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Design and implementation of a steer-by-wire bicycle

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ABSTRACT

Since the 1800s, the design of bicycles involves a mechanical linkage between the handlebar and the fork assembly. Herein, we propose an innovation, where the traditional mechanical connection between the handlebar and fork is decoupled and replaced with sensors, servomotors and a microcontroller allowing artificial manipulation of the bicycle and steering dynamics. The purpose of our steer-by-wire bicycle is to investigate the influence of handlebar torque feedback on rider control in order to understand rider control on a bicycle. In addition, steer-by-wire bicycles have the potential to be used as stability-enhancing support systems which can improve cycling safety. We demonstrate the design and performance of the steer-by-wire bicycle in computer simulations as well as real-life tests. Preliminary rider tests showed a perceived near-to-identical behaviour of the steer-by-wire system to a mechanical connection at steering frequencies below 3 Hz.

Keywords: bicycle, steer-by-wire, stability, control, rider control.

1 INTRODUCTION

Already some time ago, electronic enhancements regarding vehicle behaviour has made its way into the aviation and automotive industry by the term ”by-wire” technology. This covers technology like fly-by-wire, drive-by-wire, brake-by-wire and steer-by-wire. Electronic sensors and actuators are used to replace traditional mechanical systems, and software running on a controller is used to operate the actuators in a way that it is not possible with traditional mechanical systems. The use of steer-by-wire technology can also offer great opportunities to enhance the vehicle dynamics of single-track vehicles like motorcycles, scooters and bicycles. Single-track vehicles can be laterally unstable, especially at low forward speeds and they require a relative high amount of rider control [1], [2].

In the open literature there is currently no research available which experimentally evaluates a steer-by-wire system on single-track vehicles. Only a few theoretical publications proposing enhancements in motorcycle handling [3], [4] are available. Marumo and Nagai [3] introduce a steer-by-wire system on a motorcycle which removes the counter steer behaviour to initiate a turn, where it remains questionable if this is beneficial. On the other hand, the possibility of a lane keeping assistance system on motorcycles by Katagiri et al. [4] can greatly improve safety. This is also demonstrated by Seiniger et al. [5] by actively assisting the motorcycle rider's steer input to hold its driving path during extensive in-corner braking manoeuvres. Schwab et al. [6] were the first who actually investigated in practice the impact of active steer-torque control on the lateral stability of a bicycle. Their results showed a considerably lower rider steer effort and increased stability at low forward speeds.

Alternatively, steer-by-wire technology can also serve as a versatile experimental platform for identifying rider control in bicycling. Still, the question remains how the rider stabilizes the lateral motions of the bicycle when driven at low (unstable) forward speeds and how the rider follows a desired path. The control input probably comprises of haptic, vestibular and visual cues; however the impact of these sensory cues on rider control is still unknown. With a steer-by-wire system we can investigate the importance of haptic feedback on steering behaviour for a given set of control tasks. For instance, the task of bicycle stabilization with and without handlebar feedback can be investigated, which is not possible with a mechanical steered bicycle. The evaluation of handlebar torque feedback on rider control might lead to the development of new design criteria for safer bicycles

The work presented here, is focused on the modelling and experimental validation of a steer-by-wire system on a bicycle. Such a system can be either used as a research tool to investigate rider behaviour or as a platform to test and implement support systems enhancing bicycle balance and handling. The paper is organized as follows. After this brief introduction, the model for the system design is described and simulation results are shown. Next, the experimental setup is described and preliminary test results are shown. The paper ends with conclusions.

2 SYSTEM DESIGN AND SIMULATION

A linear multibody model for the bicycle was used to design the system. The model is based on the three-degree-of-freedom Whipple/Carvallo bicycle model [1]. This model is extended by separating the handlebar assembly from the fork assembly, which introduces an additional rotational degree of freedom, see Figure 1.

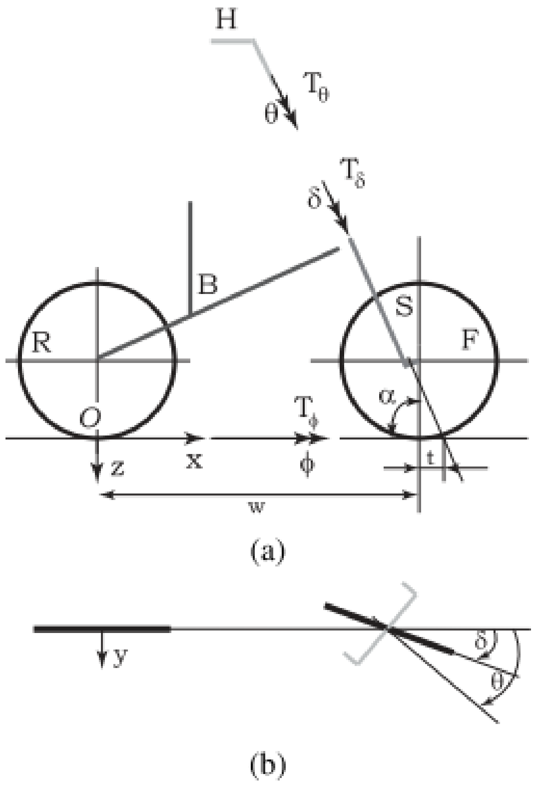


Figure 1. Steer-by-wire bicycle model, side view (a) and top view (b), together with the lateral degrees of freedom, rear frame roll angle ϕ, fork angle δ, and handlebar steering angle θ, and some geometry variables. This model, based on the Whipple/Carvallo bicycle model [1], shows the addition of a separate handlebar body H and the possibility to have unequal fork angle δ and handlebar angle θ and torques T_δ and T_θ.

The lateral degrees-of-freedom of this extended model are: the rear frame roll angle ϕ the fork angle δ, and the handlebar steering angle θ. Since we are only interested in the lateral dynamics, the forward speed v, which is a degree-of-freedom of the Whipple model [1], is treated as a parameter. Combining the lateral degrees-of-freedom in a generalized coordinate vector $\boldsymbol{q} = [\theta, \phi, \delta,]^T$, the linearized equations of motion for the extended bicycle model can be expressed by,

$$\bar{\boldsymbol{M}}\ddot{\boldsymbol{q}} + \bar{\boldsymbol{C}}\dot{\boldsymbol{q}} + \bar{\boldsymbol{K}}\boldsymbol{q} = \bar{\boldsymbol{f}}, \tag{1}$$

with the mass matrix $\bar{\boldsymbol{M}}$, damping matrix $\bar{\boldsymbol{C}}$ and stiffness matrix $\bar{\boldsymbol{K}}$ given by,

$$\bar{\boldsymbol{M}} = \begin{bmatrix} \boldsymbol{I_\theta} & 0 \\ 0 & \boldsymbol{M} \end{bmatrix}, \bar{\boldsymbol{C}} = \begin{bmatrix} 0 & 0 \\ 0 & v\boldsymbol{C1} \end{bmatrix}, \bar{\boldsymbol{K}} = \begin{bmatrix} 0 & 0 \\ 0 & g\boldsymbol{K0} + v^2\boldsymbol{K2} \end{bmatrix}, \tag{2}$$

and the right hand side forcing term $\bar{\boldsymbol{f}} = \left[T_\theta, T_\phi, T_\delta\right]^T$, which contains the handlebar torque T_θ, the external rear frame roll torque T_φ (usually zero), and the fork torque T_δ. The matrices $\boldsymbol{M}$, $\boldsymbol{C1}$, $\boldsymbol{K0}$ and $\boldsymbol{K2}$, are the two-by-two matrices from the linearized equations of motion of the original Whipple/Carvallo model [1], I_θ is the mass moment of inertia of the handlebar assembly, v is forward speed and g is the gravitational acceleration. There is a small coupling between handlebar steering angle and the other degrees of freedom, but given the already small value for the mass and inertia of the handlebar assembly H, the off-diagonal terms in the matrices (2) are neglected here.

As a reference case, it will be desirable to simulate a direct connection between the handlebar and the fork. To minimize the difference between the handlebar angle θ and the fork angle δ, tracking control has been implemented. In this way, the steer-by-wire system should behave like an ordinary, mechanically steered bicycle, when the rider applies a steer torque at the handlebar. Two proportional-differential PD-controllers are implemented in order to provide an action-reaction torque T_{PDH} to the handlebar and T_{PDF} to the fork assembly. Angular velocity $\dot{\theta}$ and $\dot{\delta}$ are estimated by taking the time derivative of angular position θ and δ respectively, for a fixed time interval of 1 ms. The double PD-configuration can also be used to manipulate the steer feedback torque independent of the tracking performance. The double PD-controller is of the following form,

$$T_{PDH} = K_{PH}(\theta - \delta) + K_{DH}(\dot{\theta} - \dot{\delta}), \tag{3}$$

$$T_{PDF} = K_{PF}(\theta - \delta) + K_{DF}(\dot{\theta} - \dot{\delta}), \tag{4}$$

with proportional gains K_{PH}, K_{PF} and differential gains K_{DH}, K_{DF} respectively. The torque T_{PDH} is applied at the upper motor, and the torque T_{PDF} at the lower motor (thus ignoring motor dynamics). The forcing term in equation (1) then becomes,

$$\bar{\boldsymbol{f}} = \begin{bmatrix} T_\theta \\ T_\phi \\ T_\delta \end{bmatrix} = \begin{bmatrix} T_h - T_{PDH} \\ 0 \\ T_{PDF} \end{bmatrix}, \tag{5}$$

with the rider applied steer torque T_h at the handlebar, and zero applied roll angle torque. Ideally the two controller torques could be identical, however we have the freedom to choose the gains K_{PH} *and* K_{PF} and K_{DH} and K_{DF} differently. From the outside a rider perceives only the in series combined stiffness and damping. The double PD-controller configuration combined with steer-by-wire bicycle plant model can be visualized in a block diagram as shown in Figure 2. For the PD-controllers, the proportional and differential gains in equation (3) and (4) are chosen such that a critically damped system response is obtained to ensure a fast and accurate response without overshoot.

As an example, we use the parameters of the benchmark bicycle [1], with a handlebar inertia of I_θ=0.001 kgm^2, proportional gain K_P=90 Nm/rad, and differential gain K_d=0.6 Nms/rad. As a first measure of performance the tracking transfer function of the steer-by-wire system, defined as $HSBWe(s)=\delta(s)/\theta(s)$ is analysed. The tracking transfer function magnitude shows the angular error between the handlebar and the fork whereas the phase describes the latency in the system response.

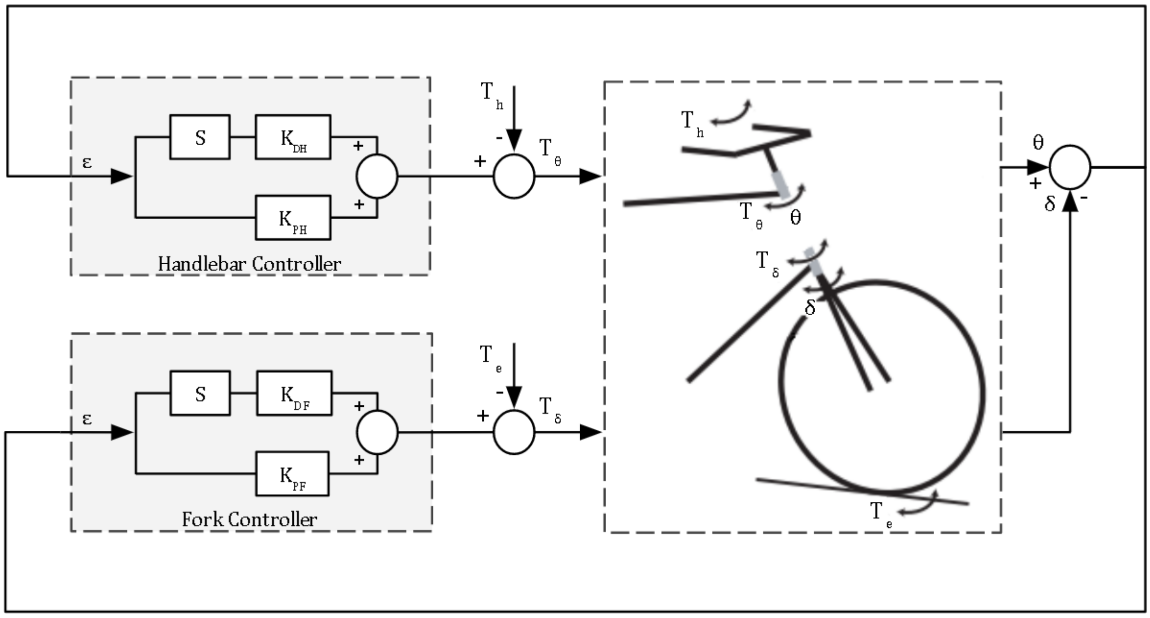


Figure 2. Block diagram of the steer-by-wire bicycle model, which includes a double PD-configuration, with rider applied steering torque T_h, handlebar steering feedback torque T_θ, handlebar steering angle θ, fork applied torque T_δ, ground reaction T_e and fork angle δ.

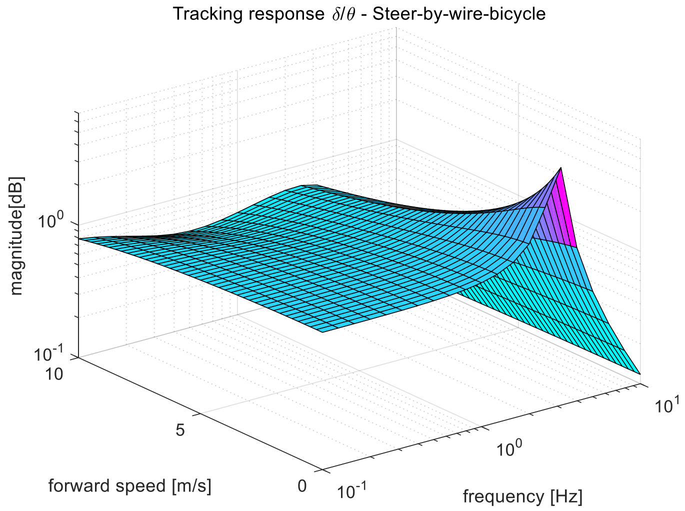
In Figure 3 the tracking transfer function magnitude (a) and phase (b) are presented as a function of input frequency and forward speed. In general the gain is close to one and phase is close to zero indicating a behaviour approximating a mechanical connection.

At a steering frequency of 3.1 Hz and a bicycle speed below the weave speed (4.29 m/s), a substantial increase in tracking magnitude occurs (as shown by the resonance-like upward peak at 0 m/s). A slight tracking magnitude decrease occurs above 7 m/s at an input steering frequency range of 0.2-0.8 Hz. The increase in the tracking magnitude near the resonance-like peak makes the bicycle hard to control, but this occurs only at very low speeds combined with steering input above 2.5 Hz.

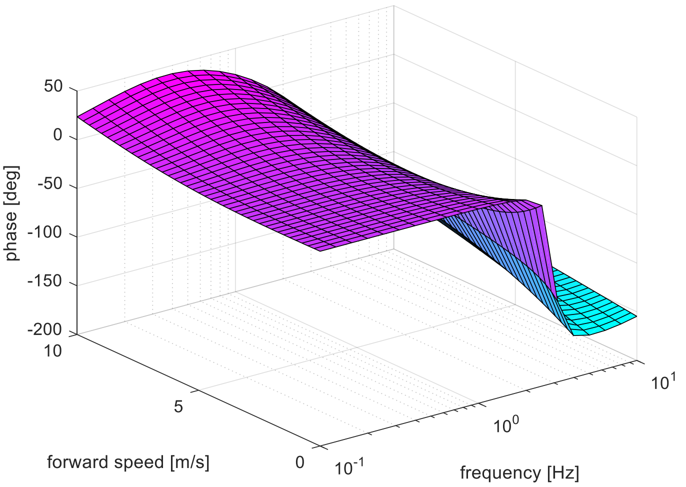
Phase lead is noticed for an input steering frequency of 0-1.2 Hz and an input speed above 7 m/s. On the other hand, the phase lag fluctuates between 0-20 deg below 7 m/s up to 2.5 Hz. Above 2.5 Hz the phase lag decreases for all speed ranges with a slope of approximate -86 deg/Hz until it reaches a plateau of about -150 deg above 4 Hz. The phase lag decrease above 2.5 Hz indicates a large latency in the system response. The fork can no longer follow the handlebar commanded steer angle within a given time threshold. The increased time delay above certain frequencies might increase the steer effort and lead to loss of control in case the phase lag becomes too large.

The resonance-like peak in the tracking magnitude at 3.1 Hz and the increase in phase lag above 2.5 Hz is due to the finite stiffness of the tracking controller, whereas the slight tracking magnitude decrease and phase lead at forward speeds above 7 m/s is due to the velocity-dependent properties of the stiffness matrix $K2$. In summary, the simulated controller shows good tracking performance in a frequency range of 0-2.5 Hz and in a speed range of 0-10 m/s;

above this frequency range, the tracking magnitude and phase lag significantly increase, especially at low forward speeds.



(a)

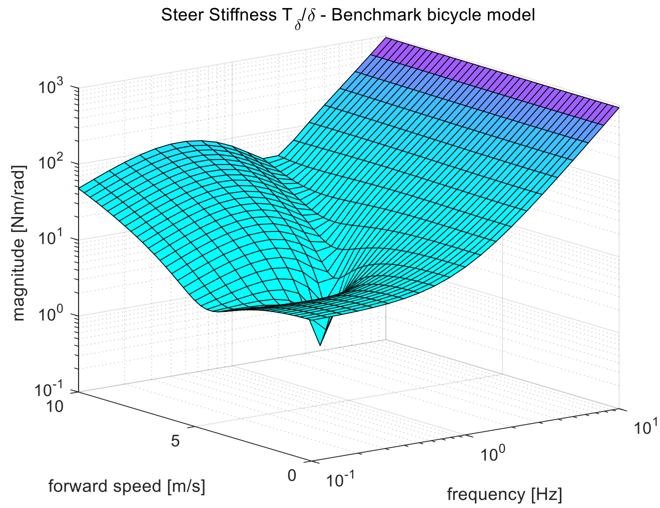


(b)

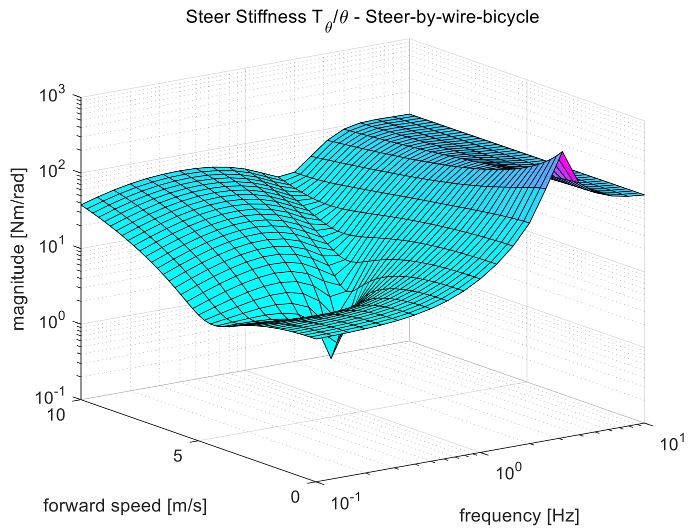
Figure 3. Magnitude and phase response of tracking transfer function of steer-by-wire system $HSBe(s)$, as a function of forward speed υ and frequency.

As a second measure of performance we compared the steer stiffness transfer functions of the benchmark bicycle and the steer-by-wire bicycle. The steer stiffness transfer function for the benchmark bicycle model is defined as $HBB(s)=T\delta(s)/\delta(s)$ and for the steer-by-wire bicycle are defined as $HSBW(s)=T\theta(s)/\theta(s)$ and as $HSBWS(s)=T\theta(s)/\delta(s)$. The steer-by-wire transfer function $HSBW(s)$ describes the steer stiffness perceived by the rider at the handlebars, whereas $HSBWS(s)$ describes the in series stiffness and damping of the handlebar and fork assembly as a combined system. The steer stiffness transfer functions of the benchmark bicycle $HSBW(s)$ and steer-by-wire system $HSBWS(s)$ are influenced in the same way and for this reason they are not discussed.

In Figure 4 the steer stiffness magnitude of the benchmark bicycle model (a) and the steer stiffness magnitude of the steer-by-wire bicycle (b) are presented as a function of input frequency and forward speed. At higher frequencies the steer stiffness is primarily defined by the mass and inertia properties of the bicycle. A significant drop in the steer stiffness magnitude relation occurs in both models at the weave speed (υ_w=4.29 m/s) and the corresponding weave frequency (0.55 Hz) of the bicycle shown by the downward peak approaching a zero stiffness. The steer stiffness of the steer-by-wire bicycle at higher frequencies is primarily defined by the stiffness and damping properties of the PD-controller, whereas the resonance-like upward peak at low forward speeds is caused by the PD-controller coefficients and the mass and inertia properties defined in the system matrix. The steer stiffness comparison shows an almost identical handlebar stiffness behaviour in a frequency range of 0-3 Hz and in a forward speed range of 0-10 m/s. Above 3 Hz the steer stiffness of the steer-by-wire bicycle shows a plateau whereas the steer stiffness of the benchmark bicycle continues to increase until it reaches a maximum of about 920 Nm/rad at 10 Hz. In other words, the steer-by-wire bicycle handlebars are more compliant compared to the benchmark bicycle, a larger steer angle is obtained for the same steer torque input. The difference in the handlebar steer stiffness might also indicate higher rider steer effort but does not influence bicycle stability.



(a)



(b)

Figure 4. Magnitude of the steer stiffness transfer function, as perceived by the rider, as a function of forward speed υ and frequency, (a) on the benchmark bicycle *HBB(s),* and (b) the steer-by-wire bicycle *HSBW(s)*.

3 EXPERIMENTAL SETUP DESCRIPTION

A custom made bicycle frame with two headtube assemblies was designed and built, see Figure 5. The basic geometry configuration of the custom frame was selected based on the dimensions of a Dutch city bicycle (Batavus Browser 54 cm). To adapt the frame geometry for an increased stack height (due to the extended headtube assembly) rattleCAD software was used. RattleCAD software provides a fully parametric bicycle model and was used in this case to evaluate the impact of an increase stack height on the rest of the frame geometry. A frame stack height of 713 mm and a frame reach of 358 mm were selected for this particular bicycle frame.

Figure 5. Prototype of the steer-by-wire bicycle with steering and handlebar actuators, sensors, digital controller and custom made battery pack.

Two identical Maxon EC45 brushless DC motors coupled with two Maxon GP42C planetary gearhead (36:1 reduction ratio) are used to actuate the fork and handlebar assembly. Belt drive (1:1 reduction ratio) transmissions are used to transmit torque from the motor to the fork and handlebar respectively. The existing drive configuration allows a maximum continuous torque of 7.5 Nm and a maximum instantaneous torque of about 11.3 Nm. The total backlash of the system is about 0.8° (mainly due to the planetary gearhead configuration). To provide the handlebar and fork tracking controllers with the required states two identical RMB20SC 13 bit absolute angular encoders are used. The simulation of the control loops, state estimation and data logging are implemented in a Teensy 3.6 microcontroller that runs an update loop at a rate of 1 KHz. The constant update loop rate is achieved by the implementation of a real time operational system (RTOS) called Teensy threads.

Additional sensors are implemented on the bicycle to acquire knowledge of the rest of the bicycle states. More specific, an inertial measurement (IMU) unit MPU9250 is located inside the electronic control box at the back of the bicycle rack, see Figure 5. The IMU is used to monitor 6D translational and rotational accelerations. Two gearwheel rotary encoders with a resolution of 192 counts /revolution combined with GTS35 reading head are used to measure pedal cadence and forward velocity at the rear wheel. A rear hub magic pie 5 motor driven by a throttle controller is used to cruise at high speeds without pedalling if needed. A deadband mechanism with ±5 deg play is designed as a safety to enable steering upon system failure. The locations of the sensors (excluding the IMU), actuators and safety mechanism are shown in Figure 6.

Figure 6. Steer-by-wire bicycle component layout, showing the physical placement of the actuators and sensors, with (a) headtube assembly with motor sensors and safety pin, (b) pedal speed sensor, and (c) rear wheel with forward speed sensor and hub motor.

4 TRACKING CONTROLLER PERFORMANCE

The proposed PD-controller gains of K_P=90 Nm/rad and of K_d=0.6 Nm/rad of the benchmark bicycle simulation would in practice result in unrealistic high actuator torques. On the other hand, unmodeled actuator and controller dynamics of the fork and handlebar assembly can also cause unstable oscillatory modes. For these reasons, the double PD-controller gains were selected experimentally to maximize the tracking performance without forcing the handlebar or fork assembly in an unstable mode. The effective stiffness and damping in the steer torque path and in the handlebar torque path are K_{PH}=0.9 Nm/rad, K_{DH}=0.012 Nm/rad and K_{PF}=2 Nm/rad, K_{DF}=0.025 Nm/rad respectively.

To test the tracking performance, five participants were asked to perform a set of slalom maneuvers in a controlled environment in the stable and unstable bicycle speed region. All participants could easily control the steer by wire bicycle without any training. The frequency steering range during the 90 second task for all participants swept between 0 and 2 Hz.

The tracking response of the controller in the time domain for one of the participants is shown in Figure 7. As can be seen, a maximum tracking error of approximately 0.05 rad can be noticed for large steering amplitudes ±0.4 rad and a tracking error of about 0.02 rad for smaller steering amplitudes ±0.25 rad. The mean tracking error during the 90 second stability tasks is 0.002 rad.

The frequency response of the tracking controller can be approximated by the empirical tracking transfer function, the ratio of the Discrete Fourier Transform (DFT) between the fork angle δ and handlebar angle θ. The empirical steer-by-wire tracking transfer function is defined as $HESBWe(s)=\delta(s)/\theta(s)$.

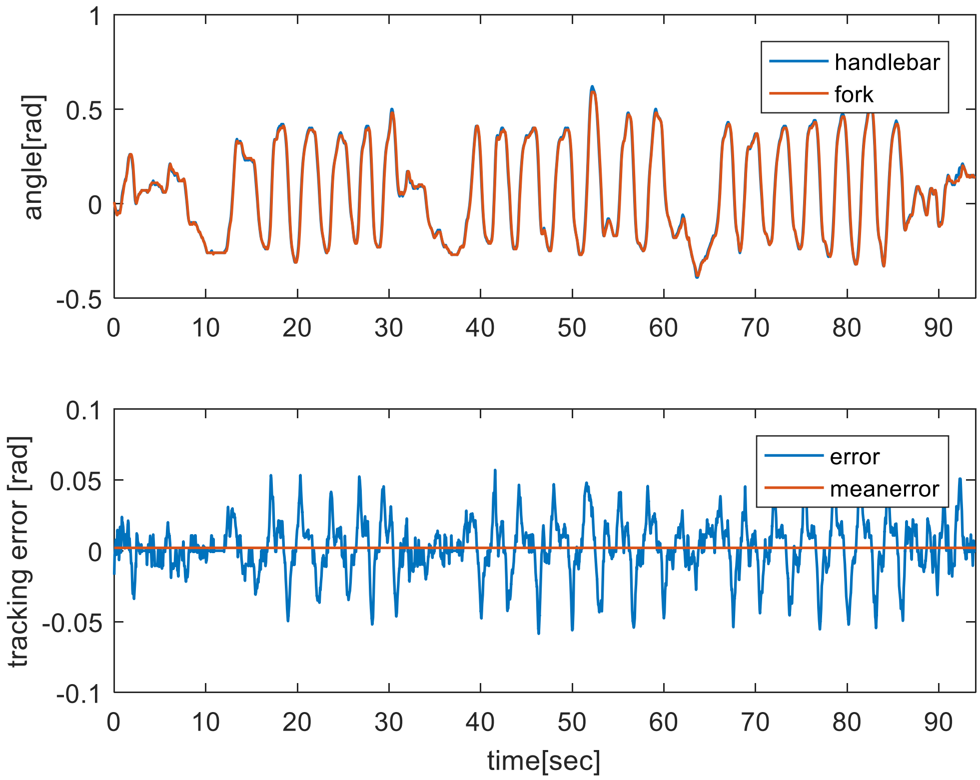


Figure 7. Commanded handlebar steering angle and actual fork angle, tracking error and mean tracking error, while a human subject performs a slalom.

A Savitzky-Golay filter (2ndorder, 113 frame length) is used to smooth the magnitude and phase data. The tracking magnitude and phase for a frequency range between 0-2 Hz is shown in Figure 8. As can be seen, the tracking amplitude ratio fluctuates between 0.95-1.1 for the entire frequency range, whereas the phase lag increases for higher frequencies and reaches a maximum of 24 deg at 2 Hz. In general the gain is close to one and the phase lag fluctuates between 10-20 deg indicating a behaviour approximating a mechanical connection. Future studies are recommended using feedforward control algorithms to improve system performance at higher frequencies.

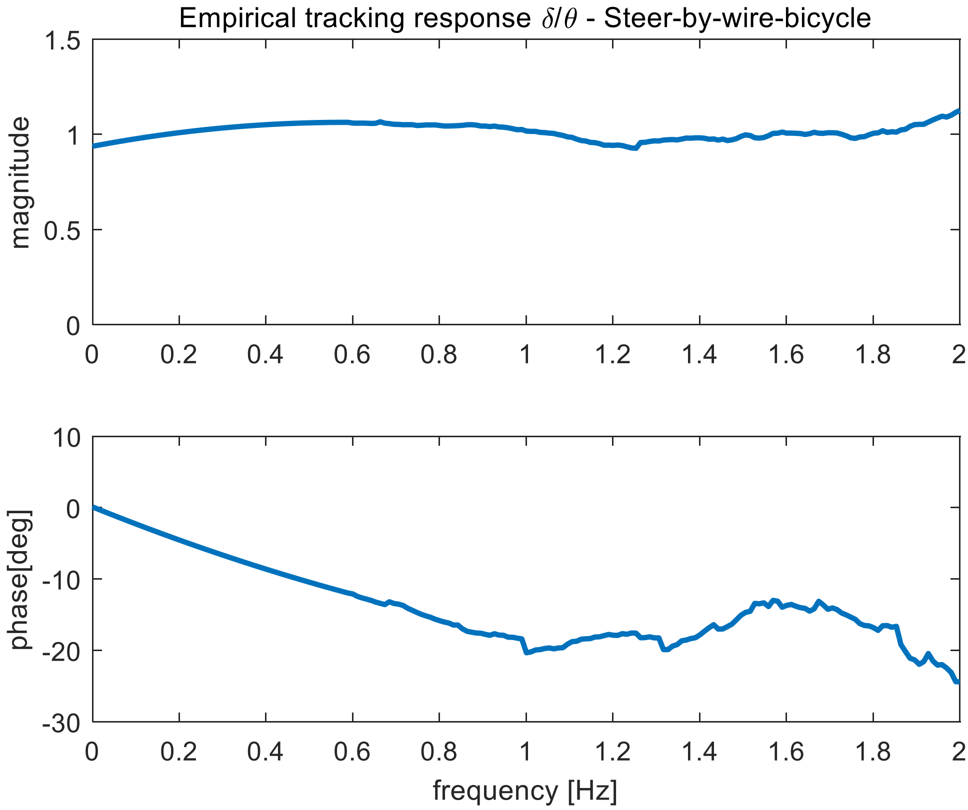


Figure 8. Magnitude and phase response of empirical steer-by-wire tracking transfer function $HESBe(s)$, as a function of frequency.

5 CONCLUSIONS

A steer-by-wire bicycle with a double PD-controller configuration has been designed and built. Simulations showed good tracking performance in a frequency range of 0-2.5 Hz and almost identical steer stiffness with the Whipple/Carvallo model [1] in a frequency range of 0-3 Hz and in a forward speed range of 0-10 m/s. Preliminary testing also showed a perceived near-to identical behavior of the steer-by-wire system to a mechanical connection. More specifically, a mean tracking error in the steer angle of 0.02 rad and a mean phase lag of 2 deg is noticed during the stability experiments. In future research the steer-by-wire bicycle will serve as a versatile experimental platform for identifying human rider control in bicycling [7], and development of support systems enhancing balance and handling.

6 ACKNOWLEDGEMENTS

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