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Original Article

Experimental Study of Dynamic Parameters on the Model of Electric Power Steering System

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Abstract *- This study presents an experimental survey on the dynamic parameters of an Electric Power Steering System (EPS) using a dedicated experimental model developed for dynamic performance investigation. The proposed steering system model simulates steering wheel rotation and road-induced reaction forces through adjustable wheel loads, while simultaneously measuring key characteristic variables, including steering torque, steering angle, motor current, angular velocity, and reaction force at the steering wheel rim. Based on the experimental data, systematic analyses and comparisons between theoretical predictions and experimental results are conducted to determine the key factors affecting system dynamics and operational stability. The results indicate that the proposed experimental model accurately represents the dynamic characteristics of the EPS system under various operating conditions. Experimental investigations are performed at different simulated vehicle speeds of 0 (km/h), 40 (km/h), 60 (km/h), and 100 (km/h). The findings reveal a significant variation in the correlation between the driver's input torque and assist torque as a function of vehicle speed. The maximum deviation between experimental results and the theoretical model across all speed conditions is less than 0.03%, confirming the high accuracy of the developed model and its effectiveness in capturing the dynamic behavior of the EPS. The proposed approach provides a reliable experimental platform for dynamic analysis and design optimization of EPS.*

Keywords *- Assist torques, Electric Power Steering systems, Experimental modeling, Steering system dynamics, Steering torques.*

1. Introduction

The EPS system is not only responsible for reducing the driving force for the driver, but also directly affects the driving feel, controllability, and vehicle stability. The dynamic characteristics of the system, including power assist torque, equivalent rigidity and damping, mechanical friction, response hysteresis, and dynamic characteristics of the electric motor, determine the level of precision and stability of the steering control process. In operating situations such as turning at low speeds, changing lanes quickly, or cornering at high speeds, unreasonable variation of these parameters may result in a lack of road surface response, steering wheel oscillation, or impairment of the driver's steering feel. The Electric Power Steering system (EPS) supports the driver's steering force, creating a steering feel, stabilizing steering, and increasing vehicle stability. The dynamics of the EPS system depend on a number of parameters, such as steering torque, power assist torque, steering wheel rotation angle, resistance torque of the guide wheel, friction torque in the system, etc. In different operating conditions, depending on speed, lane change, road surface, etc., the steering force, steering feel, and vehicle stability are affected.

There are many studies on EPS systems, including research on standards and safety requirements for EPS systems [1, 2, 10]; research on theoretical modeling and dynamic analysis of EPS systems [3, 11, 12]; research on the design and selection of electric drive systems for EPS [4, 5]; research on intelligent control algorithms and adaptive control to increase the sense of steering [7, 8, 9]; simulation studies and building experimental models for steering systems [6], [12]. These studies have provided an important theoretical and technical basis for the development of EPS.

However, it is difficult to accurately evaluate the dynamic parameters of the EPS system under real working conditions. In-vehicle direct tests require high costs, complex test environments, and a limited number of tests. To overcome the above limitations, this study builds a semi-experimental model of an electric power steering system using a laboratory test stand.

The construction model combines theoretical research and experimental measurement, allowing for the change of the load applied to the guide wheel and evaluating the vehicle speed-dependent power assist characteristics (speed simulation) without the need for full-vehicle testing.

Through the comparison of theoretical simulation results with experimental results at different vehicle speeds, the research method reliably analyzes the dynamic behavior of the EPS system and provides and calibrates the control parameters.

The novelty of this study is the experimental testing of the dynamic parameters of EPS on a semi-experimental model with variable loads in accordance with the actual operating conditions of the vehicle. Allows direct comparison between theory and experiments on multiple vehicle speeds. Provides an accurate and efficient experimental method for EPS to optimize the system.

2. Theoretical Background and System Modeling

2.1. Building Dynamics Equations

The electric power steering system consists of: steering rim, steering axle, power assist electric motor, steering mechanism, steering links, and guide wheels (Figure 1). The dynamics model is built with the following assumptions: The links in the system are mechanical bonds and are represented by a spring-damping system.

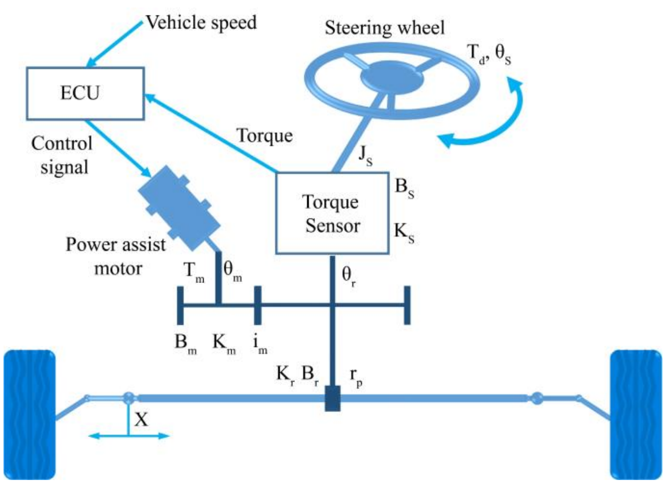

Fig. 1 Diagram of the dynamic model of the electric power steering system

The motion equations are built on the dynamics and adjusted for EPS [13]:

Equation from the steering wheel to the steering column:

$$J_s\ddot{\theta}_s = T_d - K_s(\theta_s - \theta_r) - B_s\dot{\theta}_s \tag{1}$$

Equation of the rack and pinion with displacement x:

$$m\ddot{x} = \frac{1}{r_p}[K_m(\theta_m - i_m\theta_r)i_m + K_s(\theta_s - \theta_r)] - B_r\dot{x} - K_r x \tag{2}$$

Where:

T_d: input torque; J_s: represents the moment of inertia; K_s, B_s: stiffness and damping coefficients; K_r: stiffness coefficient of the steering rack; B_r: elastic damping coefficient of the tie rod ends; m: mass of the steering rack; i_m: gear reduction ratio; r_p: radius of the rack; θ_s, θ_r, θ_m: rotational angular positions of the steering wheel, steering shaft, assist motor, respectively; x: lateral rack displacement.

Mathematical model of the power assist system:

$$J_m\ddot{\theta}_m = T_m - K_m(\theta_m - i_m\theta_r) - B_m\dot{\theta}_m \tag{3}$$

Where:

T_m: output torque of the assist motor; J_m: moment of inertia; K_m: stiffness; B_m: elastic damping coefficient.

2.2. Mathematical Model of the Electric Motor

The electric motor considered is a PMSM drive motor that receives control signals from the ECU to generate appropriate assist torque. The following assumptions are made: Current losses and hysteresis effects are ignored; the motor current is a symmetric three-phase sinusoidal waveform; resistance and inductance values are considered constant.

Fundamental equations of the PMSM are as follows [4, 13].

Voltage equations for the three stator windings:

$$\begin{cases} u_{sa}(t) = \dfrac{d\psi_{sa}(t)}{dt} + R_s i_{sa} \\ u_{sb}(t) = \dfrac{d\psi_{sb}(t)}{dt} + R_s i_{sb} \\ u_{sc}(t) = \dfrac{d\psi_{sc}(t)}{dt} + R_s i_{sc} \end{cases} \tag{4}$$

Where: R is the resistance of the stator phase winding; $\psi_{sa}, \psi_{sb}, \psi_{sc}$ are the magnetic flux linkages of stator windings A, B, and C.

Applying the voltage equation, we have:

$$u_s(t) = \frac{2}{3}[u_{sa}(t) + a.u_{sb}(t) + a^2 u_{sc}(t) \tag{5}$$

Substituting the phase voltages from (4) into (5), we obtain the stator voltage equation in vector form:

$$u_s = \frac{d\psi_s}{dt} + R_s i_s \tag{6}$$

Equation (6) is derived from analyzing the three-phase stator winding system; therefore, it can be rewritten as:

$$u_s^2 = \frac{d\psi_s^2}{dt} + R_s i_s^2 \tag{7}$$

Transforming Equation (7) into the d-q coordinate system, we obtain:

$$u_s^f = \frac{d\psi_s^f}{dt} + R_s i_s^f + j.\omega_s \psi_s^f \quad (8)$$

The stator flux vector ψ_s^f has only a real component ψ_p because the real d-axis aligns with the main flux vector ψ_s^f, thus $\psi_s^f = \psi_p$.

The stator flux vector ψ_s^f consists of two components: One component due to the self-inductance of the stator windings, and another component, ψ_p, which is the induced flux:

$$\psi_s^f = L_s i_s^f + \psi_p^f \quad (9)$$

Torque equation of the motor:

$$T_M = \frac{3}{2} p_c (i_s \psi_s) \quad (10)$$

Equation of motion:

$$T_M = T_T + \frac{j d_\omega}{p_c dt} \quad (11)$$

In a synchronous motor, a fixed rotor flux direction always exists. Therefore, for this motor, the mathematical description is based only on the d-q coordinate system observation.

2.3. ECU Control Model

The ECU collects signals from the torque and vehicle speed sensors to select the appropriate motor current, and executes control through algorithms that compare the input signals with feedback signals. The assist characteristic of the EPS system defines the correlation between the input signals and the output signals.

The linear assist characteristic can be expressed as:

$$i = \begin{cases} 0 & (0 \le T_d < T_{d0}) \\ k(v)(T_d - T_{d0}) & (T_{d0} \le T_d < T_{dmax}) \\ i_{max} & (T_{dmax} \le T_d) \end{cases} \quad (12)$$

Where:

i: output current of the motor; T_d: torque on the steering wheel; T_{d0}: steering torque at which assistance begins; T_{dmax}: represents the steering torque at which the maximum assist is achieved; $k_{(v)}$: assistance coefficient, dependent on vehicle speed; i_{max} is the maximum motor current.

3. Materials and Methods

On the basis of the theory of building a dynamic representation of the electric power steering system and a survey using Matlab - Simulink software, this study conducted an experimental study on the model dynamic representation of the electric power steering system and, specifically, the operation within the steering system and distribution valve assembly using Dewesoft X software. Thereby, comparing the results between the surveyed theory and the experiment, and simultaneously completing the simulation connection between the model and computer software, improves the efficiency of the paper.

The survey model is a semi-experimental model, allowing the replacement of the main components, such as power assist motors, torque sensors, and ECUs, so that they can be tested for many types of vehicles, including commercial vehicles that meet conditions close to reality.

The model can also be adjusted to upgrade the load-generating unit to handle large loads for testing vehicles with large loads; at the same time, it is possible to add a module to simulate tire interaction with the road surface in accordance with actual conditions.

3.1. Design 3D of the EPS

The design experimental model meets a number of requirements, such as: The design model is similar to the real EPS system on the vehicle; it meets the changing load placed on the guide wheels; there is a car speed simulator.

Based on these requirements, the experimental model was designed using Autodesk Inventor, as shown in Figure 2.

Fig. 2 Experimental model of the electric power steering system

The loading mechanism on the steered wheels is designed to apply an adjustable load to the steered wheels by varying the jack height (Figure 3). The steering wheels are fixed to the steering system frame. The load is applied by varying the height of a jack positioned under the wheel assembly. As the jack height increases, the corresponding load applied to the wheels also increases. A specialized load cell is installed between the steered wheels and the jack to measure the force exerted by the jack accurately.

Fig. 3 Design of the jack for applying a load to the steered wheels

3.2. Fabrication of the Experimental Model

After completing the 3D design, 2D CAD drawings of the model components were generated, and the prototype was fabricated according to the design specifications. Trial runs were then conducted, and necessary adjustments were made to eliminate design errors and ensure proper functionality.

Following fabrication, sensors were installed on the prototype (Figure 4), and a measurement procedure was developed for the EPS system parameters. The measurable parameters include steering torque, steering speed, the number of steering wheel rotations, lateral rack displacement, and the load applied to the two steered wheels. Data acquisition was performed using the Dewesoft DEWE-43 system, which collects signals directly from the installed sensors (Figure 5).

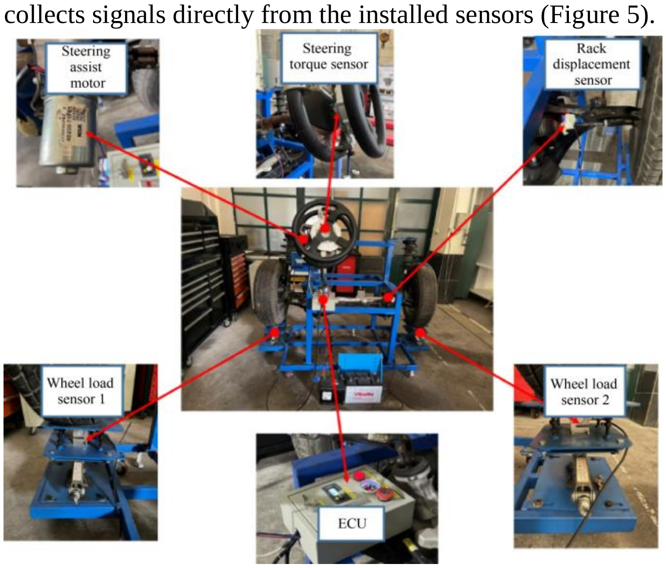

Fig. 4 Completed experimental model of the electric power steering system

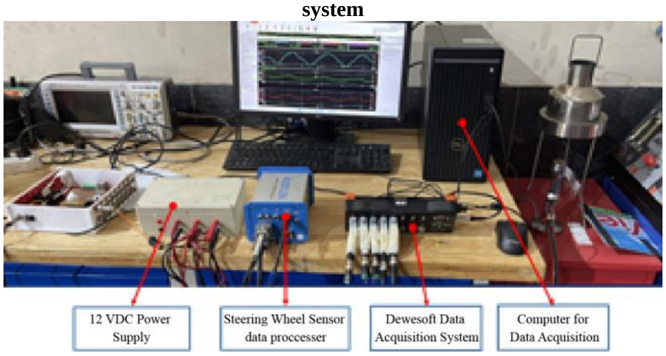

Fig. 5 Data acquisition system with sensors, DEWE-43 acquisition device, and computer

4. Results and Discussion

Figure 6 illustrates the change in the steering wheel angular response over time. On the basis of this input signal, the response of the EPS is assessed at different simulated vehicle speeds, including 0 (km/h), 40 (km/h), 60 (km/h), and 100 (km/h). The simulation results of steering torque, engine torque, and power assist torque are presented in Figures 7 through Figure 10, respectively.

The change in the steering wheel angle following a sinusoidal profile is illustrated in Figure 6.

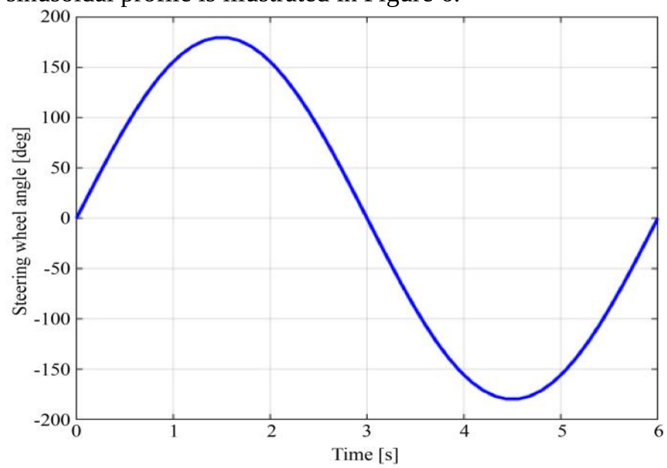

Fig. 6 The variation of the steering wheel angle over time

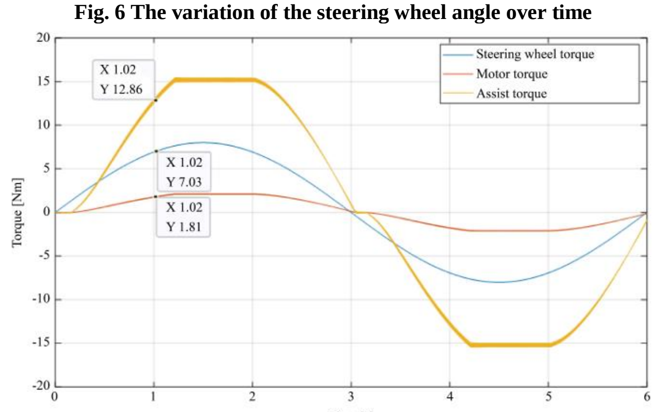

Fig. 7 Steering wheel torque and assist torque at 0 km/h

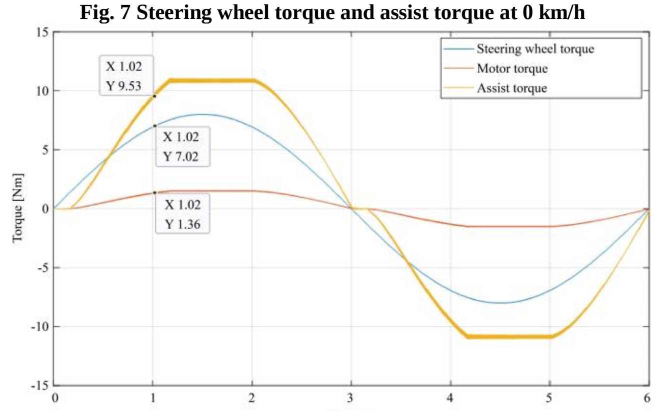

Fig. 8 Steering wheel torque and assist torque at 40 km/h

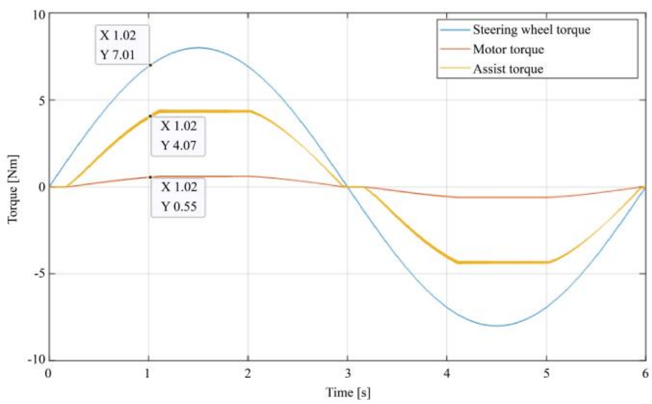

Fig. 9 Steering wheel torque and assist torque at 60 km/h

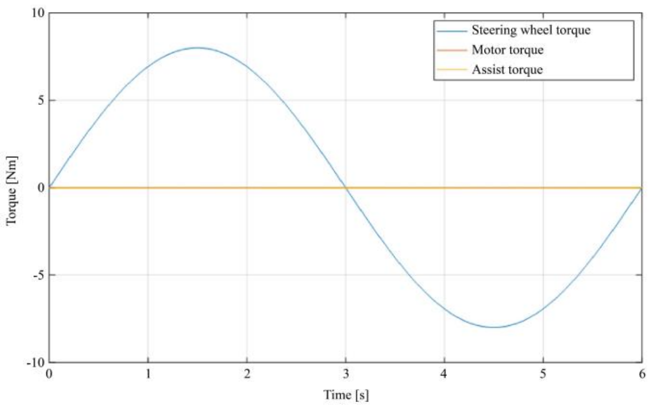

Fig. 10 Steering wheel torque and assist torque at 100 km/h

Table 1. Comparison between theoretical and experimental EPS assist torque at different vehicle speeds

Vehicle speed (km/h)	Maximum driver input torque Td (Nm)	Max. assist torque (Theory) (Nm)	Max. assist torque - Exp (Nm)	Max. dev (%)
0	7.0	15.00	14.96	0.27
40	7.0	10.20	10.17	0.29
60	7.0	6.80	6.78	0.29
100	7.0	0.00	0.01	0.14

Table 1 presents the simulation results for the maximum Electric Power Steering (EPS) values obtained from the theoretical model and empirical measurement results at different vehicle speeds, with the driver's torque kept constant at 7.0 Nm. The theoretical model gives a maximum assist torque value of 15.00 Nm, while the experimental measurement value reaches 14.96 Nm, corresponding to a maximum deviation of 0.27%. This large level of power assistance significantly reduces the steering rim control force in on-site turns and parking situations.

As the vehicle speed increases to 40 km/h, the maximum power assist torque decreases to 10.20 Nm (theoretical model) and 10.17 Nm (experimental measurement), with a deviation of 0.29%. The same trend was recorded at 60 km/h, where the maximum power assist torque was 6.80 Nm (theoretical model) and 6.78 Nm (experimental measurement), respectively, with a deviation of 0.29%. These results show the speed-dependent power assistance characteristic of the EPS, where the level of power assist gradually decreases as the speed of the vehicle increases to enhance driving stability and the feeling of feedback from the road surface.

At high vehicle speeds (100 km/h), the theoretical model almost completely suppresses the power assist torque (0.00 Nm), while the experimental measurement value is only 0.01 Nm, corresponding to a deviation of 0.14%. This proves that the ECU has effectively controlled the reduction of power assist at high speeds, avoiding the phenomenon of the steering rim being too sensitive and contributing to improving the directional stability of the vehicle.

The results showed that the maximum deviation between theoretical and experimental results across all survey conditions was less than 0.3%, indicating a very high degree of relevance of the proposed quasi-experimental model. The minor discrepancies that still exist mainly stem from factors that have not been taken into account in the model, such as mechanical friction, elasticity of the joints in the steering mechanism, and the delay caused by steering wheel rotation during the experiment.

This result confirms that the research model is highly accurate and reliable and demonstrates its applicability to dynamic analysis, parameter calibration, and the optimization of EPS design.

Measured torque and steering angle from the experimental model, recorded at the same speed in the simulations, are presented in Figures 11 to 14.

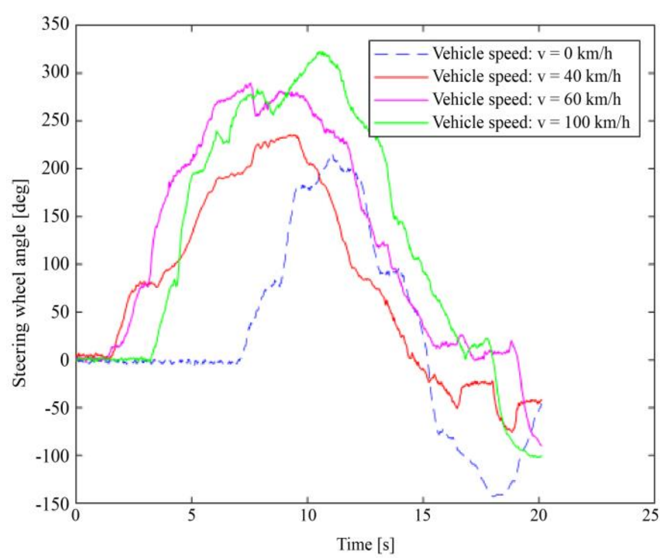

Fig. 11 Steering wheel angular displacement at different vehicle speeds under non-assisted conditions

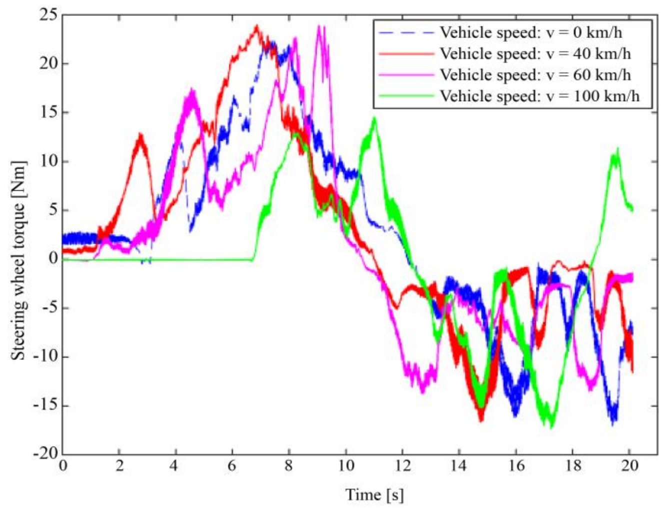

Fig. 12 Steering wheel torque response at different vehicle speeds under non-assisted conditions

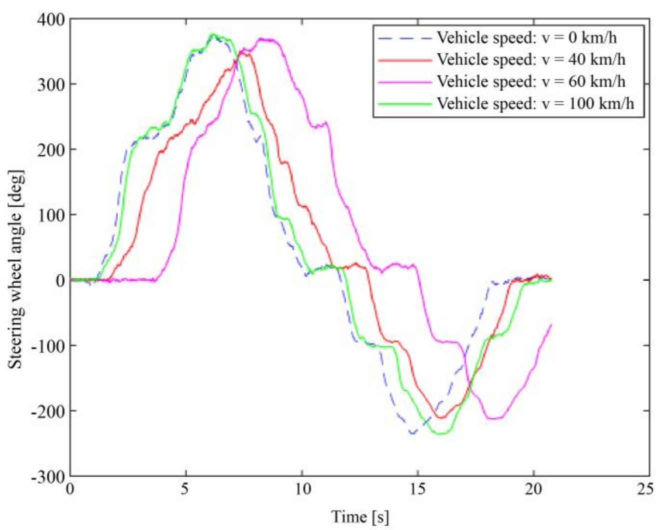

Fig. 13 Steering wheel angular displacement at different vehicle speeds with power assistance

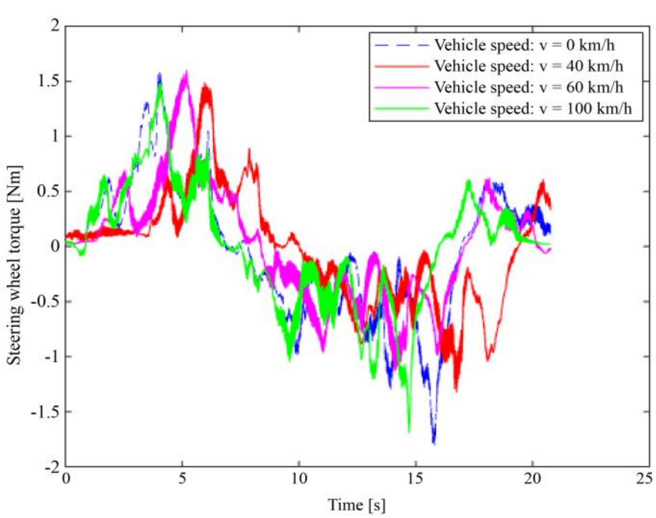

Fig. 14 Steering wheel torque response at different vehicle speeds with power assistance

Figure 11 shows that the steering wheel rotation angle amplitude reaches a maximum value of approximately 300 ÷ 330 (deg) for 40 (km/h), 60 (km/h), and 100 (km/h) cases, while at 0 km/h it is around 200÷220 (deg). Reflects the effect of steering and load resistance on the wheels as the vehicle speed increases.

Figure 12 shows a marked increase in steering rim torque as the vehicle speed increases in the absence of power assist. The maximum steering wheel torque at 0 (km/h) is about 12÷15 Nm, while at 40 (km/h) and 60 (km/h), this value increases to about 20÷23 Nm. At high speeds (100 km/h), the steering wheel torque remains large and fluctuates more strongly, especially at times of steering wheel rotation.

Comparison of Figure 13 with Figure 11 shows that with EPS assist, the steering wheel rotation angle remains at the same level, with a maximum value of about 350÷380 (deg) for different speed cases. The deviation between speeds is negligible and less than 5÷10%, showing that the EPS system does not alter the driver's trajectory or control.

Figure 14 shows the decrease in the torque exerted on the steering wheel. The maximum steering wheel torque in the case of power assist is only about 1.2÷1.6 Nm at all speed levels, which is lower than in the case without power assist (Figure 12). Thus, the EPS reduced the maximum steering wheel torque from 20÷23 Nm to less than 2 Nm, reducing it by more than 90% under the survey operating conditions.

From the above analysis, it can be confirmed that the EPS substantially decreases the steering wheel torque (>90%), proving that the EPS has completed the power assist function well. The results obtained from Figures 11 to 14 demonstrate the reliability of the semi-experimental model and the control of the power unit.

5. Conclusion

The paper has built a model to investigate the dynamics of Electric Power Steering systems in automobiles by simulation methods on computer software, as well as experimental research using models. From there, the application of driving system dynamics survey, comparison, and comparison between theoretical and experimental results, and a conclusion on factors affecting the dynamics and working stability of the system.

However, the current model ignores tire friction and does not take into account the vehicle's dynamics, such as horizontal acceleration. These are factors that significantly affect driving characteristics in real-world conditions, especially in high-speed spinning situations or when steering fast. Another limitation is that parameters such as pavement hardness have not been included in the analysis, which also affects the nonlinear model of the steering system.

Further development research will focus on tire and road interaction, vehicle lateral dynamics, a construction model with a tire-pavement interaction simulation module, and the addition of sensors to further improve the model's accuracy in real driving conditions.

Acknowledgments

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