IMPROVED HANDLING CHARACTERISTICS OF OFF-ROAD VEHICLES BY APPLYING ACTIVE CONTROL OF STEERING WHEEL TORQUE

J. Kis, L. Jánosi

Institute for Mechanical Engineering Technology, Faculty of Mechanical Engineering,
Szent István University, Gödöllő, Hungary

Abstract Driving speed of agricultural mobile machines have been increased in the recent years, raising serious questions about vehicle handling characteristics considering the high center-of-gravity, multi-mass configuration and rear-wheel-steering of these vehicles. The next generation of steering systems on off-road vehicles will incorporate a steering column mechatronic subsystem which will generate tactile feedback for operator. This paper presents our research work to utilize steering wheel torque to improve off-road vehicle handling characteristics.

Keywords agricultural mobile machines, rear wheel steering, off-road vehicles, haptic feedback, handling

1 INTRODUCTION

To gain productivity the weight and speed of off-road vehicles, particularly agricultural mobile machines is continuously increasing. Operation on the field (or in some circumstances at least retain mobility) and have safe and comfortable vehicle handling on public roads have contra verse requirements on vehicle design [6] Good example for the contra verse requirements is the tire design and pressure of agricultural tires. The design of the steering system is also a result of a suboptimal compromise between road transport and field operation. Some vehicle have articulated steering Figure 1/b which allows only very limited speed, because of the steering kinematics. Self propelled forage harvester or combine harvesters are equipped with rear wheel steering, which have several benefits for the field operation in terms of header operation, and also in machine design in terms of body clearance. This concept has also serious drawbacks in terms of vehicle handling in transport mode, because of the stability problems with rear wheel steering.

In Steer-by-Wire (SBW) systems there is no mechanical or hydraulic link between hand wheel and road wheels. The hand wheel angle is measured by redundant rotary sensors, processed by an electric control unit to obtain road wheel angle, and calculates also the set value for the actuator control. [3] This concept enables decoupling the motion and the forces of the road wheels from the hand wheel motion and torque, which makes possible to implement various driver supporting control systems. Using an active force feedback device on hand wheel unit, the SBW system can provide various type of haptic signals for the operator.

In this paper we investigate the stability conditions for the heavy-duty agricultural vehicle with rear wheel steering. Considering of a SBW system with active force feedback, we present one method to emulate synthetic torque on hand wheel, which corresponds to the ideal aligning moment of a virtual front wheel steered vehicle. The paper will provide information about the calculation of the state feedback gain, and the stability of the modified system. The concept uses the state variables of the vehicle motion, which can be estimated on various ways, and these methods are not part of this paper.

2 INFLUENCE OF MODEL PARAMETERS ON STABILITY

2.1 Vehicle model

For the investigation of the rear wheel steering system, and the handling behavior of the heavy-duty agricultural mobile machine we use a linear single track model. The model and the resulting differential equations are derived in and analyzed for the front wheel steering system but there is limited information about pure rear wheel steering systems [2,4,7].

The model shown on Figure 1/a is body with 3 degree of freedom, whereby motion in z-plane is allowed. The differential equations for lateral and rotational motions are:

$$m\,(r+\dot{\beta})v = F_{SR} + F_{SF} \qquad \dot{r}I = a\,F_{SF} - b\,F_{SR}$$

The model is valid below 0.4g lateral acceleration, and below 5 degree lateral tire slip. We investigate the vehicle under these conditions in road transport mode. Assuming a constant velocity along x axis the state variables of the model can be reduced to the rotational speed *r* and slip angle β. There are two lateral tire forces acting on the chassis, whereby longitudinal components are ignored.

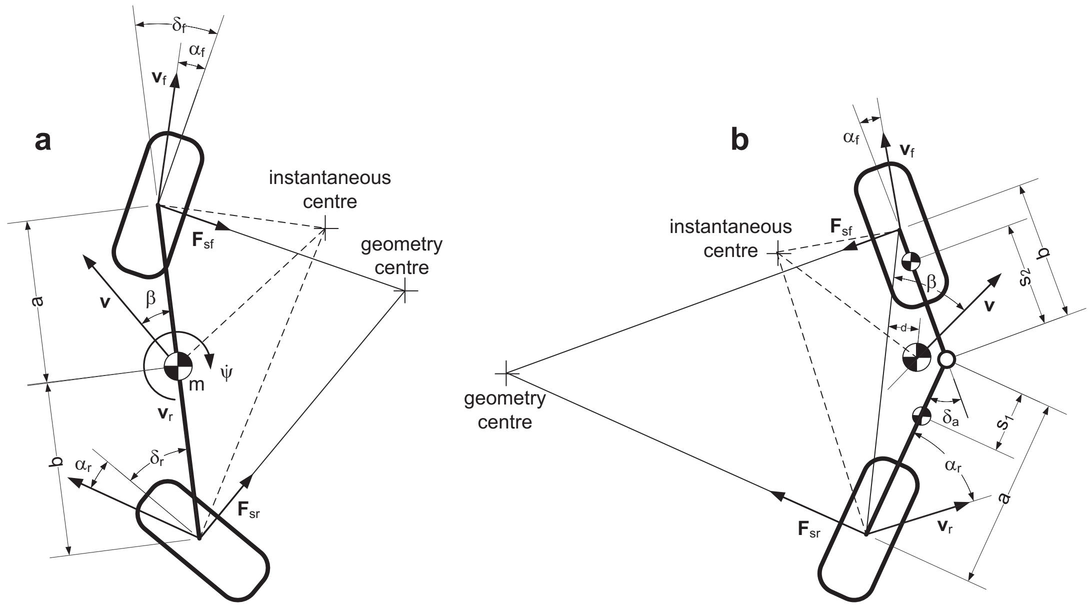


Figure 1: Linear single track models for front/rear and articulated steering

Because vehicle behavior is function of tire lateral force and moments, therefore model accuracy is significantly influenced by the accuracy of tire models. There are widely accepted nonlinear tire models, e.g. the “magic formula” to cover different aspects of tire-road interactions. Due to small lateral slip, in our analysis we use a simple linear relationship between the lateral force and lateral slip, and aligning moment and lateral slip. The constant parameter used in the model is the cornering stiffness ($C_{F,R}$). In our analysis we don’t consider the effect of suspension flexibility on lateral tire forces, and so the elasto-kinematic trail, because we investigate a system with no suspension and very rigid steering linkage (hydraulic cylinders acting directly on the wheel). We ignore the fact also that most of the mobile machines equipped with rear wheel steering are designed with large castor angle and mechanical trail. The model consider only the effect of the pneumatic trail on aligning moments.

$$F_{SF} = \alpha_F C_F \qquad F_{SR} = \alpha_R C_R$$

Using Figure 1/a we can obtain the equations for the tire lateral slip angles from rotation and slip angle:

$$\alpha_F = \delta_F - \frac{a\,r}{v} - \beta \qquad \alpha_R = \delta_R + \frac{b\,r}{v} - \beta$$

Combining the equation from above we can set up the differential equations for the single track model in the form where the derivatives of state variables are on the left hand side:

$$\dot{\beta} = \frac{(v\,\delta_R - \beta v + b\,r)C_R + (v\,\delta_F - \beta v - a\,r)C_F - mr\,v^2}{m\,v^2}$$

$$\dot{r} = -\frac{(b\,v\,\delta_R - b\beta v + b^2 r)C_R + (-av\delta_F + a\beta v + a^2 r)C_F}{vI}$$

Using these equations we can obtain the state equations of the system in matrix form $\dot{x} = A\,x + B\,u$.

$$\begin{bmatrix}\dot{\beta}\\ \dot{r}\end{bmatrix} = \begin{bmatrix} -\frac{C_R + C_F}{m\,v} & \frac{b\,C_R - a\,C_F - m\,v^2}{m\,v^2} \\ \frac{b\,C_R - aC_F}{I} & -\frac{b^2 C_R + a^2 C_F}{v\,I}\end{bmatrix}\begin{bmatrix}\beta\\ r\end{bmatrix} + \begin{bmatrix}\frac{C_F}{m\,v} & \frac{C_R}{m\,v}\\ \frac{a\,C_F}{I} & -\frac{b\,C_R}{I}\end{bmatrix}\begin{bmatrix}\delta_F\\ \delta_F\end{bmatrix}$$

The system state variables are $x_1=\beta$ $x_2=r$ and the inputs are the steering angles $\delta_{F,R}$. The differential equation system in matrix form.

2.2 Stability of rear wheel steered vehicles

Why pure rear wheel steering is not used in passenger cars has several reasons. One is that the tire aligning moment acts on opposite direction, and changes direction in transient situation. Therefore mechanical force feedback can not be realized. The other reason is, that because of the flexibility of the steering system and the suspension, an elasto-kinematic steering angle created, if lateral force is present. The opposite aligning moment increases this angle, therefore there is no stable operating point for this situation.

Driving the vehicle on steady-state path, the realized elasto-kinematic steering angle reduces the lateral force, which has an impact on the rear axle cornering stiffness and so on over steering behavior of the vehicle.

2.3 Modeling the mechanical steering system

To compare front- and rear wheel steering, the single track vehicle model is extended with the model of the steering system. In this model the inertia, viscous friction and (as an external disturbance) hand wheel steering torque are modeled, and it can be regarded as a feedback system. The differential equation used for the steering system model:

$$\ddot{\theta} J_S = -\tau_{VISC} + \tau_{EX} - \tau_{AL}$$

The viscous friction is proportional to the angular speed of the steering system. This parameter can be part of the control system and can represent a virtual friction to reduce system oscillations. The aligning moment depends only on the pneumatic trail and lateral force. The kinematic relationship between hand wheel angle and road wheels are modeled with fixed ratio i_S.

$$\tau_{VISC} = \dot{\theta} d_S \qquad \tau_{AL} = \frac{tF_S}{i_{ST}} \qquad \delta_F = \theta\, i_S$$

Combining the equations from above, and expressing the angular acceleration the differential equation of the steering system:

$$\ddot{\theta} = -\frac{\dot{\theta}\, v\, i_S^2\, d_S - v\, \tau_{EX}\, i_S^2 + (-\beta\, t\, v - a\, r\, t) C_F\, i_S + t\, \theta\, v\, C_F}{v\, i_S^2 J_S}$$

Substituting the rear wheel steering angle in the differential equations of the single track vehicle model:

$$\dot{\beta} = -\frac{(\beta\, v - b\, r)\, i_S C_R + \left((\beta\, v + a\, r) C_F + m r\, v^2\right) i_S - \theta v\, C_F}{m\, v^2 i_S}$$

$$\dot{r} = \frac{(b\, \beta\, v - b^2 r) i_S C_R + (-a\, \beta v - a^2 r) C_F i_S + a\, \theta\, v\, C_F}{v\, i_S I}$$

This gives the state equations for the front wheel steering system in the following form:

$$\begin{bmatrix} \dot{\beta} \\ \dot{r} \\ \ddot{\theta} \\ \dot{\theta} \end{bmatrix} = \begin{bmatrix} -\frac{C_R}{mv} - \frac{C_F}{mv} & \frac{bC_R}{mv^2} - \frac{aC_F}{mv^2} - 1 & 0 & \frac{C_F}{i_S mv} \\ \frac{bC_R}{I} - \frac{aC_F}{I} & -\frac{b^2 C_R}{vI} - \frac{a^2 C_F}{vI} & 0 & \frac{aC_F}{i_S I} \\ \frac{tC_F}{i_S J_S} & \frac{atC_F}{v i_S J_S} & -\frac{d_S}{J_S} & -\frac{tC_F}{i_S^2 J_S} \\ 0 & 0 & 1 & 0 \end{bmatrix} \begin{bmatrix} \beta \\ r \\ \dot{\theta} \\ \theta \end{bmatrix} + \begin{bmatrix} 0 \\ 0 \\ \frac{1}{J_S} \\ 0 \end{bmatrix} \theta$$

The state equation for the rear wheel steered vehicle :

$$\begin{bmatrix}\dot\beta\\ \dot r\\ \ddot\theta\\ \dot\theta\end{bmatrix}=\begin{bmatrix}-\frac{C_R}{mv}-\frac{C_F}{mv} & \frac{bC_R}{mv^2}-\frac{aC_F}{mv^2}-1 & 0 & \frac{\boldsymbol{C_R}}{i_S mv}\\ \frac{bC_R}{I}-\frac{aC_F}{I} & -\frac{b^2C_R}{vI}-\frac{a^2C_F}{vI} & 0 & -\frac{\boldsymbol{bC_R}}{i_S I}\\ \frac{tC_F}{i_S J_S} & \frac{atC_F}{v i_S J_S} & -\frac{d_S}{J_S} & -\frac{\boldsymbol{tC_R}}{i_S^2 J_S}\\ 0 & 0 & 1 & 0\end{bmatrix}\begin{bmatrix}\beta\\ r\\ \dot\theta\\ \theta\end{bmatrix}+\begin{bmatrix}0\\ 0\\ \frac{1}{J_S}\\ 0\end{bmatrix}\theta$$

The difference between the two system matrix (A) are marked bold. We can see the negative feedback of the hand wheel angle on the yaw rate change and the angular acceleration, which cases the system instability.

In the following section we will analyze the stability of this system in function of different model parameters using the eigenvalues of the system matrix. It is known, that at given model parameter set, if all the eigenvalues of *A* have negative real parts the system is stable. Figure 2 shows the eigenvalues of the front and rear wheel steered system at changing model parameters.

Figure 2/a shows the effect of increasing ground speed on the system pole locations. Both pole pairs are slowing down, and the damping of the system is not changing. Because of the increasing phase lag the system poles get positive real parts, and at 25 m/s the system get unstable. Figure 2/b shows the effect of decreasing rear tire cornering stiffness. The damping of the faster pole pair is decreasing, due to the decreasing aligning moment. The slower pole pair get into the unstable region because of the over steering.

It is obvious if the pneumatic trail get reduced (Figure 2/c) the controllability of the system decreases, which shows over damped poles at not changing viscous friction. If speed increases a higher viscous fraction value is required to keep vehicle stable (Figure 2/d). This is one of the concept used in the following section to modify vehicle handling characteristics. Figure 2/e shows the system poles of the rear wheel steered vehicle at increasing speed. It is obvious that this system is unstable, because it has one real poles in the right hand plane.

3 ALTERING VEHICLE HANDLING CHARACTERISTICS

In this paper a concept will be presented to alter vehicle handling using an active force feedback device which will generate virtual aligning moment for the operator to keep the vehicle stable. In the following model we assume, that the wheels are controlled by an electromechanical actuator, which can “keep-up” with the hand wheel dynamics, disregarding the control effort problem in this model.

3.1 Haptic feedback modeling

The haptic feedback is realized with a brushless DC motor and teethed belt transmission. The haptic feedback contains inertia, viscous friction, a ratio to hand wheel and a ratio the steered wheels. The torques acting on the motor shaft are:

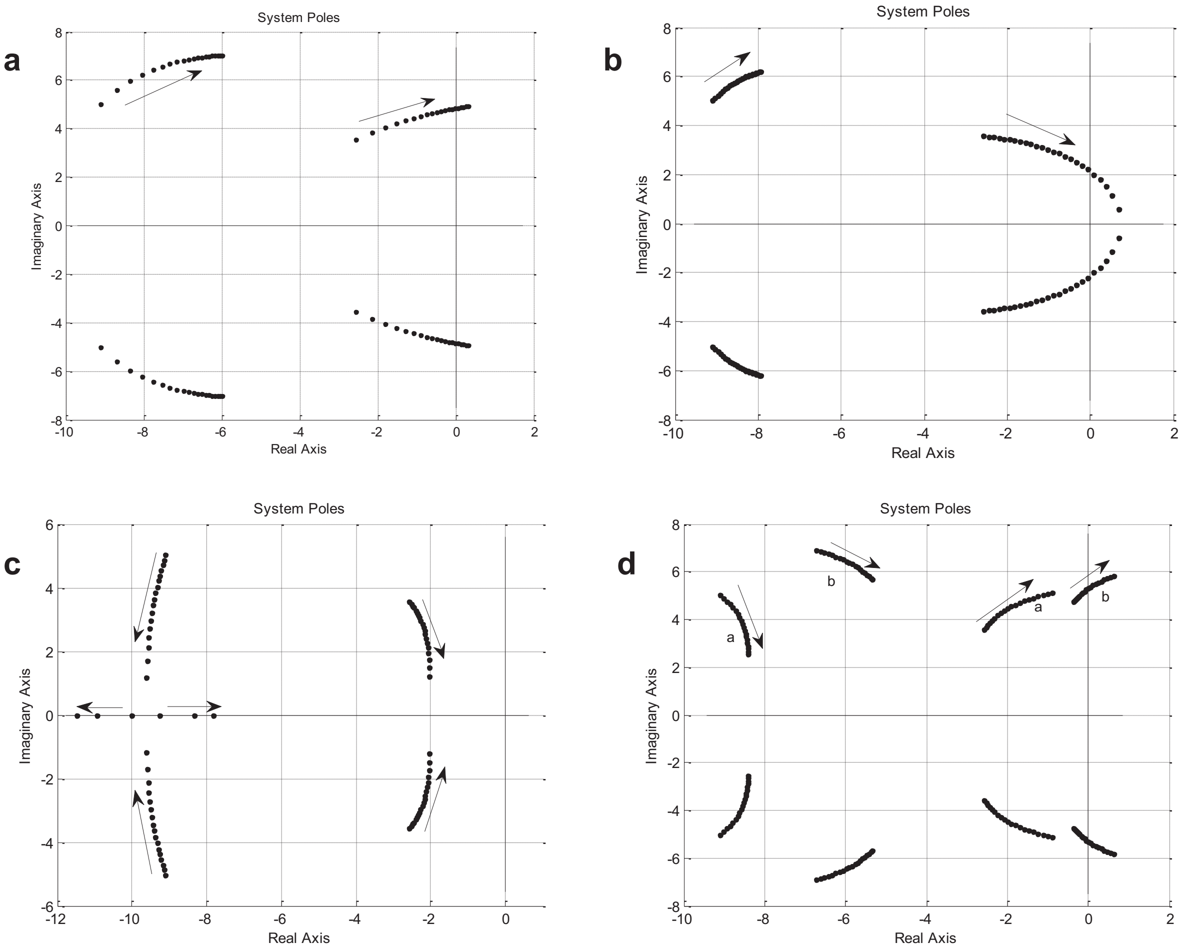
$$\ddot\varphi\, J_H = i\, K_M + \frac{\tau_D}{i_H} - \dot\varphi\, K_F$$

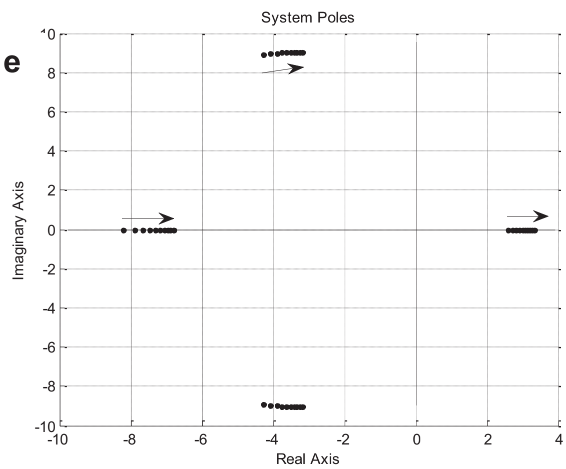
The generated back electromagnetic force (BEMF) using the motor parameters:

$$v_B = \dot\varphi\, K_B$$

The electrical differential equation with impedances for the brushless dc motor:

$$v - v_B = i\, R + \frac{di}{dt}\, L$$





a. increasing ground speed: loosing stability at 25mps (positive real part of system matrix eigenvalues)
b. System poles at decreasing rear wheel cornering stiffness: loosing stability
c. decreasing pneumatic trail, remaining viscous friction
d. decreasing viscous friction, at 10 m/s (a) and 20 m/s (b)
e. unstable Rear Wheel Steering poles at increasing speed

Figure 2: System pole locations with changing model parameters.

Combining the mechanical and dynamic differential equations we get for the haptic feedback:

$$\ddot{\varphi} = \frac{i\, i_H\, K_M - \dot{\varphi}\, K_F\, i_H + \tau_D}{J_H i_H}$$

$$\frac{di}{dt} = \frac{i\, R + \dot{\varphi}\, K_B - v}{L}$$

The state equations for the haptic feedback device:

$$\begin{bmatrix} \dot{i} \\ \ddot{\varphi} \\ \dot{\varphi} \end{bmatrix} = \begin{bmatrix} -\frac{R}{L} & -\frac{K_B}{L} & 0 \\ \frac{K_M}{J_H R} & -\frac{K_F}{J_H} & 0 \\ 0 & 1 & 0 \end{bmatrix} \begin{bmatrix} i \\ \dot{\varphi} \\ \varphi \end{bmatrix} + \begin{bmatrix} \frac{1}{L} & 0 \\ 0 & \frac{1}{J_H i_H} \\ 0 & 0 \end{bmatrix} \begin{bmatrix} v \\ \tau_D \end{bmatrix}$$

Because of the limited dynamics of the whole system, in the simulation we ignore the inductive impedance. And therefore we can reduce the number of state variables to angular speed and angle position. The reduced state equations are:

$$\begin{bmatrix} \ddot{\varphi} \\ \dot{\varphi} \end{bmatrix} = \begin{bmatrix} -\frac{K_B K_M}{J_H R} - \frac{K_F}{J_H} & 0 \\ 1 & 0 \end{bmatrix} \begin{bmatrix} \dot{\varphi} \\ \varphi \end{bmatrix} + \begin{bmatrix} \frac{K_M}{J_H R} & \frac{1}{J_H i_H} \\ 0 & 0 \end{bmatrix} \begin{bmatrix} v \\ \tau_D \end{bmatrix}$$

The input of the haptic feedback is the armature voltage which is controlled by the state feedback controller.

3.2 Explicit calculation of feedback gain for emulated front wheel steering

The control system is shown on Figure 3. There are several methods for optimal design for the feedback gain, e.g. the concept of the linear quadratic regulator (LQR). It will give optimal solution for full state feedback at given transient and steady-state constrains.

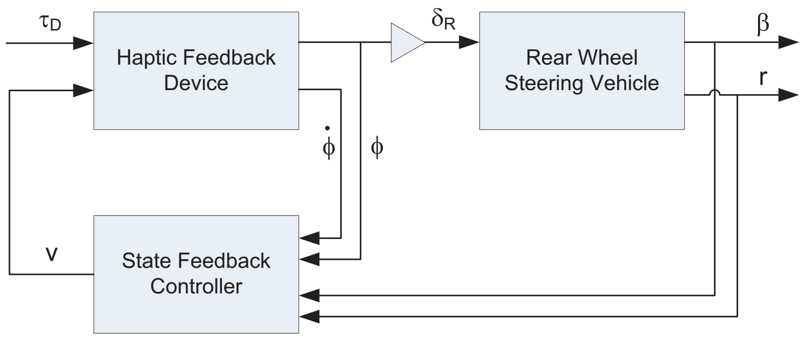


Figure 3: State feedback controller for rear wheel steered vehicle (without steering actuator dynamics) based on emulated front wheel aligning moment and added viscous friction

We follow a different method by explicit calculating the feedback gain, because we would like to emulate a steering system which has a front wheel steering. So the system may work suboptimal compared to the LQR design but gives the operator the feeling of an front wheel steered vehicle. The controller has to calculate the necessary voltage to create realistic steering torque:

$$v = \frac{\tau_M R + \phi \, K_B K_M}{K_M}$$

The emulated hand wheel torque can be calculated on the virtual aligning moment on the front wheels in case of an front wheel steering plus the synthetic viscous friction in the system, which is used to modify system dynamics.

$$\tau_M = \frac{\tau_{AL}}{i_H \, i_M} - \tau_{VISC}$$

Substituting into the electric equation we get for the voltage:

$$v = \frac{\left(\frac{t\, C_F\left(-\frac{\varphi}{i_H i_S} - \frac{a\, r}{v} - \beta\right)}{i_H i_M} - \phi K_F\right) R + \phi\, K_B\ K_M}{K_M}$$

After modifying the equation above we can describe the four feedback gains for all four state variables:

$$v = \left\{K_B - \frac{K_F R}{K_M}\right\}\phi - \left\{\frac{t\, C_F R}{i_H^2 i_M i_S K_M}\right\}\varphi - \left\{\frac{t\, C_F R}{i_H i_M K_M}\right\}\beta - \left\{\frac{a\, t\, C_F R}{v\ i_H i_M K_M}\right\}r$$

Some parameters of the control system are given by vehicle geometry or electrical components(a, b, R, etc.) State variables are directly or indirectly measurable like yaw rate, lateral acceleration angular speed, angle. Other parameters like cornering stiffness must be estimated by online system identification algorithms. The results of the simulation on unit hand wheel torque is shown on figure 5.

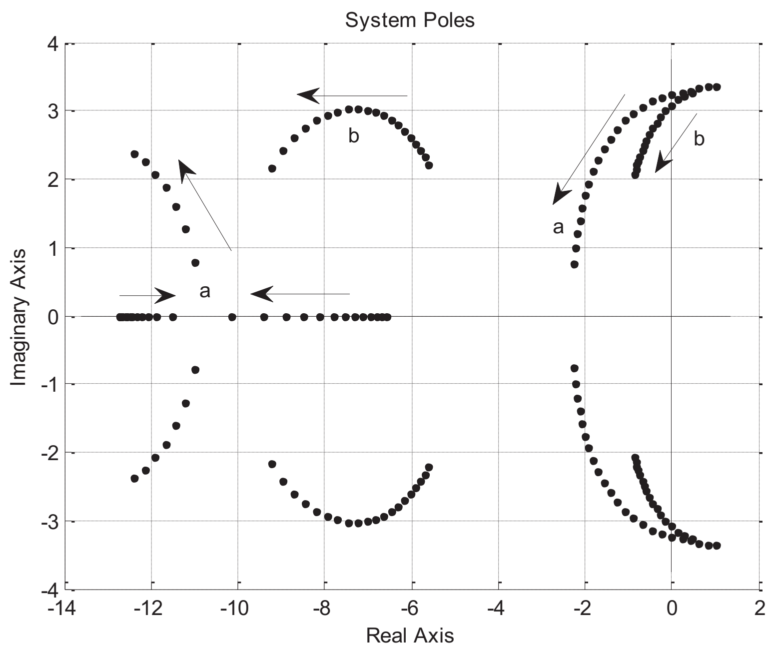


Figure 4: System poles of the single track model with haptic feedback at increasing damping at 10 mps(a) and 20 mps(b)

4 CONCLUSIONS

Off-road vehicles with rear wheel steering have acceptable handling only at lower speeds. This behavior can be significantly improved by using active haptic feedback systems. The presented method of explicitly calculating the feedback gain to emulate front wheel steering feedback can improve the stability of system. This is shown on Figure 4, whereby the closed loop poles of the system with increasing damping at different ground speed are on the left hand plane, If appropriate synthetic viscous friction has been set.

Further work is necessary to consider a road wheel actuator with finite dynamics, analyze the effect of the state estimation and parameter identification. The presented concept must be also verified on real vehicle.

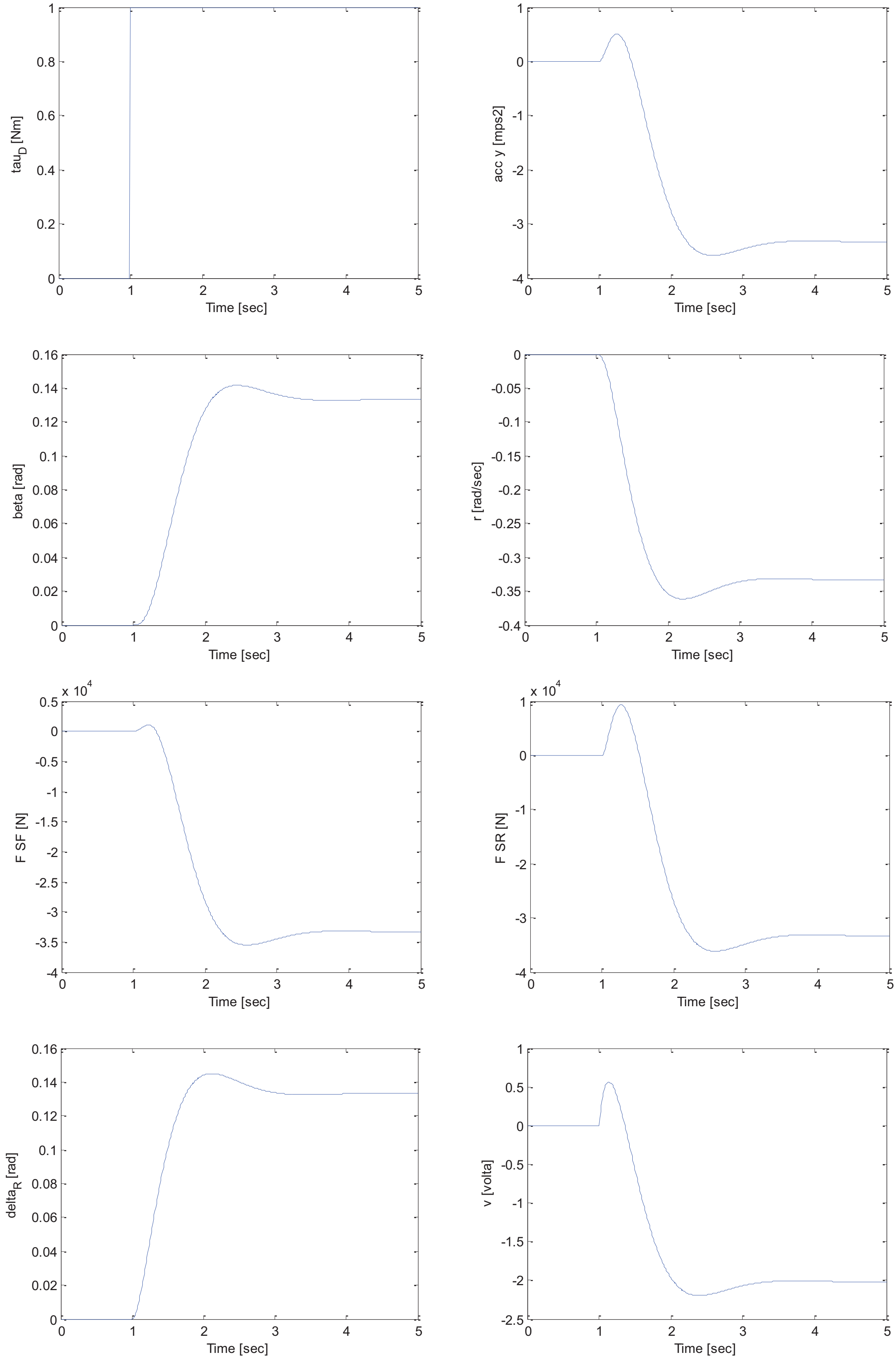


Figure 5: Step response of the system to unit hand wheel torque

5 NOMENCLATURE

$F_{SF,SR}$	Lateral tire force	N	τ	Torque on hand wheel	Nm
$C_{F,R}$	Cornering stiffness	N/rad	i_S	Steering ratio	-
$\alpha_{F,R}$	Tire slip angle	rad	ϕ	Motor shaft angle	rad
$\delta_{F,R}$	Steering angle	rad	J_H	HapticFeedback inertia	kgm2
a,b	acc to fig. 1	m	ι	Current	A
r	Yaw rate	rad/s	K_M	Motor torque constant	Nm/A
β	Slip angle	rad	K_F	Viscous constant	Nms/rad
v	Ground speed	m/s	K_B	BEMF constant	Vs/rad
v	Voltage	V	R	Resistance	Ohm
I	Inertia	kgm2	i_H	Haptic ratio	-
θ	Hand wheel angle	rad	i_M	Force feedback ratio	-
J_S	Steering inertia	kgm2			

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