The simulation results indicated that the steering linkage gain was almost linear for a steering angle range of −30° to +30°. The linear range was only a portion of the vehicle steering angle range (approximately −58° to +58°), but an off-road vehicle would mostly operate within this range during normal operation. Therefore, it is reasonable to assume that the steering linkage is a linear plant for position control in steering operation. However, the corresponding steering torque on the front wheel kingpins is highly non-linear (Fig. 5).

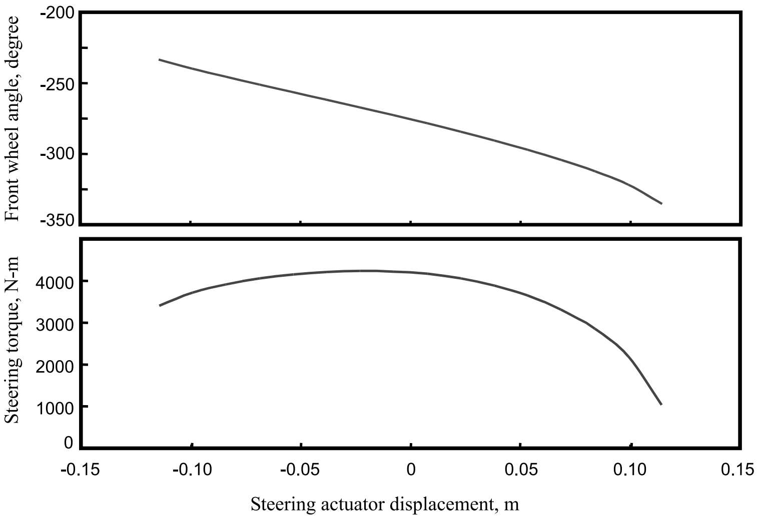


Fig. 5. The relationship between the front wheel angle or steering torque (at left-side kingpin) and the displacement of the steering cylinder actuator.

### 2.2. Model of E/H steering system

First-order and second-order dynamic models are commonly used to describe the behavior of an electrohydraulic actuating system [8]. This research evaluated both models in representing the dynamic behavior of the E/H steering system and identified the parameters via system identification. The first-order model was an idealized representation of the E/H steering system,

$$G(s) = \frac{1}{\tau s + 1}. \tag{6}$$

The only system parameter identified in the model was the time constant, $\tau$, which was determined according to the system response to an input step steering command. The first-order model could not adequately describe the dynamic behavior of the E/H steering system, since it could not reflect the effects of system dynamic characteristics. A second-order model was evaluated to solve this problem,

$$G(s) = K_i \frac{\omega_c^2}{s^2 + 2\xi_c \omega_c s + \omega_c^2}. \tag{7}$$

The second-order model incorporated the dynamic behavior of the E/H steering system in the form of system damping ratio and system natural frequency. There was a large deadband and saturation in the E/H steering system. The steady-state relationship between the input steering command and the front wheel angle was represented using a piecewise linear function (Fig. 6),

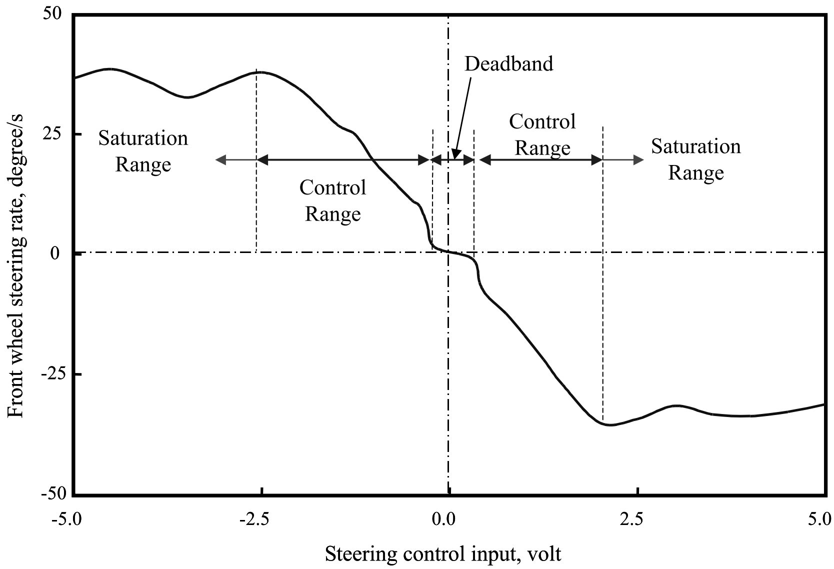


Fig. 6. Steady-state characteristics of electrohydraulic steering system of an off-road vehicle.

$$K_i = \begin{cases} 0 & \text{if in deadband or in saturation range,} \\ 1 & \text{if not in deadband or in saturation range.} \end{cases} \tag{8}$$

The steady-state gain was used to represent the actual system dynamics or to attenuate the effects of system non-linearity, mainly the system deadband and saturation, according to the domain of the input steering control signal. This non-linear gain approach represented the system dynamics more accurately by ignoring the effects of deadband and saturation in system response to input commands. Therefore, this adjustment allowed a second-order model to represent the dynamic behaviors reasonably well in light of the E/H steering system non-linearities.

The high-frequency behaviors of an E/H system are difficult to identify. The pressure wave within the cylinder chamber shows up as high-frequency oscillations in the feedback signal from the position sensor. The solenoid actuators of the E/H valve also contribute to the high-frequency behaviors of the steering system. Other noise sources, including pump pressure and tire–ground interaction, also affected the dynamic behavior of the system. These factors make it difficult to develop an exact system model of the E/H steering system using analytical methods. This research used a system identification approach to characterize the E/H system that, in turn, supported the control system design.

### 2.3. *Identification of system non-linearity*

Agricultural tractor steering systems have several non-linear characteristics including deadband and saturation. The deadband affects the dynamic performance of the steering controller and results in lag and/or unstable response. Saturation limits the operating range of the steering control system. Quantitative evaluation of non-linear elements was applied in control system design. Stombaugh [9] investigated the non-linearity of an E/H steering valve and found that the non-linearity was influenced by the valve, cylinder and control system characteristics as well as by the wheel–ground interactions.

To develop a complete and accurate model of E/H steering for an off-road vehicle, the steering system was tested under both heavily loaded (tractor traveling at higher speeds on asphalt) and no-load (vehicle stationary with steering axle lifted into the air) conditions. In system identification tests, steering commands were sent to control the E/H steering valve, and the system pressure and tractor front wheel angle were recorded. A variety of steering inputs, including step and ramp inputs, were used to identify the steering system model. The test results showed that the static friction in the steering system and the dynamic steering torque varied with vehicle speed and with variations in the wheel–ground interaction [10]. The influence of unknown factors on both the static and dynamic behavior of the electrohydraulic steering system was treated as disturbances in the analysis.

Steering deadband was defined as a region in which a change in input signal produced no output response. The deadband was determined by slowly applying a ramp input to the electrohydraulic valve and monitoring the steering angle feedback to determine when the system reacted. The deadband was measured for several different road surfaces and conditions. Steering deadband was caused by several factors including valve spool overlap, hydraulic system pressure, and fluid temperature. It is difficult to develop an exact multi-variable function to account for the contribution of each of the factors. System identification test results (Fig. 6) showed that the deadband was asymmetric and varied with steering direction. The results also showed a correlation between asymmetry and ground condition.

The response range of the electrohydraulic steering valve and the controller should be properly matched to avoid driving one element into saturation. For the test vehicle, the A/D output card from the steering controller was capable of providing a voltage output from $-5$ to 5 V with 0 V representing a zero-angle steering input. However, the steering controller hardware was limited to a range of 0.65–4.6 V and was non-linear between the voltage limits (Fig. 6).

### *2.4. Development of E/H steering model*

The linearity of the test system was evaluated by recording the steering rate under a constant input. The results showed that the steering rate was almost linear until the system saturated (Fig. 6). The relationship between input voltage and steering rate implied that the E/H steering system acted like an integrator, which resulted in higher-order dynamic behaviors.

A non-linearity gain (defined by Eq. (8)) and a transfer function were used to represent the E/H steering system. An inversed transfer function representing the steady-state characteristics of the system was developed from the open-loop identification test. The inversed transfer function, developed based on the results showed in Fig. 6, was applied to the system to reduce the effect of system non-linearities.

To obtain satisfactory control performance, the sampling and control interval (0.01 s) were set to one-eighth to one-tenth of the time constant of the system. In addition, precautions were taken to avoid driving the output into saturation. Because the natural frequencies of the E/H valve and the pressure wave in cylinder chamber were always higher than the natural frequency ($\omega_c$) of the cylinder, and those higher-frequency components were difficult to distinguish, an additional integrator was introduced to represent the effects of such higher-frequency behaviors of the steering system. As discussed in Section 2.2, a second-order model was used to represent the basic dynamic behaviors of the steering system. The cylinder phase crossover frequency, as shown in Fig. 7, was 15 rad/s (2.4 Hz). The curves in Fig. 7 also indicate that the electrohydraulic steering system was an over-damped system. With the additional integrator, the dynamic model of the steering system was, therefore, identified as

$$G(s) = \frac{K_i}{s} \frac{225}{s^2 + 135s + 225}. \tag{9}$$

The integrator made the E/H system model a velocity control model, which matched the nature of E/H system control. The identified model was used as the base model for designing an E/H steering controller. Fig. 8 shows the block diagram of the steering controller developed. In this controller, a feedforward signal was added to overcome the large deadband [11].

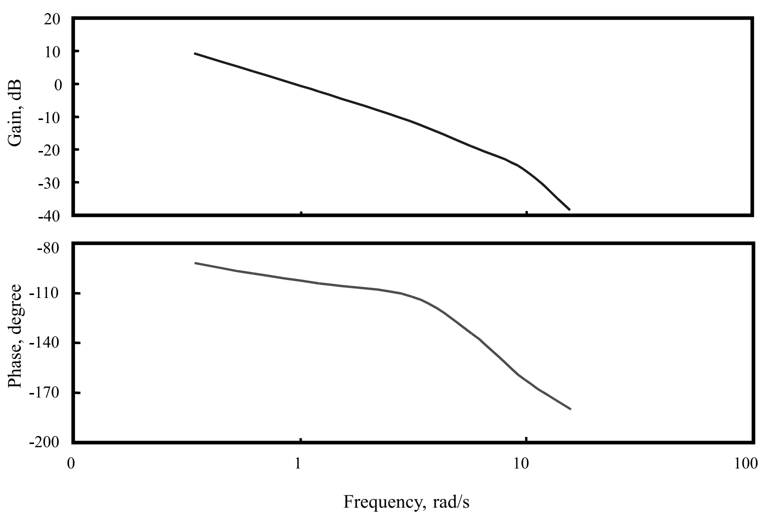


Fig. 7. Gain and phase characteristics of electrohydraulic steering system of an off-road vehicle.

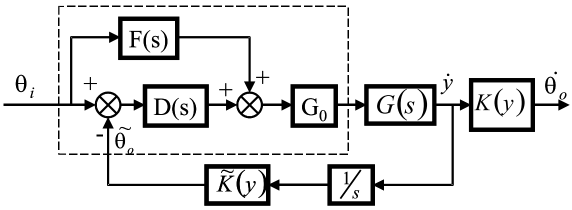


Fig. 8. Block diagram of E/H steering control system for the agricultural tractor.

## 3. Model simulation and validation tests

The model of the E/H steering control for the agricultural tractor was evaluated by digital simulation and on-vehicle tests. The simulation model was developed using the identified E/H steering model, and was programmed using the Matlab Simulink toolbox. The performance of the steering controller was evaluated using on-vehicle tests and compared with the simulation results. In the simulation and validation tests, both modulated steering commands (step and sinusoidal inputs) and actual steering commands (recorded from typical steering operations) were used. The vehicle tests were conducted on paved road and in fallow fields.

Fig. 9 shows the results from both the simulation and on-vehicle steering validation test on paved road from a step steering input. The validation test indicated that the simulation model could accurately predict the E/H steering system dynamics

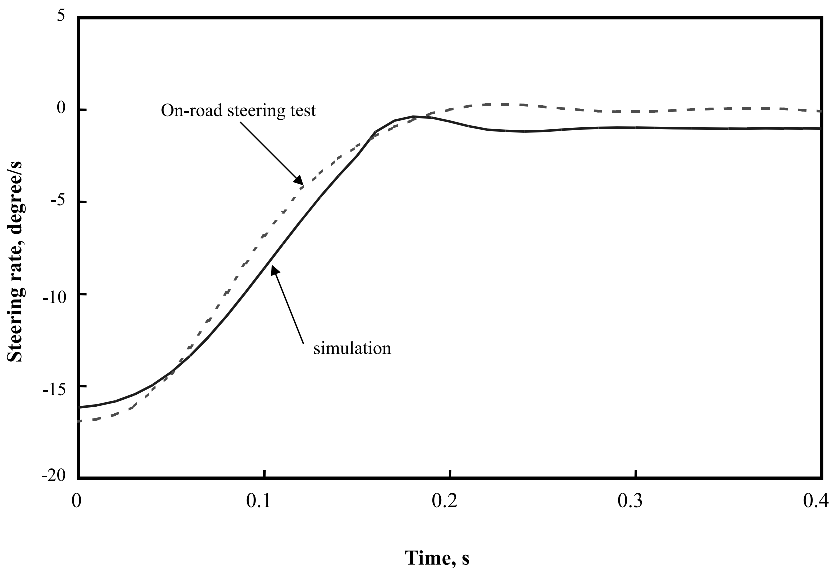


Fig. 9. Comparison of simulation and test results on front wheel steering rate control using a step-input control signal. The vehicle test was conducted on paved road.

in on-road steering. Little overshoot corresponding to the step input verified the conclusion that the E/H steering system had a high damping ratio.

A sine wave input was used to further evaluate the controller performance (Fig. 10). The off-road steering test indicated that there was a noticeable lag between the simulation and the actual response in the wheel angle while the vehicle was steering on a fallow field. This resulted in a maximum error of 1° in wheel angle for input sinusoidal signals between 0.1 and 0.5 Hz. Note that the simulation model was identified based on "no-load" test results, which did not take the tire stiffness and ground condition into consideration. Such lag in response reflected the effect of vehicle–ground interaction. As shown in the test results, the steering controller still achieved the desired steering angle even with the noticeable lag in response.

Further tests were performed to evaluate the steering controller performance on different ground conditions using an actual steering command recorded from a normal steering operations (Fig. 11). The results from simulation analysis and vehicle tests indicated that the simulation model could predict the actual steering performance reasonably well even there were significant deviations in realized steering angle control when the vehicle was traveling on different surface conditions. The steering controller was capable of achieving prompt and accurate steering control of an off-road vehicle regardless of the non-linear nature of its E/H steering system and the large variation in ground conditions.

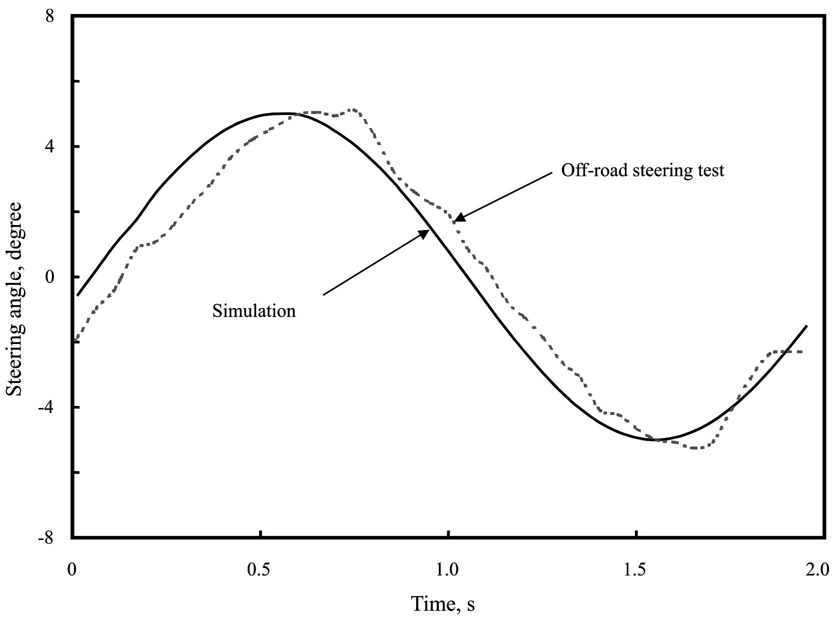


Fig. 10. Comparison of simulation and test results on front wheel angle control using a sine wave control signal. The vehicle test was conducted on fallow fields.

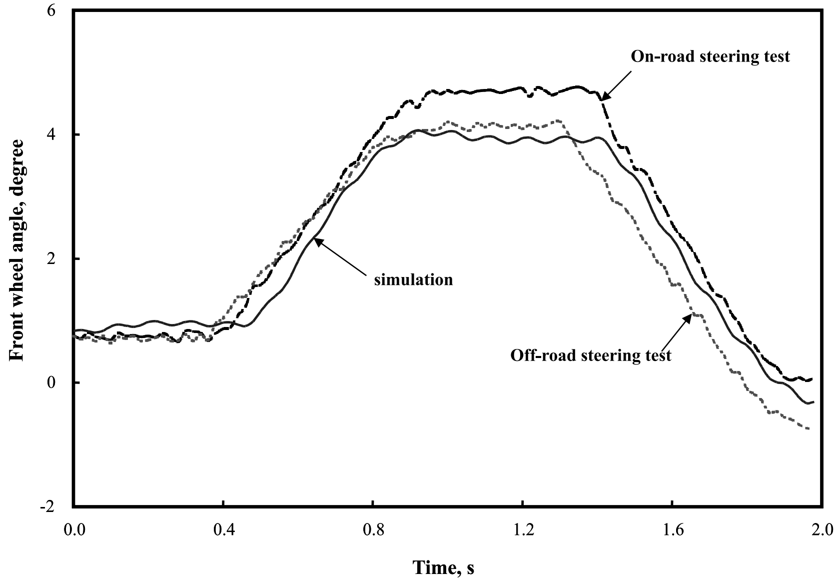


Fig. 11. Comparison of simulation and test results on controlling the front wheel angle using an actual (recorded) steering control signal.

## 4. Conclusion

An accurate steering controller is essential for autonomous off-road vehicles. Many off-road vehicles use an E/H actuator to implement automated steering control. This paper presents a practical method of designing an automated E/H steering controller for off-road vehicles. This method identifies the dynamic system parameters of an E/H steering control model based on a series of static and dynamic steering tests. Test results were used to identify the non-linear and dynamic characteristics of the E/H steering system.

Nonlinear gains were used to compensate deadband and saturation. The steering linkage gain relating the steering actuator displacement to the wheel angle was calculated from the steering geometry. The system model was used for closed-loop steering control based on the feedback signal from the hydraulic steering actuator rather than from the front wheels.

The performance of the steering controller was simulated using a Matlab model and tested in an off-road vehicle. The simulation results were validated with both on-road and off-road steering tests. The simulation and subsequent validation proved that it was possible to successfully develop a steering controller using a model identification approach. The developed steering controller was capable of achieving prompt and accurate steering control on an off-road vehicle regardless of the non-linear nature and the changing ground conditions.

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