KORMÁNYMECHANIZMUS TERVEZÉSE KÖNNYŰSZERKEZETES JÁRMŰHÖZ

STEERING MECHANISM DESIGN FOR LIGHTWEIGHT VEHICLE

Zoltán PUSZTAI, Péter KŐRÖS***

**Széchenyi István University Research Center of Vehicle Industry, Győr, Hungary, pusztai.zoltan@ga.sze.hu*

***Széchenyi István University Research Center of Vehicle Industry, Győr, Hungary,* korosp@ga.sze.hu

ABSTRACT

In this paper the main design concepts and calculations of an alternative steering mechanism are presented, which are dedicated for purpose-made lightweight vehicles. Losses are generated during steering as the tires are not rolling perfectly in curves, what are caused by lateral sliding. The widely used Ackerman steering geometry has its specific error. In order to avoid Ackermann error, the design of an alternative steering mechanism was carried out. This mechanism aims to keep both steered wheels at ideal turning angles. This cannot be achieved using the common rack and pinion steering, therefore a mechanism was designed based on our optimal spiral path method. The advantages of this steering besides eliminating Ackerman error, is that the mechanism is able to self-locking. This mechanism is ideal base for autonomous application as the system doesn't require energy to hold the turning angle, it just consumes energy when the angle is changed. This article covers the main theoretical concept, design fundamentals, FEM simulations and calculations.

1. INTRODUCTION

The alternative vehicle development has great tradition in the Széchenyi István University in Győr. The electric vehicle development team – SZEnergy Team – has been participating in Shell Eco-marathon (SEM) since 2008, which is the world largest energy efficiency competition.

Figure 1. SZEmission, the most recent SEM race car of SZEnergy Team 2019

The participating vehicles have different propulsion system compared to pneumatic vehicles, but still has a lot in common. Our development goals are to reduce the vehicle mechanical, aerodynamical and electrical losses. In this article we're presenting special steering mechanism design, which is also applicable in autonomous driving challanges as well. The most recent battery electric vehicle of the SZEnergy Team can be seen in figure 1.

2. GEOMETRY FUNDAMENTALS

One of the most crucial sub-components of these special vehicles is the steering system. The steering geometry design largely influences the vehicle dynamics. Ideal Ackerman geometry could be defined as a state, when the tire's rotational axes are perfectly intersecting the center of the turn. [1]

This ideal geometry can be defined based on the given vehicle data. The SZEmission parameters are shown in table 1.

Table 1. – SZEmission vehicle data

Wheelbase (L)	1300 mm
Front track	1000 mm
Rear track	800 mm
Drag link radius (Ls)	19,5 mm
Kingpin distance (b)	961 mm
Minimal turning radius (rho)	5676 mm

These vehicle data were directly acquired from the 3D CAD models of the car. The abbreviations can be found in figure 2., which clearly shows the main principle of Ackerman steering geometry. Based on the presented kinematics scheme the following equations can be established. The outer wheel angle can be calculated from the vehicle data as follows [2].

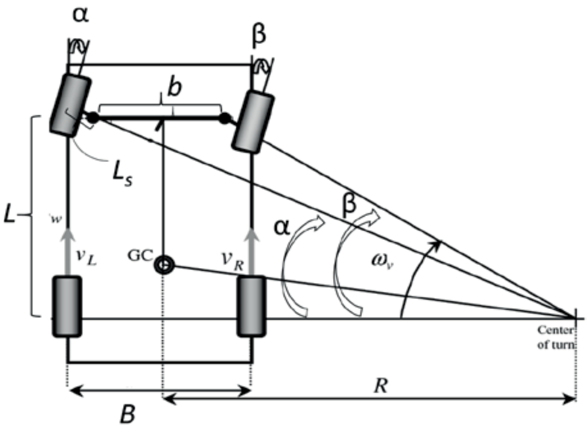

Figure 2. Ackerman steering geometry [2]

$$tg\alpha = \frac{L}{R + \frac{b}{2}} \qquad (1)$$

The inner wheel angle can be determined similarly as the outer.

$$tg\beta = \frac{L}{R - \frac{b}{2}} \qquad (2)$$

The radius of curve depends on the wheel-base and steering angle.

$$R = \frac{L}{tg\alpha} \qquad (3)$$

The vehicle's turning ability relates to the smallest turning radius, which can be defined as the outer front wheel's center distance from the curves center. The following equation describes the minimal turning radius, marked as *rho*.

$$rho = \sqrt{(R + \frac{b}{2})^2 + L^2 + l_s} \qquad (4)$$

(4)

In case of Ackerman steering geometry, the inner and outer wheels are in different angle position. The inner wheel angle *(β)* is always greater than the outer wheel angle *(α),* when we're observing normal Ackerman geometry. SZEmission's optimal wheel angles depending on the actual turning radius can be seen in figure 3.

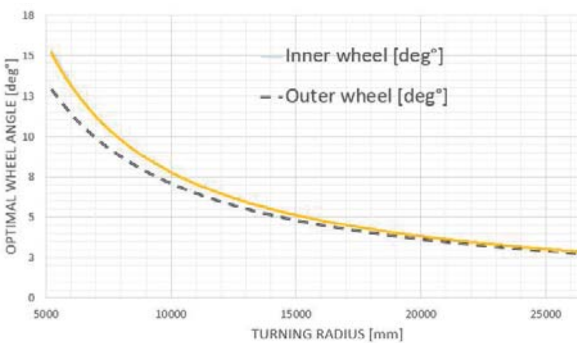

Figure 3. Characteristics of SZEmission's wheel angle

There are methods when the wheels are turned by computer controlled actuators to their ideal position. These steer-by-wire methods have no direct physical connection between the steering wheel and the rotating wheel, that's why safety issues arise related to those systems.

Based on SEM regulations, we have to design a physically connected steering system, which has the lowest possible backlash and closest to ideal Ackerman geometry in our operating range. The situation become more complex if we examine the parts deformation under operating loads and variable suspension geometry design with different loadings. In this article we're designing unique steering mechanism, which is using optimal spiral path calculated from vehicle data with our method to guide the wheels according to ideal Ackerman geometry. The calculations were carried out using simplified rigid suspension kinematic model. [2]

3. OPTIMAL SPIRAL PATH METHOD

In order to apply the method, first we have to create the movement function of steering shaft position compared to the rotation of the knuckle. The function's directional correlation is shown in the figure 4.

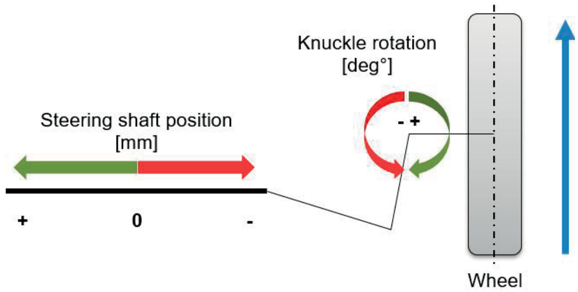

Figure 4. Directional correlation in our movement function

Further on, we also used the previously presented vehicle data table with adding other geometric data such as base radius of our spiral geometry. From CAD models we have to acquire the appropriate data pairs of the ideal Ackerman angle and the translational position change of our steering shaft.

Table 2. – Abbreviation for optimal spiral path calculation

Base radius for spiral path	r_{base}
Steering shaft position – x direction	x_{shaft}
Optimal wheel angle	λ

The value of base radius depends on the steering torque, drag link radius and the expected turning radius. Due to the modification of the base radius value we can achieve different force ratios in the steering mechanism. The listed parameters and their abbreviations can be seen in table 2.

The optimal spiral path x and y coordinates can be determined by applying the following geometry transformation.

$$x = (r_{base} + x_{shaft}) * \cos(\lambda) \tag{5}$$

$$y = (r_{base} + x_{shaft}) * \sin(\lambda) \tag{6}$$

This x and y coordinates exactly describe the optimal spiral path. The rotating spiral path guides the steering shaft translational movement which keeps the angles of the steered wheels in the position when their rotational axis perfectly intersect the center of the turning radius, achieving the ideal Ackerman geometry. Due to this mechanism we can notice remarkable decrease in the suspension losses, particularly in curves. The spiral path x and y coordinates are illustrated in figure 5. The points are connected with spline interpolation and the base radius is also shown in the figure.

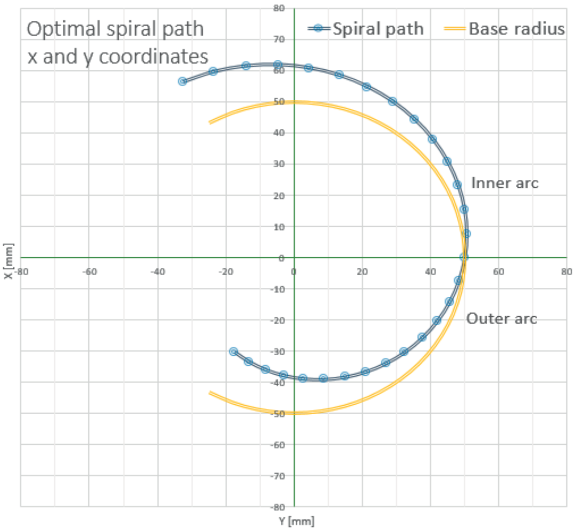

Figure 5. Spiral path x-y coordinates

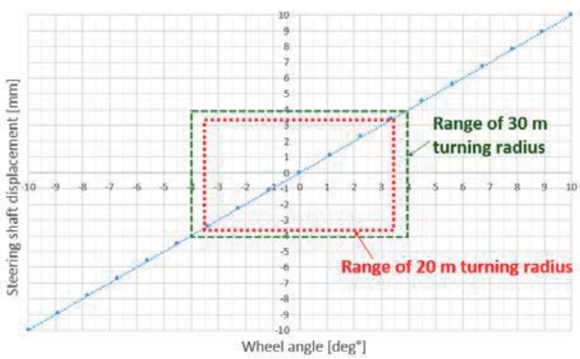

Figure 6. Relation between steering shaft displacement and wheel angle

The relation between the shaft displacement and the wheel angle is almost perfectly linear as it is shown in the figure 6.

Based on the calculations we graphically display two rectangles in figure 6. These rectangles shows the range of 20 m turning radius and 30 m turning radius from inside to outside respectively. These are the most commonly measured turning radius values in a SEM race track.

It's noticeable that the steering system mostly needs to cover only 4 mm shaft displacement in its operating conditions.

4. STEERING MECHANISM DESIGN IN CAD ENVIRONMENT

CAD design implementation immediately started after the calculations of the optimal spiral path were carried out. The most important part is the spiral disk. This part was design in two different concepts.

In one hand, the spiral path equation is drawn in CAD with using spline tools and this curve is offset with given width. On two sides of resulted spline there are urethane molded bearings, which are connected to the steering rods. In this case the spiral guides the rods through rolling bearings.

On the other hand, the disk can be designed with doubling the extruded curve to create a hollow between the curves. The inside of the hollow could guide a properly fitted shaft. In both case the rotation of the spiral disk is transformed into the steering rods translation movement. The main difference between these options is the way of friction and the clearance between the parts, which could easily cause undesired backlash in the whole system.

The spiral disk and its shaft has to be supported by bearings from both side. We designed a framed construction, which connects the bearing housings. The frame and the housings are fitted with positioning pins to keep the shaft in its designed place.

The steering rods are connected to the spiral disk through adjustable arms. The rods are made from hard anodized square sectioned aluminum bar. The steering rods are supported and mounted by plastics crossbars. The whole construction is placed in U-profile aluminum frame, which unites parts and increases the system rigidity. The figure 7. shows the CAD assembly of the designed spiral path steering.

SZEnergy Team is making great effort in autonomous driving development, that's why an autonomous steering extension was designed to the presented steering system, illustrated in figure

8. The steering is driven by a DC motor through timing belt connection. The motor controller gets position information from linear sensor placed on the steering rod. The autonomous extension is separate unit, so it could easily attached to the steering system.

Figure 7. CAD assembly of the steering system

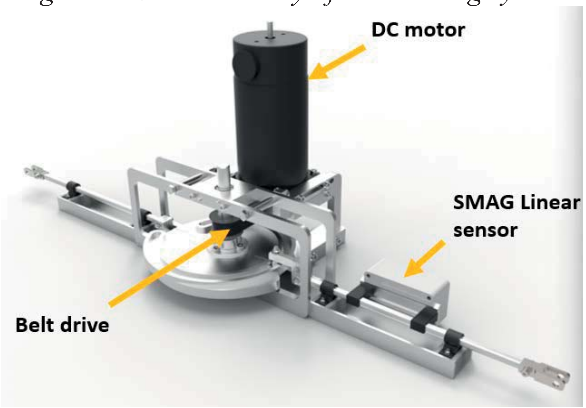

Figure 8. Autonomous steering system

5. STATIC STRUCTURAL AND RIGID BODY DYNAMICS SIMULATION IN ANSYS

The CAD design process was supported by plenty of FEM simulations. Traditional static mechanical simulations were made to determine the different dimensions of the presented parts. Main goal was to specify the width of the contacted spiral disk. In case of static simulation we have to regulate the system's degree of freedom (DOF) with constrains and apply loads to the system. We made simulation cases with different sample loads as we had no exact data about the occurring loads. The load cases with different CAD data were matched. The results of these cases were compared to each other to determine the trends.

We used the available basic ANSYS material models such as aluminum and structural steel. Tetrahedron meshing with local mesh tools in the main contact area was chosen. We tried to make mildly detailed mesh to check the models with reasonable accuracy.

Some simulation input were received from CAD design such as the geometry data of the bearings. These commercial products have exact size limitation, so we have to adjust our parts dimension to them. We targeted to find dimensions, which give the part near the same strength as the bearings. It means that we didn't examined exact numeric results just compared the different simulation cases. One of the static structural simulations isoline results can be seen in figure 9.

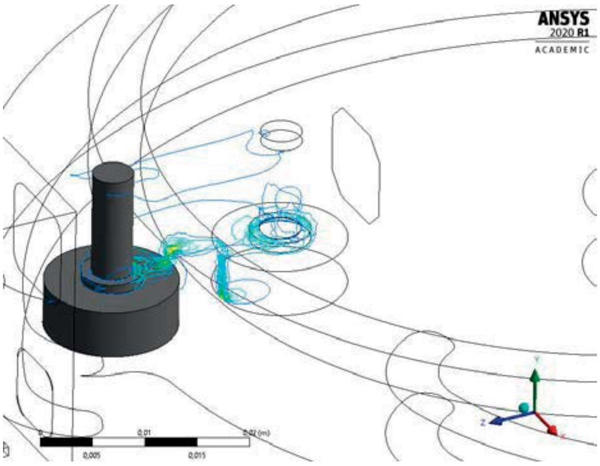

Figure 9. Static structure isoline result

Evaluating the simulation results we agreed on 5 mm spiral curve width, which satisfy both mechanical and manufacturing demands. The manufactured and pre-assembled spiral disk can be seen in figure 10.

Figure 10. Pre-assembled spiral disk

Rigid body dynamics module of ANSYS was also used to receive approximate data of the system movement characteristics. Firstly, we had to modify the simulation constrains and adjust the system's DOF to create a periodically repeated movement. The main simulation constrains and joints are illustrated with the simplified model in figure 11.

The sample spring load is similarly determined as in the static structural module as the exact loads are unknown yet. The spring load connects the fixed knuckle (simplified as a rectangular body) to the steering rod. The steering rod can move translationally in the supports. The polymer crossbars are also fixed to the ground. No separation contact is applied between the roller bearing's cylindrical surface and the designed spiral path surface, the figure 12 shows that in details.

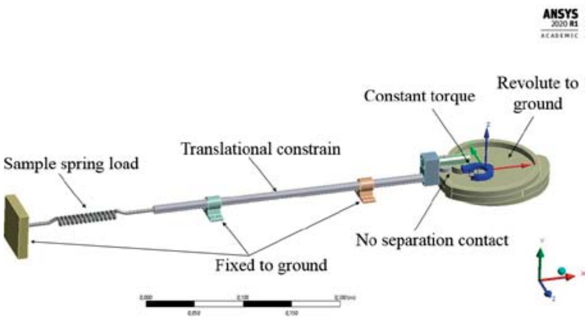

Figure 11. Constrains and joints in ANSYS Rigid Dynamics module

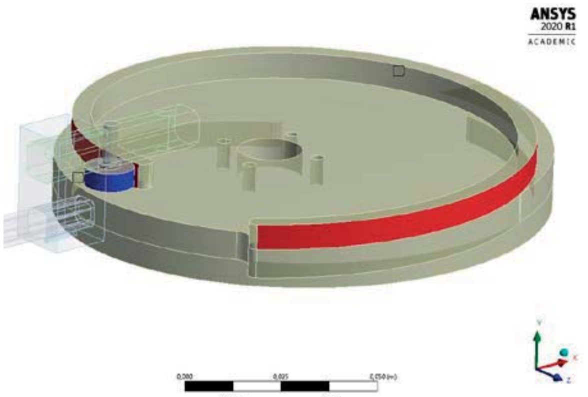

Figure 12. No separation contact on the spiral path

The spiral disk is able to revolute around its local Z axis. This revolute joint is driven with constant torque. The rotation is limited from 0° to 315°, it's the physical limit of the spiral path. The movement starts from the maximum distance from the knuckle. The spring is unloaded at the beginning. The rotating disk moves the steering rod only in the global X direction, the rod compress the spring which generates reaction force in the knuckle. When the angle limit of the spiral disk is reached, reversed rotation starts to take the simulation back to the initial condition. The simulation runs periodically. Joint and position probes were queried to get the movement, velocity (angular and translational) and force results. The comparison of the translational and angular velocity is illustrated in figure 13.

The rotational and translational performance can be also calculated from the queried simulation results as it is shown in figure 14.

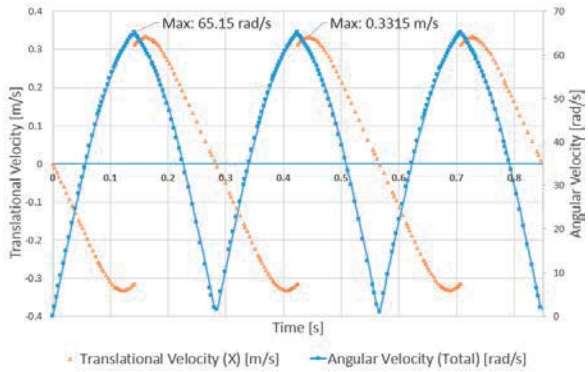

Figure 13. Translational and angular velocity results from ANSYS Rigid dynamics

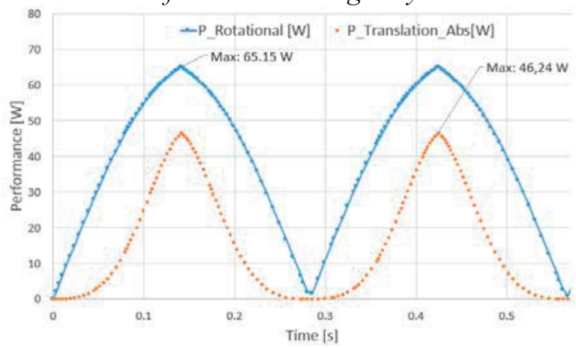

Figure 14. Performance results from ANSYS Rigid dynamics

5. SUMMARY

Special steering mechanism concept based on the calculation of the optimal spiral path was presented in this paper. This spiral path creates unique curves depend on different suspension characteristics. The curves follow the ideal Ackerman angle in every position, which extensively reduce the suspension losses. The main design and simulation principles of the steering systems were also presented.

6. ACKNOWLEDGEMENT

The research presented in this paper was carried out as part of the "Autonomous Vehicle Systems Research related to the Autonomous Vehicle Proving Ground of Zalaegerszeg (EFOP-3.6.2-16-2017-00002)" project in the framework of the New Széchenyi Plan. The completion of this project is funded by the European Union and co-financed by the European Social Fund.

6. BIBLIOGRAPHY

[1] N.W. Troupe.: Optimization of Steering Geometry For Ultra-High-Mileage Vehicles, The Pennsylvania State University, 2011.

[2] K. Hartani, Y. Miloud, A. Miloudi.: Electric Vehicle Stability with Rear Electronic Differential Traction, EFEEA'10 International Symposium on Environment Friendly Energies in Electrical Applications, Ghardaïa, Algeria, 2-4 November 2010.