An Approach for Steering Advancement in Motorcycle Riding Simulation

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Abstract. This paper presents the development of a steering sensor system for use in motorcycle riding simulation based on the forces acting on the handlebar. An overview of motorcycle dynamics models for riding simulation is also given. In the section Related Work, different approaches to steering a motorcycle simulator are evaluated and divided into force-based and angle-based. Based on this, a recommendation is given on how to measure and evaluate steering interaction separately on the left/right handlebar stubs. The development is described with the following subsections steps: technical development, software development and determination of the steering forces/torques based on a calibration test bench. This is followed by a validation method of the sensors and an evaluation of the ridability using this handlebar. Finally, the steering angle, the steering torque and the roll angle are compared with the previously described motorcycle riding dynamics. The evaluation of the interaction (rider control behavior) during a lane change is presented based on the applied forces on the handlebar.

Keywords: Motorcycle steering · Riding simulation · Force based steering · Isometric · Strain gauges · Rider steering interaction · Counter steering

INTRODUCTION

Driving simulation has a long history and is used for different purposes. The first driving simulator for cars was introduced in 1972 by the Highway Safety Research Institute (Campbell, 1972). Since then, there has been continuous development of driving simulators for entertainment, training, research and development (Fisher et al. 2011). The automotive industry has focused on the evaluation of Human Machine Interface (HMI) for the approval of assistance systems (Maag et al. 2012). Recently (2018), driving simulators have been used to evaluate chassis concepts and for vehicle dynamics testing (Brems, 2018). It can be deduced, that car driving simulators have reached a level of development with high degree of validity in the mapping of driving dynamic conditions. In motorbike driving simulation, the state of the art is comparable to the middle development phase of driving simulators. These simulators are used exclusively for HMI concepts and are often criticized for their ridability (Stedmon et al. 2011) and (Werle et al. 2021). The development setback could be caused by the lack of motivation for further development due to the lower number of applicants such as researchers or engineers. Popov et al. (Pless et al. 2016) summarized that high-quality motorcycle dynamic representation is now widely understood and available. It is important to build realistic rider simulators to study rider control behavior and understand how the rider interprets and uses the information to perform the steering control task. In this paper, we summarize the state of the art in motorcycle riding simulation and describe a method on how to develop a force-based rider steering system for riding simulators. The results and conclusion are shown and discussed for further research.

Related Work: Motorcycle riding dynamics have already been researched for many years with various approaches. The first real time capable models were used to simulate simple riding dynamics such as roll-, pitch- and yaw rotation. Later models were used for the evaluation of stability control and phenomena such as wobble or weave (Sharp, 2001). (Sharp, 1974) first described the influence of the relationship between the tires and the road in 1971. This simplified model allowed study of the relationships between steering, roll- and yaw angles with a focus on stability. The main conclusion was that a proper representation of the free control characteristics requires at least an eight-order model with a valid tire model. (Koenen, 2017) investigated the dynamic behavior of a motorcycle when running forward in a straight line and cornering. He developed a nine-body motorcycle model and described the physical tire characteristics. This model made it possible to simulate the main aspects of oscillatory dynamic behavior: weave and wobble. Later (Pacejka et al. 1992) presented a mathematical-based method for an accurate description of tire forces and moments, which is the state of the art for dynamics research and simulation. Popov et al. (Pless et al. 2016) summarized the research approaches and concluded that these models should be used for riding simulation because there is sufficient accuracy. The only requirement is the real-time capability of the models that is fulfilled by the models described below. Commercial riding dynamic models are also available and based on previous work; the most cited for motorcycles are BikeSim® and BikeRealTime® (Haas et al. 2020). BikeSim® is a real time model used for various applications as riding simulation (Roe et al. 1978). The multi-body model includes several tire models such as: The Magic Formula, the FTire or the MF-Tyre, which can be chosen for the specific requirement. The multi-body model was described by (Evangelou, 2004), (Sharp et al. 1974), as well as (Sharp et al. 2004) and can be modified for higher demands as nonlinear characteristic for specific bodies or components. BikeRealTime® is a high-performance motorcycle model which can be used as a real time or non-real time application. A Matlab/Simulink co-simulation can be used during real time environment. For this application, the nonlinear model is automatically transformed into a linearized model with 11 degrees of freedom (dof) and 6 bodies (2 wheels, fork upper and lower, body and swingarm) without the rider. The tire force and torque are calculated using the magic formula (Paceijka et al. 1992). In summary, full real time motorcycle models are not available for free. Only motorcycle dynamics models are available for use in driving simulations. In this research paper, we use BikeRealTime® because of the easiness of model integration and the evaluation of (Haas et al. 2020).

Steering a motorcycle is described by riders in various ways. Some say that turning into the corner is done by the upper body or that the handlebar must be "pushed"; another differentiation based on this is: positive or counter steering (Frank et al. 2020), (Kawakoshi, 2014). Both types occur during a ride depending on the speed, vehicle state and driver steering strategy. In this paper, a measurement method is developed that allows to measure the force of a push to enable so-called counter steering riding. (Cox, 1998) discussed the physical relationship of the counter-steering phenomenon, beginning with the following sentence, quoting a professional rider: “You turn right to go left. It is the only way to turn a motorcycle.” This quote implies that steering interaction is based on a steering angle. (Cox, 1998) describes that counter steering is initiated by applying forces on the handlebar and assumes that counter steering occurs above 32 km h^{-1}. From a physical point of view, he describes the basic relationship between the application of force resulting in torque, and the rolling of a single-track vehicle.

(Cossalter et al. 2002) used a driving simulator for the investigation of riding ability and for testing riding assistant systems. His was a force-based setup and was tested for various maneuvers. He used measured data from a lane change maneuver with a speed of 52 km h^{-1} for the description of the counter steering phenomenon. In this publication, we have reproduced this simulation at a speed of 85 km h^{-1} for a lane change maneuver. Figure 1 shows the steering torque, steering angle and roll angle curves that are comparable with the literature, where criteria 1 - 4 describe how counter steering can be recognized. The subdivision of the criteria is based on (Cossalter et al. 2002).

(1) *Initiate lane change* Counter steering can start torque or angle based. In the case of the input method steering angle, torque has to be simulated with up to 200 N m s^{-1}, which is known as force feedback. For input method steering torque, the steering angle has to follow the model with a minimum of 8 °s^{-1}, which is called position feedback. At the same time, the roll angle of the vehicle begins to increase slowly. (Cossalter et al. 2002) used degrees per second for the evaluation. We have chosen degrees for better understanding.
(2) *Maximum steering angle* The maximum steering angle is reached (1.74°) in the middle of the counter steering phenomena and begins to decrease at that point. Steering torque has also reached its maximum at 39 Nm and stays there until the end of the counter steering process. The roll angle is still increasing, which represents the lane change.

(3) *Reverse steering and roll angle* At the end of counter steering, the steering angle has its zero pass with the begin of decreasing steering torque and roll stabilization.
(4) *Stabilization phase* The steering angle has a phase lead of approx. 90° to the steering torque and also starts the transient process, which indicates the stable state, without which the front wheel would tilt and the vehicle would fall.

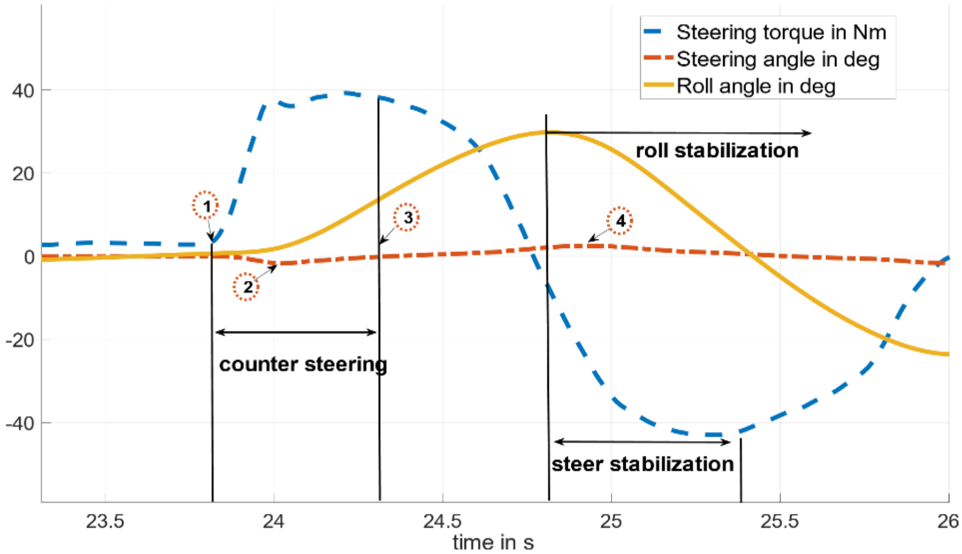

Fig. 1. Phases of the counter steering phenomenon (Cossalter, 2006)

Table 1 shows various configurations of steering a car by the steer by wire structure described by (Huang et al. 2004). Transferring these concepts into a motorcycle application, it can be seen that a distinction is made between two basic approaches. One is position-feedback and the other is force-feedback steering. An isotonic steering concept is not usable due to the missing proprioceptive feedback (Bubb, 1978). Isometric steering has not yet been investigated as part of a riding simulation. Isomorphic steering is based on both inputs and feedbacks; this approach has not been published yet. The progression of the steering angle can be separated into positive or counter steering. If valid motorcycle driving dynamics are considered, the counter steering phenomenon always occurs up to the condition of maneuvering. (Huang et al. 2004) also introduced the consideration of active and passive steering. Active steering describes that the model-steering angle or the model-steering torque is fed back to the rider. Alternatively, it is possible to use a return spring, which corresponds to passive steering.

Table 1. Comparison of steering approaches based on (Huang et al. 2004)

Input	Feedback	Input	Feedback
Position- Feedback Steering		Force-Feedback Steering	
Force	Position	Position	Force
(Pless et al. 2016), (Cossalter et al. 2006)		(Guth, 2017), (Benedetto et al. 2014)	
Isometric Steering		Isotonic Steering	
Force	None	Position	None
No publication		Not applicable	

The interaction between the rider and the vehicle depends on the configuration of the riding simulator. Therefore, only the most advanced simulators are considered. Control of the lateral position can be accomplished by steering or by shifting the center of gravity of the upper body (open loop). These two methods have completely different influences on the dynamic of the vehicle. The consideration of open loop driving is therefore discarded. Various steering methods are considered in current research and are shown in table 1 with a subsequent discussion.

Position based An angle-based approach reported by (Will et al. 2016) suggests that no sufficient riding steering concept has been achieved. The reason was a lack of ridability and intervention by controllers to stabilize the vehicle. The latest report (2020) by (Grottoli, 2021) also describes that the angle-based approach is only useable for a low speed of 30 $km/h^{-1.}$ which severely limits its usability. (Nehaouas et al. 2010) describes a method on how to estimate steering torque by measuring steering angle. Considering figure 1. this means that the rider has to replicate the steering angle trace, which has not been discussed any further in ridability.

Force based Force- based input methods have also been used in various riding simulators. (Pless et al. 2016) describe a method on how force-based steering can be used for riding simulation. In order to be able to apply a force, a reaction force must be present. This means that a motor or a mechanical fixation has to be used. Therefore, he implemented a controller that determines the torque according to the force and angle. Based on the calculated torque, movement of a steering angle is performed, which is known as impedance control (Sharp et al. 2001). The evaluation showed that ridability increased at higher speeds. The angle was also implausibly large. This can be explained by the fact that the steering angle was not used as a state variable from the dynamic model. Another force-based setup evaluated by (Westerhof, 2018) showed that impedance control is not necessary. The used configuration was a steering measurement setup with a handlebar control loader to simulate tire forces (Force-Feedback). Lane deviation was statistically evaluated and showed a potential for improvement.

The angle-based and force-based steering approaches show advantages and disadvantages for various characterizations in the configuration of steering methods for riding simulators. In summary, there is no satisfying steering method for the stable and unstable states of riding simulation. All authors narrow down the methods to the corresponding speed ranges and recommend further investigations. In the next sections, the development of a force-sensor in the handlebar stubs is explained. Additionally, we show how to calibrate the system by creation of a 2D force vector, selection of filter parameters, evaluation of the signal and validation of the measuring method. The results include ridability in a stable speed range in regard to phases of the counter steering phenomenon and validation of the sensors. The current simulator configuration consists of a steering based setup with a motorcycle dynamics model by VI-Grades BikeRealTime®. The aim is to investigate and evaluate ridability for both steering methods in different speed ranges and configurations.

METHOD

The riding simulator environment is a modular setup with real time software interface modules. For measurement and control, a Matlab Real Time Workshop is used with a separated measuring CANbus for fast signal transmission. The signal measurement is realized with fast signal measurement technology with 8 strain gauge input channels provided by Weber Electronic and Race Engineering. The sampling rate is 4 kHz with an automatic conversion to Controller Area Network (CAN) and a 1 MBaud s^{-1} Peak CAN to USB interface. This setup enables fast data acquisition and processing, which is important for motorcycle riding simulation. The chosen handlebar is a two-component structure from the BMW 1600 GT. This setup has the advantage that the stub of the handlebar shown on the left side of figure 3 is separated by the shaft and has slightly lower stiffness, thus allowing more accurate measurements. Each shaft is applied with 4 strain gauges as shown in the middle of figure 3. With this Wheatstone bridge application, it is possible to measure an elongation and compression (applied force) at the same time, which produces a more accurate signal. Figure 3 right, shows the measurement axis for EX+= F_{xsg} force strain

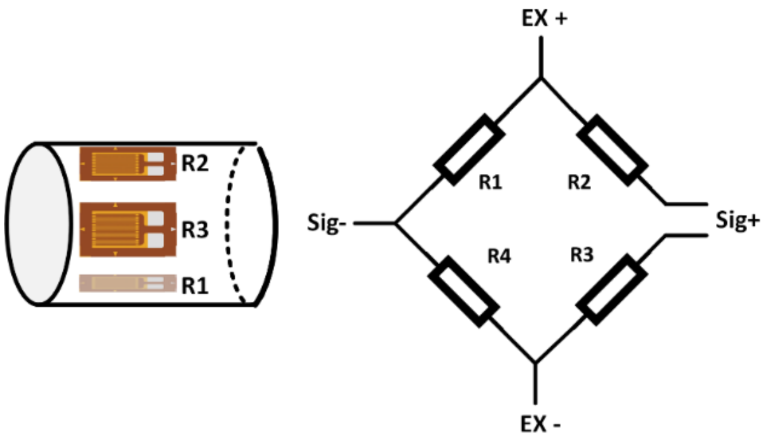

Fig. 3. Strain gauges application and wiring.

gauges' longitudinal axis and for EX− = F_{zsg} force strain gauges' vertical axis, the designation R1 to R4 refers to the positioning and wiring of the strain gauges. To resolve inaccuracies in the strain gauge application, the coordinate system of the strain gauges will be corrected.For the calibration process, we developed a test bench shown in figure 4. It allowed us to determine the linearity of the material. Since we want to infer the steering force from the bending of the handlebar, a precise understanding of the material characteristic is required. For this we rotated the load cell between 0° - 360° and applied increasing force Fc in 5 N steps up to 50 N. Also, it was possible to evaluate the application and adjust the strain gauge coordinate system shown in figure 3. Due to the tilt of the stubs, we implemented a secondary frame to apply orthogonal forces Fc for the calibration process shown in figure 4 as a dashed orange line. After determining the strain gauge coordinate system axes F_{zsg} and F_{xsg} relative to the inertial coordinate systems F_{zi} and $F_{xi,}$ it was possible to align them. We used this method for both stub handlebars; this made it possible to have a parallel coordinate system which is important for calculating the steering force Fs from the total force Ft. The red box in figure 4 symbolizes the calibration load cell. For the calculation of the steer- and prop up torque we determined the lengths L_x and L_z for each axis from the pivot see figure 5.

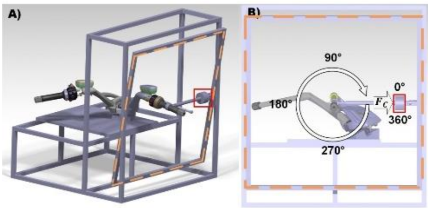

Fig. 4. Calibration test bench.

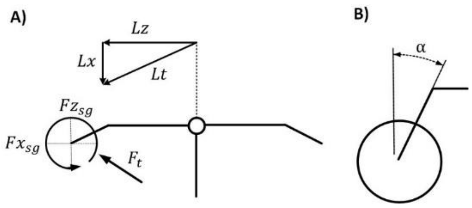

Fig. 5. Torque components.

Based on this consideration, it is possible to calculate the total force with the measured forces F_z and F_x. The distribution of F_s steering and F_p prop up forces depends on the angle α, which represents the steering head angle, see figure 5 B. Subsequently, the division into the M_s steering and M_p prop up torque is performed. A sign convention for determining the total torque was used as follows: left stub +M and right stub -M in reference to [5]. The total steering torque can be calculated by an addition of both terms.

RESULTS

This chapter shows the results. First, we show that the metrological setup works for this we describe the strain gauge coordinate system and the acquisition and subdivision of the forces. The last part is a validation run for pushing, pulling, and double lane change. There we describe ridability in an objective way by comparing the curves: steering angle, steering torque and roll angle in accordance with the introduced four criteria. The validation of the measurement system was done with calibrated weights: 5 N, 10 N, 20 N and 50 N. We attached them to the place of calibration in the middle and the outer point of the stub (see figure 4). For the middle position, we measured an average deviation of 2.0%, and 25% for the outermost point. The validity measurement was carried out for both axes and stubs. For the validation of the rotational measurement we used a validation system that applied a defined force between 0° - 360°. The evaluation showed that the force axis alignments F_{zsg} and F_{xsg} are orthogonal to each other. This allows the calculation of the partial torque, total torque and the determination of the angle applied to the stubs. The following qualitative assessment shows the results of the implemented measuring system in accordance to the evaluation of the forces, angle computation and determination of the steering torque for the right and for the left stub. Figure 7 A) shows the angle progression between 90° and 270° based on the described calibration method on the test bench. Figure 7 B) shows the orthogonality of the coordinate system, as the cosine (Z axis) and sine (X axis) components are shifted by 90°. Due to the fact that the previous calibration describes accuracy, no rotatory measuring instrument was used for this measurement. Figure C) shows the course of the total force over time, whereby we applied a force by hand between 35 N and 50 N. Figure D) shows the calculation of the steering torque of the right stub with a steering head angle of 26.5°. The change of signs results from the direction definition via the steering axis. A positive steering torque causes the direction of movement to the right (pulling on the stub), which is recognizable by the transition from left to right rotation of the steering axis. Overall, it can be seen that the sensor can determine the forces and the additional information such as the course or the determination of the angle in the total force. This makes it possible to carry out studies on the interaction between the driver and the mock-up handlebars or by real ride investigation. For the evaluation of ridability, we set up the steering handlebar on the mock-up with a fixed steering axle. The fixation of the axis was needed to measure the torque. For the evaluation of ridability. we tested speed ranges between 70 km h^{-1} and 120 km h^{-1} with force-based steering and compared it to the described phases of counter steering in chapter 1 referencing (Cossalter et al. 2006). The double lane change was created in accordance with ISO 3888-2, but to decrease the level of stress, we increased the distances for better ridability. Finally, the cones (marked as black circles) had a distance of 140 m and a width of 2.5 m shown in figure 8. The riders were able to perform lane changes without touching the cones once. It can be seen in figure 8 that the counter steering phenomenon occurs, due to the progressions of the steering torque, the steering angle and roll angle, which means that the torque measurement works. The increasing curves fulfill the criteria 1, 2, 3 and 4, by which it is possible to recognize the counter steering phenomenon described in chapter 1 Initiation of lane change, stabilization of the roll angle and the final steering stabilization for riding straight ahead can be seen. The second part of the lane change shows the same progression except for the right side. The trajectory figure shows the driven trajectory and the occurred roll angle. Comparing this with the literature [5], similarity is recognizable for torque, roll and steering angle. Figure 9 shows an evaluation for the upper scenario in figure 8. For clarity, the criteria 1-4 were added and marked with a dotted circle. Also, it is to be noted that the axes of the force's distribution left and right of the handlebar are already signed. The evaluation refers to lane changes and is divided into the areas of Push, Push support and Push balancing.

Idle and preparation phase Between 25.5 s and 26 s a stabilization of the vehicle is recognizable (1) (Push balancing) by applying the force alternately. Between 26 s and 26.5 s, preparation of the lane change is noticeable by pushing the right handlebar (2) (Push) and decreasing the force on the left handlebar. **Counter steering phase** Between 26.5 s and 27 s, counter steering behavior can be recognized by an increase in the left handlebar stub (steer force left) and a decrease in the right handlebar stub (steer force right). For the right side, it can thus be assumed that the right side supports (3) (Push support) the left steering behavior. **Lane change phase** Between 27 s and 28 s, the lane change is performed by applying a force on the left handlebar stub (3) (Push), and at the same time a supporting force on the right side (3) (Push support). After the lane change at 28 ms, balancing steering (3) (Push balancing) begins and then the vehicle aligns with a roll angle of +- 5°.

CONCLUSION

The described method with the aim to develop a force and angle-based measuring and control system for the application in riding simulation shows a valid approach for the torque stub measurement. Mechanical application of the strain gauges to span a force plane across the grips allowed the measurement of the total force and determination of the force as a function of the angle. Also, the separation of the measurements in the left and right stubs enables new research in the context of understanding of the interaction

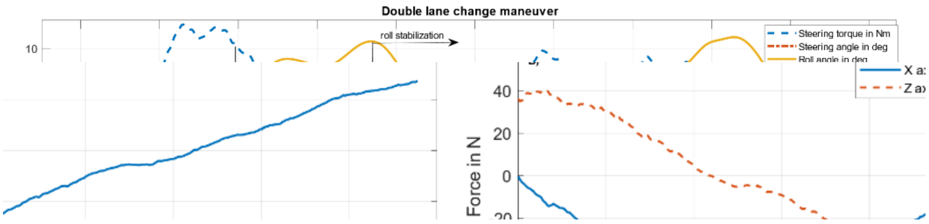

Figure 7. Sensor force and angle validation

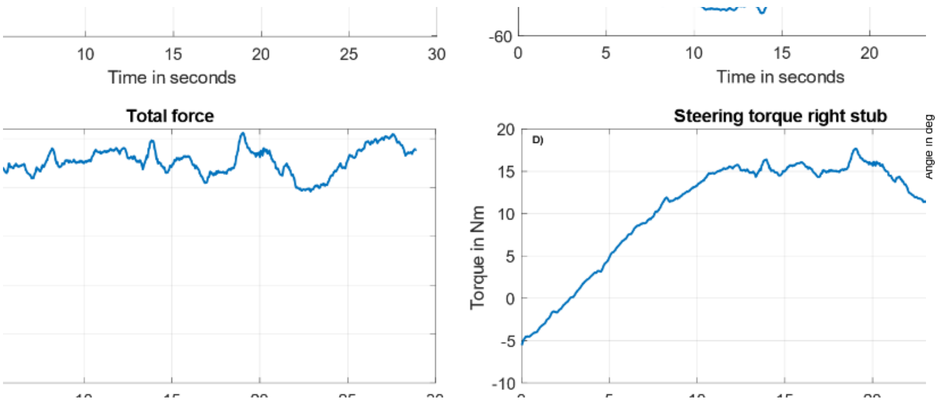

Fig. 8. Lane change maneuver

of motorcycle steering, as already suggested by (Popov et al. 2010). In terms of measurement accuracy, we found that this approach is sufficient, but reaches its limits as soon as mechanical changes are made to the handle. For our purpose, calibration of the offset had to be performed several times. Another advantage of this measurement setup results from the independence of the mock-up. Since we use a fixed plane as the basis, it is possible to calculate the steering head angle stored in the vehicle dynamics model for the steering force. The model would otherwise have had to be adjusted with the mock-up. One of the most important steps has proven to be the calibration and deriva-

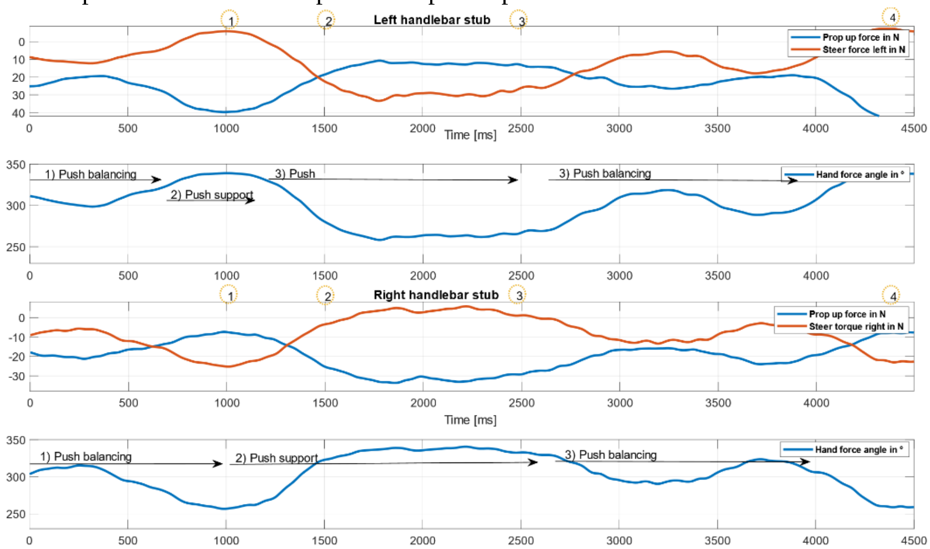

Fig. 9. Steer evaluation.

tion of the force map. With this, we obtained the adjustment if the material behaved linearly as a function of the angle. It was also possible to check the linearity of the force. In addition, we have calibrated weights to show that the calibration test bench has sufficient accuracy. The chosen handlebar stubs had a high stiffness with a 15 mm material thickness that should be considered for future developments. For the evaluation of ridability, we conducted an exploratory investigation, which showed that this approach is target-oriented and shows qualitative good ridability by various riders. Due to the depth of content of the technical setup, we refrained from a detailed subject study. The used model shows sufficient controllability, whereby a detailed investigation should be accomplished for the fulfillment of the driving task - which will be evaluated in the future. Further investigation should be performed on the pressing and pushing rates on each stub, which is the main benefit of the separated measurement. Another fact is the fixed steering handlebar that should be investigated if a steering angle is necessary as feedback for the rider. This should be done with comparison with other steering configurations. By measuring cognitive load, handling the vehicle in standard as well as limited situations and evaluation of this setup is suitable for the investigation of systems.

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