Simulation of a Steer-by-Wire System with Speed-Dependent Variable Steering Ratio

Boobalasenthilraj A K*, Abishek A, Rohith R. R. Menon, Shri Krishnaa G, Joshika M, Thiruselvan KJ, Yuvaraj D

Department of Automobile Engineering, Sri Venkateswara College of Engineering, Pennalur, Sriperumbudur – 602 117, Tamilnadu, India

*Corresponding author: Boobalasenthilraj A K

Received:Sep 26, 2026 Accepted:Oct 23, 2026 Published:Oct 24, 2026

Abstract

Steer-by-wire (SbW) systems replace the mechanical link between the steering wheel and the road wheels with sensors, a controller and actuators, so the steering ratio can be changed in real time according to vehicle speed and driving mode. This paper presents a MATLAB/Simulink model of an SbW system with a speed-dependent variable steering ratio and uses it to estimate the steering effort that the actuator must overcome, to examine the ratio schedule and the closed-loop response, and to quantify the mass saved by the change of architecture. The model combines a static steering-effort calculation for a compact electric SUV (kerb mass 1400 kg) with a closed-loop model that includes a proportional–integral–derivative (PID) controller, an electric road-wheel actuator and a lookup-table variable-ratio subsystem. The static analysis gives a steering-wheel torque of 46.0 N·m and a rim effort of 153.2 N for the worst-case parking condition, and a sensitivity analysis shows how these values scale with the friction coefficient and scrub radius. The simulated torque and effort responses are smooth and well damped for the tested scenario. Replacing a conventional power-steering assembly (14.0 kg) with an SbW assembly (6.5 kg) saves 7.5 kg (53.6%), but the resulting range gain for a 30.2 kWh battery is below about 2 km even under an upper-bound assumption, so the case for SbW rests on functional rather than mass benefits. Limitations and future work are discussed.

Keywords: steer-by-wire; variable steering ratio; MATLAB/Simulink; steering effort; electric vehicle; weight reduction

1. INTRODUCTION

Conventional steering systems connect the steering wheel to the road wheels through a column, joints and a rack-and-pinion gearbox, assisted by a hydraulic or electric motor. The mechanical path fixes the relationship between steering-wheel angle and road-wheel angle and limits how the system can be packaged. Steer-by-wire technology removes this path: the

driver's input is measured by sensors, processed by an electronic control unit and executed by actuators at the steering rack, while a separate motor generates the steering feel at the wheel [1–3].

Two consequences make SbW attractive. First, the steering ratio becomes a software parameter and can be scheduled with vehicle speed, driving mode or driver preference [3, 4]. Second, the absence of a steering column frees space, reduces mass and simplifies integration with driver-assistance and automated-driving functions [3–6]. The price is safety: without a mechanical fallback, a fault in the steering control can remove the driver's ability to control the vehicle's lateral motion. Redundant hardware and robust control are therefore essential [4], and control designs that ignore degrees of freedom and nonlinearities of the real plant may not be robust enough [7].

Previous work has addressed several aspects of SbW. Reviews describe the architecture, its potential and its challenges [1, 3, 4]. Other studies treat the control of the two-motor system, for example bilateral control of the steering-wheel and rack motors with a disturbance observer [8], and detailed plant models for robust control design [7]. Variable steering ratio strategies have also been proposed, including a design that takes the driver's handling skill into account [9, 10]. This work takes a complementary, system-level view: it links the actuator load, the speed-dependent steering ratio and the steering-assembly mass for one representative electric vehicle in a single, transparent simulation framework.

To support the design of such a system, this work builds a simulation framework for an SbW system with variable steering ratio in MATLAB/Simulink [11]. The objectives are to (i) estimate the steering effort that the actuator must overcome for a representative electric vehicle, (ii) construct a closed-loop SbW model with a speed-dependent steering ratio, and (iii) quantify the mass saving from replacing a conventional power-steering assembly with an SbW assembly. The main contributions are (a) a transparent static estimate of the steering effort, with a sensitivity analysis to the main uncertain parameters; (b) a closed-loop Simulink model in which the steering ratio is scheduled by a lookup table; and (c) a component-level mass comparison with a bounded estimate of the range benefit.

The paper is organised as follows. Section 2 summarises the SbW architecture and the idea of a variable steering ratio. Section 3 describes the static steering-effort model, the closed-loop model with the speed-dependent ratio, and the mass comparison. Section 4 presents and

discusses the results, Section 5 sets out the limitations and future work, and Section 6 concludes.

2. BACKGROUND

This section introduces the two concepts on which the model rests: the architecture of an SbW system and the idea of a variable steering ratio.

2.1 Steer-by-wire architecture

In a conventional electric power steering (EPS) system a single column links the steering wheel and the rack, and an assist motor acts on the rack [4]. In an SbW system the column is replaced by two electrically coupled units (Fig. 1): a steering feedback unit (SFU) with a motor that gives the driver reaction torque, and a steering rack unit (SRU) with a motor that moves the rack according to the commanded road-wheel angle [2, 3]. Sensors on the steering wheel measure angle and torque, and the controller exchanges signals between the two units [8, 12]. Because the commanded road-wheel angle is computed in software, the relationship between the two angles is no longer fixed by hardware, which leads to the idea of a variable steering ratio.

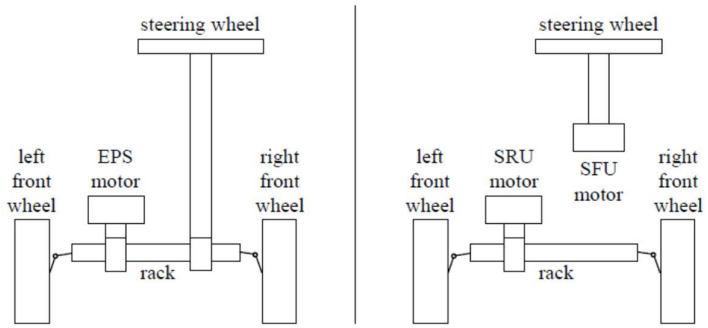


Fig. 1 Layout of (left) an EPS system and (right) an SbW system, showing the steering wheel, SFU motor, SRU motor, rack and front wheels.

2.2 Variable steering ratio

The steering ratio is the ratio of steering-wheel rotation to road-wheel rotation [13]. A ratio of 10:1 means that 10° of wheel rotation produces 1° at the road wheels and gives a direct, agile response; a ratio of 20:1 requires 20° of wheel rotation per degree at the road wheels and gives a calmer response and finer control. A fixed-ratio system must compromise between parking effort and high-speed stability. In an SbW system the ratio can instead be changed

continuously [4]. Benefits reported for this approach include less steering input during low-speed manoeuvres, a more stable response at speed, and different steering characters for sport, comfort and eco driving modes [4, 9, 10]. The model developed in this paper uses this freedom in its simplest form: a ratio scheduled against vehicle speed (Section 3.2).

3. METHODOLOGY

The methodology has three parts. First, a static calculation estimates the steering effort for a representative vehicle (Section 3.1). Second, this load is used in a closed-loop SbW model with a speed-dependent steering ratio (Section 3.2). Third, the component masses of a conventional and an SbW steering assembly are compared (Section 3.3).

3.1 Static steering-effort model

The first step was to estimate the effort that the actuator and the driver must overcome. This was done with a static, worst-case (vehicle stationary) analysis of one front wheel (steering geometry and the moments acting about the steering axis are treated in [13]). The vehicle is a compact electric SUV, the Tata Nexon EV described further in Section 3.3, with a kerb mass of 1400 kg and a 50:50 front-to-rear weight distribution [14], so one front wheel carries a quarter of the kerb mass. The parameters assumed in the calculation are listed in Table 1. The analysis assumes quasi-static conditions, full mobilisation of tyre–road friction (a worst case), equal load on the two front wheels and no power assistance; the assumed friction coefficient and scrub radius are varied in Section 4.1.

Table 1 Assumed parameters for the static steering-effort analysis

Parameter	Symbol	Value
Acceleration due to gravity	g	9.81 m/s²
Perpendicular distance from point of applied force (steering arm length)	i	120 mm
Pitch circle diameter of pinion	PCD	146 mm
Tyre–road friction coefficient	μ	1.1
Steering wheel radius	r_{sw}	300 mm
Scrub radius	r_s	20 mm

Governing equations

The load on one front wheel is

$$W = (m / 4) \cdot g \quad (1)$$

the resisting force at the wheel is

$$R_f = \mu W \tag{2}$$

and the resisting torque about the steering axis is

$$T_{res} = R_f \cdot r_s \tag{3}$$

The torque at the steering knuckle is $T = P \cdot i$, where P is the tie-rod force acting on the steering arm of length i. For the wheel to turn, T must at least equal T_{res}, so

$$P = T_{res} / i \tag{4}$$

Because the pinion force equals P, the torque required at the steering wheel is

$$T_r = P \cdot (PCD / 2) \tag{5}$$

and the effort required at the steering-wheel rim is

$$P_e = T_r / r_{sw} \tag{6}$$

where m is the kerb mass, g the acceleration due to gravity, μ the tyre–road friction coefficient, r_s the scrub radius, PCD the pitch circle diameter of the pinion and r_{sw} the steering-wheel radius (Table 1).

Calculation

Substituting the values of Table 1 (intermediate values are rounded for display) gives

$W = 350 \times 9.81 = 3433.5$ N

$R_f = 1.1 \times 3433.5 \approx 3777$ N

$T_{res} = 3777 \times 20 = 75\,540$ N·mm

$P = 75\,540 / 120 = 629.5$ N

$T_r = 629.5 \times 73 = 45\,951.7$ N·mm ≈ 46.0 N·m

$P_e = 45\,951.7 / 300 \approx 153.2$ N

The same calculation was implemented as a Simulink block diagram (Fig. 2) with gain blocks for the mass split, gravity, friction, scrub radius, steering-arm length and pinion radius, and display blocks for each intermediate quantity. The model reproduces the hand calculation exactly. The resulting effort is used as the load on the road-wheel actuator in the closed-loop model of Section 3.2.

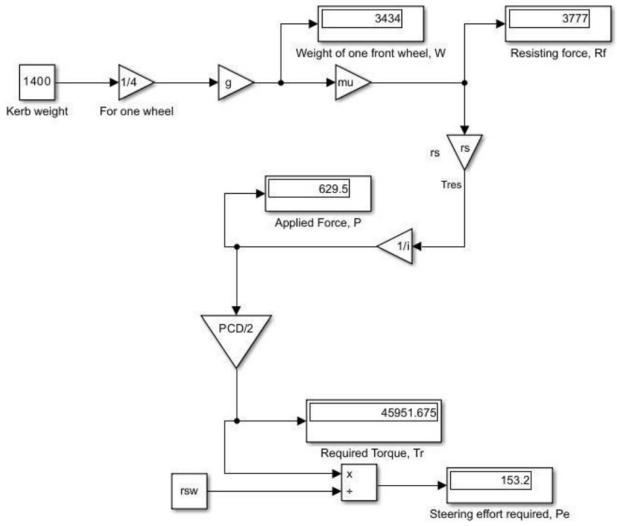


Fig. 2 Simulink implementation of the static steering-effort calculation (W = 3434 N, R_f = 3777 N, P = 629.5 N, T_r = 45 951.7 N·mm, P_e = 153.2 N).

3.2 Closed-loop SbW model with variable steering ratio

The static model was then extended to a closed-loop SbW model (Fig. 3) that links the measured signals, the controller, the actuator and the speed-dependent steering ratio. The main elements are:

- **Input signals:** a steering-wheel-angle signal, a yaw-rate signal, a steering-torque signal and a ramped vehicle-speed signal, each routed through named signal tags and combined into a bus.
- **Reference model:** a first-order yaw-rate reference (cf. single-track vehicle models [15]) is compared with the measured yaw rate to form a yaw-rate error.
- **Controller:** the bundled signals are passed to a proportional–integral–derivative (PID) controller, which produces the command for the road-wheel actuator.
- **Actuator:** an electric motor block represents the steering-rack (road-wheel) actuator; the required effort from Section 3.1 enters as the load.
- **Variable-ratio subsystem:** a Simscape-based subsystem [11] with a one-dimensional lookup table maps vehicle speed to steering ratio, and the resulting ratio is applied in the actuator loop and logged on a scope.

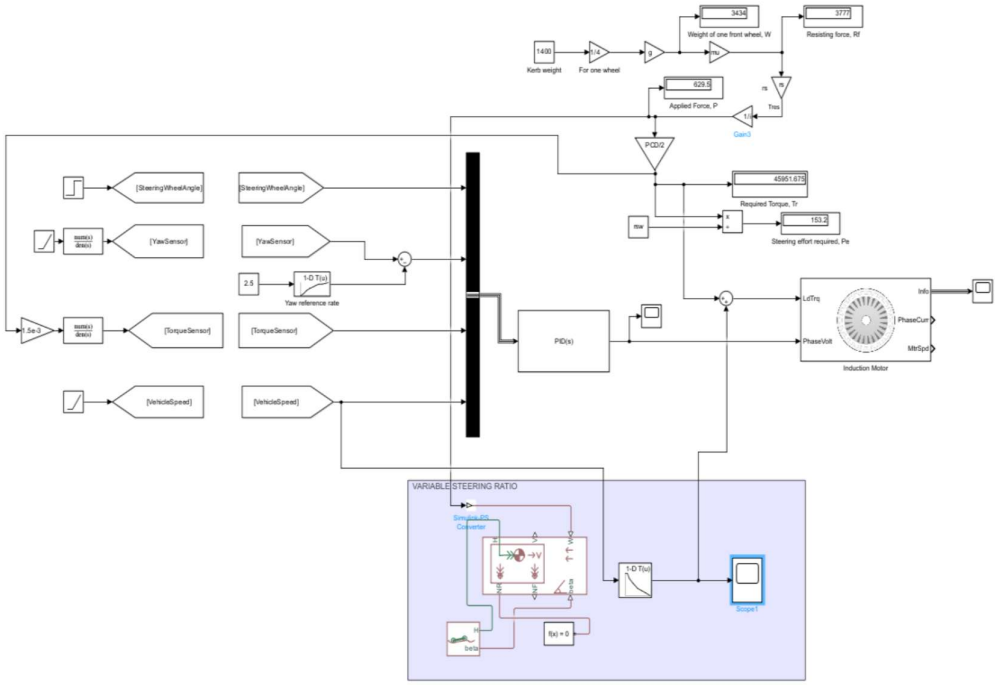


Fig. 3 Simulink model of the SbW system with variable steering ratio.

With these elements in place, the steering-ratio schedule was defined over a vehicle-speed range of 0–50 km/h. The ratio is held at 20:1 up to 10 km/h, falls to 18:1 at 20 km/h and then decreases linearly to 15:1 at 50 km/h. This schedule is the input to the variable-ratio subsystem and is plotted in Section 4.2.

The schedule is illustrative and demonstrates the scheduling mechanism. Variable-ratio strategies are commonly motivated by more responsive steering at low speed and more stable steering at high speed [9, 10]; over the speed range simulated here, the schedule above moves in the opposite direction, so it should not be read as a calibrated recommendation (see Section 5).

Simulation set-up. The closed-loop simulation uses a 5 s scenario with a ramp-type steering input and a ramped vehicle speed over 0–50 km/h. The block parameters of the model, including the controller gains and the solver settings, are contained in the Simulink model files, which are available as described under Data availability.

3.3 Mass comparison

The third part of the study addresses the weight benefit of SbW. Component masses of the conventional power-steering assembly and of an SbW assembly were compiled (Tables 2 and

3) for the Tata Nexon EV (2021) used in Section 3.1, whose kerb mass is 1400 kg and whose traction battery has a capacity of 30.2 kWh [14].

Table 2 Conventional power-steering assembly: component masses

Component	Mass (kg)
Power-steering pump	2.2
Fluid reservoir	0.5
Hoses, lines and fittings	3.0
Steering gearbox	6.8
Steering wheel	1.5
Total	**14.0**

Values are component-level estimates (see Section 5).

Table 3 Steer-by-wire assembly: component masses

Component	Mass (kg)
Steering control unit (SCU)	0.9
Electric actuator / motor	2.0
Redundancy systems	1.0
Sensors	0.5
Electronic control systems	1.0
Feedback mechanism	0.1
Steering wheel	1.0
Total	**6.5**

Values are component-level estimates (see Section 5).

4. RESULTS AND DISCUSSION

The results are presented in the same order as the methodology: the static steering effort (Section 4.1), the steering-ratio schedule (Section 4.2), the closed-loop responses (Section 4.3) and the mass comparison (Section 4.4). Section 4.5 then places these results in the context of the wider benefits and costs of SbW.

4.1 Steering effort

The static analysis shows that, with a friction coefficient of 1.1, the front wheel must overcome a resisting torque of 75.5 N·m about the steering axis. This requires a tie-rod force of 629.5 N, a pinion torque of 46.0 N·m and a rim effort of 153.2 N at a 300 mm steering wheel radius. This is a conservative estimate for a stationary vehicle on a high-friction surface without power assistance. It indicates the load the SbW road-wheel actuator must be sized to deliver, and the load that the feedback motor must be able to represent to the driver.

These values are the load specified for the actuator in the closed-loop model of Section 3.2; the relation between them and the closed-loop responses is discussed in Section 4.3.

Because the quantities in Table 1 are assumed rather than measured, the sensitivity of the result was examined. From Eqs. (1)–(6), the rim effort is proportional to the friction coefficient, the kerb mass and the scrub radius, and inversely proportional to the steering-arm length and the steering-wheel radius, so a given percentage change in any one of these parameters changes the effort by the same percentage. Table 4 gives the steering-wheel torque and rim effort for a range of friction coefficients and scrub radii, chosen here for illustration. The rim effort varies from 48.7 N to 229.8 N across the table, so the sizing of the road-wheel actuator and of the feedback motor should be confirmed once the friction coefficient and scrub radius of a specific vehicle are known.

Table 4 Steering-wheel torque T_r (N·m) / rim effort P_e (N) for different friction coefficients μ and scrub radii r_s

μ (–)	r_s = 10 mm	r_s = 20 mm	r_s = 30 mm
0.7	14.6 / 48.7	29.2 / 97.5	43.9 / 146.2
0.9	18.8 / 62.7	37.6 / 125.3	56.4 / 188.0
1.1	23.0 / 76.6	**46.0 / 153.2**	68.9 / 229.8

The base case of Table 1 (μ = 1.1, r_s = 20 mm) is shown in bold. All other parameters are unchanged.

4.2 Steering ratio versus vehicle speed

Figure 4 shows the simulated steering-ratio schedule defined in Section 3.2: the ratio stays at 20:1 up to 10 km/h and reduces to 15:1 at 50 km/h. Because the ratio is a lookup-table entry in software, changing the schedule, for example to a different curve for each driving mode, requires no hardware modification; this is the main advantage of the SbW approach to variable-ratio steering. The effect of the ratio and the load on the actuator loop is examined next.

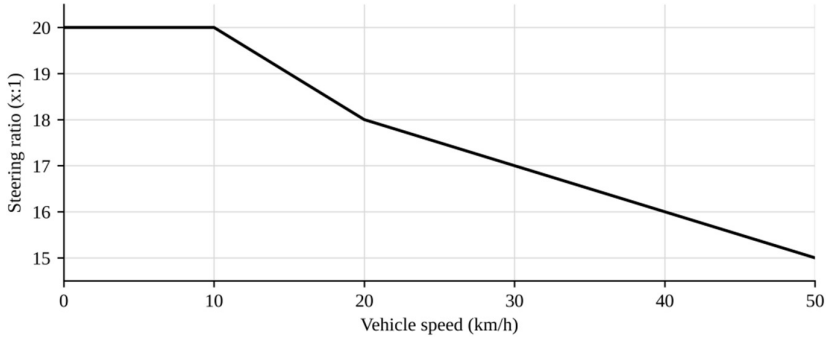


Fig. 4 Steering ratio versus vehicle speed (0–50 km/h).

4.3 Road-wheel torque and turning effort

The closed-loop responses are shown in Figs. 5 and 6. Figure 5 shows the simulated road-wheel torque over a 5 s run. It starts at about 300 N·m, rises rapidly during the first half-second and then increases almost linearly to approximately 2700 N·m at 5 s. Figure 6 shows the turning effort, which rises smoothly and with increasing slope from zero to about 1900 N at 5 s. Both responses are smooth and monotonic under the simulated input profile, with no oscillation or overshoot, which suggests well-damped behaviour of the PID-controlled actuator loop for the tested scenario; no formal stability or robustness analysis was performed.

The magnitudes in Figs. 5 and 6 are considerably larger than the static values of Section 4.1 (75.5 N·m and 153.2 N) and have not been reconciled with them in this study. The closed-loop results are therefore interpreted qualitatively, as evidence of the smoothness and damping of the loop, and not as a quantitative prediction of the steering load. The implications of such a system for steering-assembly mass are considered next.

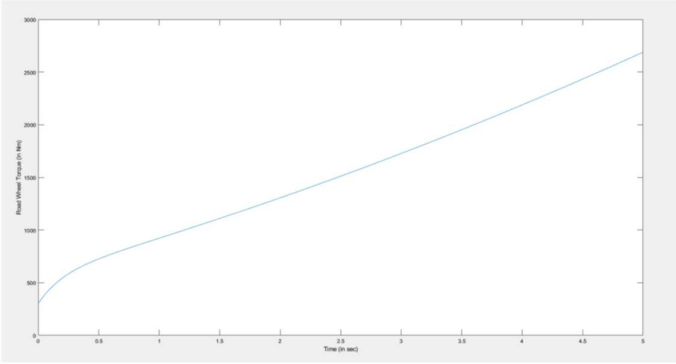


Fig. 5 Road-wheel torque (N·m) versus time (s).

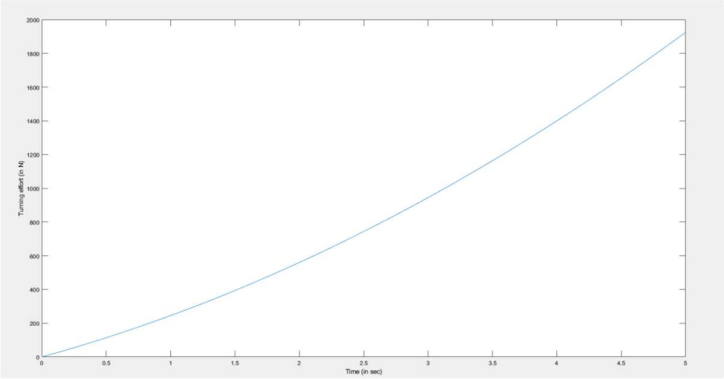


Fig. 6 Turning effort (N) versus time (s).

4.4 Weight reduction

Turning to the weight comparison, the conventional assembly has a mass of 14.0 kg (Table 2) and the SbW assembly 6.5 kg (Table 3), a saving of 7.5 kg or 53.6%. Most of the saving comes from eliminating the gearbox and the hydraulic circuit components (pump, reservoir, hoses). The saving is 0.54% of the kerb mass. If the energy consumption of the vehicle fell in direct proportion to its mass, the range would increase by the same 0.54%. This is an upper bound, because aerodynamic and auxiliary loads do not depend on mass and the actual sensitivity depends on the powertrain and the driving cycle [16]. For an independently tested real-world range of 201–216 km [17] the bound is about 1.1–1.2 km, and for the certified range of 312 km [17] it is about 1.7 km. The range gain is therefore negligible, below about 2 km even under this bound. The mass saving therefore cannot by itself justify SbW; the case rests on the functional benefits: software-defined steering ratio, packaging freedom, easier integration of driver-assistance and automated-driving functions, and fewer mechanical parts to maintain.

4.5 Expected benefits and return on investment

Because mass alone does not justify SbW, the wider benefits and costs have to be weighed. The literature and industry experience [1, 3–6] suggest the following potential benefits, which this study does not quantify: more freedom in vehicle and interior design; personalised steering characteristics; lower mechanical maintenance; easier integration with advanced driver-assistance systems; and energy use that follows demand because the actuators are powered only when steering. Any return on investment would have to be set against the cost of the redundant sensors, controllers and actuators needed for functional safety [4].

5. LIMITATIONS AND FUTURE WORK

The results above should be read within the limits of the model. The main limitations, each of which suggests a direction for future work, are as follows.

- The static analysis assumes a stationary vehicle, a single friction coefficient and a single front wheel; dynamic tyre forces, caster and kingpin geometry and steering-system friction are not modelled.
- The closed-loop simulation covers a single 5 s scenario with a ramp-type input and a speed range of 0–50 km/h. Higher speeds, step and sinusoidal steering inputs and different driving modes should be tested.
- The absolute magnitudes of the closed-loop torque and effort (about 2700 N·m and 1900 N) are larger than the static estimate of Section 4.1 and have not been reconciled with it. The closed-loop results are therefore interpreted qualitatively, and the load scaling in the model should be verified before quantitative conclusions are drawn.
- The steering-ratio schedule is illustrative and should be validated against driver-feel criteria and, ideally, in a driver-in-the-loop simulator.
- No formal stability or robustness analysis (for example gain and phase margins, or parameter uncertainty) was carried out; the statement on damping in Section 4.3 rests on the simulated responses only.
- The mass data are component-level estimates for one vehicle and should be confirmed against supplier data; the range estimate is an upper bound that assumes energy use scales in direct proportion to mass.
- The model contains no fault injection. Since the loss of a mechanical backup is the central safety issue of SbW, future work should include redundancy management, fault detection and fail-operational control [1, 4], together with a more detailed plant that includes the relevant degrees of freedom and nonlinearities [7]. Robust nonlinear position-tracking controllers for the motor drive, such as backstepping designs [18], should also be compared with the PID controller used here.
- Hardware-in-the-loop testing would be the logical step towards a physical prototype.

6. CONCLUSION

This paper developed a MATLAB/Simulink model of a steer-by-wire system with a speed-dependent variable steering ratio and used it to address three objectives. First, a static analysis for a 1400 kg electric SUV gave a worst-case steering-wheel torque of 46.0 N·m (153.2 N at the rim); because these values scale in proportion to the friction coefficient and scrub radius, they should be confirmed for a specific vehicle. Second, the closed-loop model, comprising a PID controller, an electric road-wheel actuator and a lookup-table steering-ratio schedule, produced smooth and well-damped torque and effort responses in the tested scenario, although their absolute magnitudes have not been reconciled with the static estimate and are therefore interpreted qualitatively. Third, replacing the conventional power-steering assembly with an SbW assembly saves an estimated 7.5 kg (53.6%), but even if energy use scaled in direct proportion to mass the range gain would be below about 2 km, so the justification for SbW lies in adaptability, packaging and integration with automated functions. Validation over a wider range of scenarios, fault conditions and hardware-in-the-loop testing are the main steps remaining.

DECLARATIONS

Funding The authors received no specific funding for this work.

Competing interests The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Author contributions All authors contributed to the conception of the study, the development of the simulation model, the analysis of the results and the preparation of the manuscript. All authors read and approved the final manuscript.

Data availability The Simulink model and the simulation data generated in this study are available from the corresponding author on reasonable request. The vehicle data used are from the public sources cited in the reference list.

REFERENCES

1. Eckstein L, Hesse L, Klein M (2014) Steer-by-wire, potential and challenges. In: Crolla D, Foster DE, Kobayashi T, Vaughan N (eds) Encyclopedia of Automotive

Engineering. John Wiley & Sons, Chichester. https://doi.org/10.1002/9781118354179.auto010

2. Huang P-S, Pruckner A (2017) Steer by wire. In: Harrer M, Pfeffer P (eds) Steering Handbook. Springer, Cham, Chapter 18. https://doi.org/10.1007/978-3-319-05449-0_18
3. Mortazavizadeh SA, Ghaderi A, Ebrahimi M, Hajian M (2020) Recent developments in the vehicle steer-by-wire system. IEEE Trans Transp Electrif 6(3):1226–1235. https://doi.org/10.1109/TTE.2020.3004694
4. Eckstein L (2016) Future trends for automotive steering systems. JTEKT Eng J Engl Ed 1013E:2–7. https://www.jtekt.co.jp/e/engineering-journal/assets/1013/1013e_02.pdf
5. Parsania P, Saradava K (2012) Drive-by-wire systems in automobiles. J Syst Comput 6:1–5
6. Goyal A, Thakur A (2019) An overview of drive by wire technology for automobiles. In: 2019 International Conference on Automation, Computational and Technology Management (ICACTM). IEEE, pp 108–110. https://doi.org/10.1109/ICACTM.2019.8776712
7. Irmer M, Degen R, Nübgen A, Thomas K, Henrichfreise H, Ruschitzka M (2023) Development and analysis of a detail model for steer-by-wire systems. IEEE Access 11:7229–7236. https://doi.org/10.1109/ACCESS.2023.3238107
8. Im J-S, Ozaki F, Matsunaga M, Kawaji S (2007) Design of steer-by-wire system with bilateral control method using disturbance observer. In: 2007 IEEE/ASME International Conference on Advanced Intelligent Mechatronics. IEEE. https://doi.org/10.1109/AIM.2007.4412494
9. Wu Y, Wang L, Li F (2018) Research on variable steering ratio control strategy of steer-by-wire system. SAE Technical Paper 2018-01-1583. https://doi.org/10.4271/2018-01-1583
10. Zhang P, Tian J, Hu J, Wu D (2024) Design of variable steering ratio for steer-by-wire system considering the driver's handling skill. IEEE Access 12:110599–110609. https://doi.org/10.1109/ACCESS.2024.3439697
11. The MathWorks Inc. MATLAB and Simulink. The MathWorks Inc., Natick, MA. https://www.mathworks.com

12. Robert Bosch GmbH (ed) (2014) Bosch Automotive Electrics and Automotive Electronics: Systems and Components, Networking and Hybrid Drive, 5th edn. Springer Vieweg, Wiesbaden. https://doi.org/10.1007/978-3-658-01784-2
13. Gillespie TD (1992) Fundamentals of Vehicle Dynamics. SAE International, Warrendale, PA
14. Shah S (2020) Tata Nexon EV review, test drive. Autocar India, 21 January 2020. https://www.autocarindia.com/car-reviews/tata-nexon-ev-review-test-drive-415633. Accessed 11 October 2026
15. Rajamani R (2012) Vehicle Dynamics and Control, 2nd edn. Springer, New York. https://doi.org/10.1007/978-1-4614-1433-9
16. Pagerit S, Sharer P, Rousseau A (2006) Fuel economy sensitivity to vehicle mass for advanced vehicle powertrains. SAE Technical Paper 2006-01-0665. https://doi.org/10.4271/2006-01-0665
17. Shah S (n.d.) Tata Nexon EV real world range tested, explained. Autocar India. https://www.autocarindia.com/car-news-amp/tata-nexon-ev-real-world-range-tested-explained-420958. Accessed 11 October 2026
18. Linares-Flores J, García-Rodríguez C, Sira-Ramírez H, Ramírez-Cárdenas OD (2015) Robust backstepping tracking controller for low-speed PMSM positioning system: design, analysis, and implementation. IEEE Trans Ind Inform 11(5):1130–1141