Modelling Steering Resistance to Save Energy

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Abstract. In this paper, the design and vehicle model implementation of a spiral path steering system for a lightweight vehicle is presented. The lateral sliding caused by imperfect rolling in corners means considerable loss, for lightweight vehicles. A spiral path steering mechanism provides an alternative solution for steering generated loss reduction. Theoretically, with the application of a spiral path steering system, ideal Ackerman steering geometry is feasible in all corners. The system is able to replace the common rack and pinion solution in purpose-made vehicles. The main design concepts of spiral path steering are introduced in this article. The described system was realized and installed in an experimental vehicle, where field tests were carried out, to measure the cornering losses during operation. The process of cornering loss measurement is also presented in this paper. The resistance model of cornering is elaborated in a Matlab Simulink environment, based on the measurement results. Vehicle losses are characterized by an extended resistance force model, which can be used by simulations for energy saving purposes. The optimization of vehicle operation can be achieved by simulations, where the corresponding velocity profiles of the vehicle is determined. An accurate steering model is essential for proper vehicle modelling and for the following optimization process.

1. Introduction

Achieving energy efficient vehicle operation is a fundamental goal of all electric vehicles. In urban environments the usage of electric vehicles is more than beneficial, and their CO2 reduction potential is also underestimated. In this paper, a special one seat experimental electric vehicle is investigated, which is dedicated for the Shell Eco-marathon (SEM), energy efficiency competition. The goal of the event is to complete the race in a given time by using the least amount of energy. An energy efficient vehicle operation is a complex task, where driving strategy is acquired by solving the control problem on the suitable vehicle model and objective function. A mathematical vehicle model can be formulated by discrete equations, which provide a robust solution; but in special cases other vehicle model formulation methods could achieve greater accuracy. In the presented work, measurement-based vehicle modelling was used according to the unique characteristics of the vehicle. The extreme low mass, highly aerodynamic shape and optimized drive components of the vehicle results in low overall power demand. The traction force demand is between 10–25 N, which makes high accuracy vehicle modelling reasonable. On the other hand, the arising resistance forces should also be modelled precisely, because even moderate changes could have significant effects on the vehicle’s operation. Resistance force modelling can be distinguished in straight moving and in cornering. In this work, the construction of the steering system and the related losses are investigated. [1][2]
Figure 1. SZEmission - experimental electric energy efficient vehicle

2. Ackermann Steering Geometry
The applied steering system and geometry have an important role in vehicle dynamics. In order to achieve energy efficient driving, cornering losses should not be neglected. The ideal Ackerman geometry needs to be maintained in every turn, where tire slip and loss of traction is minimized. The base concept of Ackermann steering is presented on figure 2. The inner and outer wheels are pointing exactly to the actual center of the turn, which is collinear with the rear axle. According to the geometry the inner and outer wheels have different angles. [3][4][5]
\[ \alpha = \tan^{-1}\left( \frac{L}{R + \frac{b}{2}} \right) \]  

(1)

\[ \beta = \tan^{-1}\left( \frac{L}{R - \frac{b}{2}} \right) \]  

(2)

The Ackermann angles depend on the vehicle’s characteristics, especially on the kingpin distance in the front suspension. The vehicle parameters of SZEmission are summarized in table 1.

**Table 1. Vehicle parameters of SZEmission**

| Property                | Value (mm) |
|-------------------------|------------|
| Wheelbase (L)           | 1300       |
| Front track             | 1000       |
| Rear track (B)          | 800        |
| Drag link radius (Ls)   | 19.5       |
| Kingpin distance (b)    | 961        |

Based on equation (1) and (2) and the vehicle parameters in table 1, the discrete Ackermann angles can be determined for the investigated vehicle, which are visualized in figure 3.

**Figure 3.** Calculated Ackermann wheel angles of SZEmission
3. Optimal Spiral Path Steering Method

The currently applied steering system is based on the optimal spiral path concept. This concept describes the steering spiral, which guarantees the ideal Ackerman angle in every position of the steering system, without using steer-by-wire methods, which are not allowed by the SEM regulation. In this study, the spiral disk coordinates are determined by the introduced measurement method, instead of CAD modelling. The base concept of measuring is to have a fully assembled suspension, where the displacement of the steering rod (x) can be measured, without the built-in steering system. The aim of measurement is to define the relation between the shaft displacement and wheel angle. In the investigated case, an absolute, guided linear encoder was used to monitor the shaft displacement, while the wheel was turned manually. The schematic layout of the measurement is shown in figure 4. [4][5][6]

![Figure 4. Optimal Spiral Path measuring method](image)

According to the trigonometric relationships in figure 4, the inner wheel angle (β) can be determined as follows:

$$\beta = \cos^{-1}\left(\frac{j - k}{c}\right) - 90^\circ$$  \hspace{1cm} (3)

For practical reasons the c parameter was chosen as a constant value of 1 m, while k and j distances were measured with laser distance sensors with 0.001 m accuracy. The perpendicularity of c section was guaranteed by cross line lasers, which was aligned for every measurement. Several measurements were made, while the wheel angle was manually changed. The results of measurements are presented in figure 5. The measurements were carried out on both sides of the vehicle.
Figure 5. Measured $\beta$ and steering rod displacement

Based on the measurement results, the steering rod displacements for the theoretical Ackermann angles can be defined. The steering range of the steering wheel was determined from 0° - 160° ($\lambda$), which is suitable for an urban vehicle, where the turning ability is important with even with fine steering movement. The base radius ($r_{base}$) of the spiral path needed to be defined for manufacturability reasons. In the presented work it was considered to be 70 mm.

Table 2. Denotation for spiral path calculation

| Property                      | Denotation |
|-------------------------------|------------|
| Base radius for spiral path [mm] | $r_{base}$|
| Steering rod displacement [mm]   | $x_{shaft}$|
| Steering angle [°]              | $\lambda$  |

The $x$ and $y$ coordinates of the spiral path can be calculated by using equation (3) and (4) in the whole range of $\lambda$.

\[
x = (r_{base} + x_{shaft}) \times -\cos(\lambda) \tag{4}
\]

\[
y = (r_{base} + x_{shaft}) \times \sin(\lambda) \tag{5}
\]

Based on the measurements, calculations need to be done for the left and right wheel, independently. The points can be connected with spline interpolation to form the representative spiral form. The calculated coordinates can be found on figure 6.
The vehicle is heading in $y^+$ direction and the steering rods are connecting to the spiral path at the points of intersection of the $x$ axis. The left steering rod is connected to the left side (solid line), while the right is connected to the right side (dashed line). The left and right spiral paths are facing in opposite directions to make the connection mechanically feasible. This mechanism allows the wheels to move independently, while keeping the calculated Ackermann angles in every steering position. The spiral paths can be mirrored, if symmetry is assumed. In this study the spiral paths are slightly different because of the combined effect of manufacturing related unwanted asymmetry in the suspension and measurement inaccuracies.

4. Steering System Overview
The calculated spiral disk geometry is implemented in the mechanical system, where the connecting rods are supported by square section linear bearings. An absolute linear encoder, used for the previous measurements, is also placed in the construction, beside the DC stepper motor, both are required for complete autonomous control. According to the rules, the DC stepper motor is not allowed in human driving challenges, so this part of the steering needs to be easily detachable. The drive connection is realized by a synchronous belt drive, with 1:1 gear ratio. The belt drive is also detachable, and the belt pretension can be set in the console of DC stepper motor. The complete assembly of the steering system is presented in figure 7.
The DC stepper is operated based on the commands of the high-level autonomous control system. This control system is responsible for trajectory planning of the vehicle, during autonomous operation. The trajectory planning is based on the evaluation of the data collected from camera and Lidar sensors. The stepper motor rotates the spiral disk, which makes the steering rods move translationally. The translational movement is monitored by the linear encoder providing feedback for the high-level control system. The advantage of this solution is to have accurate feedback, by including all the backlash of the mechanical system. The relation between the linear movement and cornering radius is shown in figure 8. According to this relation, the current cornering radius can be assigned to the steering rod displacement.

![Assembled steering system](image)

**Figure 7.** Assembled steering system

![Cornering radius - Steering Rod Displacement](image)

**Figure 8.** Cornering radius - Steering Rod Displacement
5. Steering Resistance Modelling

The created steering system keeps the wheels in their predetermined position maintaining ideal Ackermann angles, but the cornering still causes more resistance, than moving straight. The cornering losses need to be measured to determine the optimal driving strategy. Field test measurement methodology was introduced to measure cornering losses. Several measuring scenarios were set up with different speed levels and cornering radius values. In these tests, the vehicle speed was controlled by a real-time speed controller, while the driver manually followed the designated route with the given cornering radius. Resistance forces can be calculated from the traction force demand, as it equals to the resistance force, when the system is steady. The results were manually evaluated and combined with the free rolling test to set up the extended resistance force model. The created model was elaborated in a MATLAB environment, where the cumulative resistance forces in cornering can be determined; which includes all the arising resistance of aerodynamics, rolling, and tire slip. The model is shown in figure 9. [2]

![Figure 9. Extended resistance model of the SZEmission](image)

The physical feasibility starts from the cornering radius value of 6 m, and cornering phase lasts until the cornering radius value reaches 200 m, moving straight is considered from 200 m. In figure 9, the black plane distinguishes the left and right turn, and different resistance force values can be found in each side. The powertrain requires more energy in the right corners as the resistance forces are higher at speed levels over 25 km/h. The unequal weight distribution could be responsible for that difference. The rear left wheel is driven by the powertrain, which makes the weight distribution shift to the left. In relatively high speed right corners the rear left wheel is under load and the right wheel could lift of from the ground, decreasing the rolling and the slip of tires. This effect could cause higher resistance force values, which the measurement-based model also confirms. The resistance model is implemented in the vehicle model, where the vehicle speed and vehicle position are determined. Cornering radius can be calculated from the vehicle position on the track and actual vehicle speed, while the grade resistance can also be defined from height and length data of the track. The vehicle is accelerated by the traction force against the summarized resistance force ($\Sigma F_{\text{slow}}$). The theoretical operation of the track and resistance model is shown in figure 10. The acquired resistance force ($\Sigma F_{\text{slow}}$) is looped back to the mathematic vehicle model to calculate the vehicle speed and position.
6. Conclusion
In this paper, a novel approach of steering mechanism design and modelling for lightweight vehicles was presented. A complete process of the measurement based steering design was introduced. The design process is based on static measurements, so the dynamic behavior of the suspension is not taken into consideration. The created steering system was tested in the experimental vehicle, where resistance force measurements were carried out to define energy demand of cornering. The obtained results have shown that the resistance forces are different in the left and right corners because of the vehicle characteristics. The relatively low differences in resistance force (3-5 N) also need to be considered in the vehicle model, while creating driving strategy optimization. The proposed vehicle model implementation is suitable for creating driving strategy optimization.

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