A NEW CONCEPT OF ELECTRIC ECO-CAR

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ABSTRACT

The problems of mobility in large urban areas pose a challenge to the designers of future transportation systems. Research into ecological systems which often shows a reduction in CO2 emissions close to zero should also be cheap to maintain and user friendly. An increasing concern is the greater mobility of the disabled and elderly which will shortly represent 20-30% of the population. This paper presents the concept of a new electric ECO-car designed within the remit of the scientific and research programme at Warsaw University of Technology. The car is equipped to carry both able-bodied passengers and the disabled. The car uses an integrated drive and brake-by-wire system, as well as powered by lithium-ion batteries and a super capacitor system. During the design process special attention was also paid to ergonomic issues.

Keywords: electric car, disabled people, drive-by-wire, computer simulation, ergonomics

INTRODUCTION

This paper presents a new concept of electric car for four passengers (both for able-bodied and disabled users). The term ´adjustments for the disabled´ means the vehicle allows disabled people in hand-powered wheelchairs to be able to drive the car (incl. getting into it) as well as transporting a passenger in an electric wheelchair. In the presented version of the car, the disabled are understood as those with disabilities of the lower limbs. The car has a top speed of 50 km/h and is intended for use in enclosed areas where conventional vehicle traffic is limited or forbidden. It is designed to function as part of a car rental system such as *Park and Ride* schemes. Therefore, such infrastructure problems such as rental and battery-charging systems, etc. must be addressed. This article focuses on selected issues related to these technical aspects. Figure 1 illustrates some of the solutions used in the car's design.

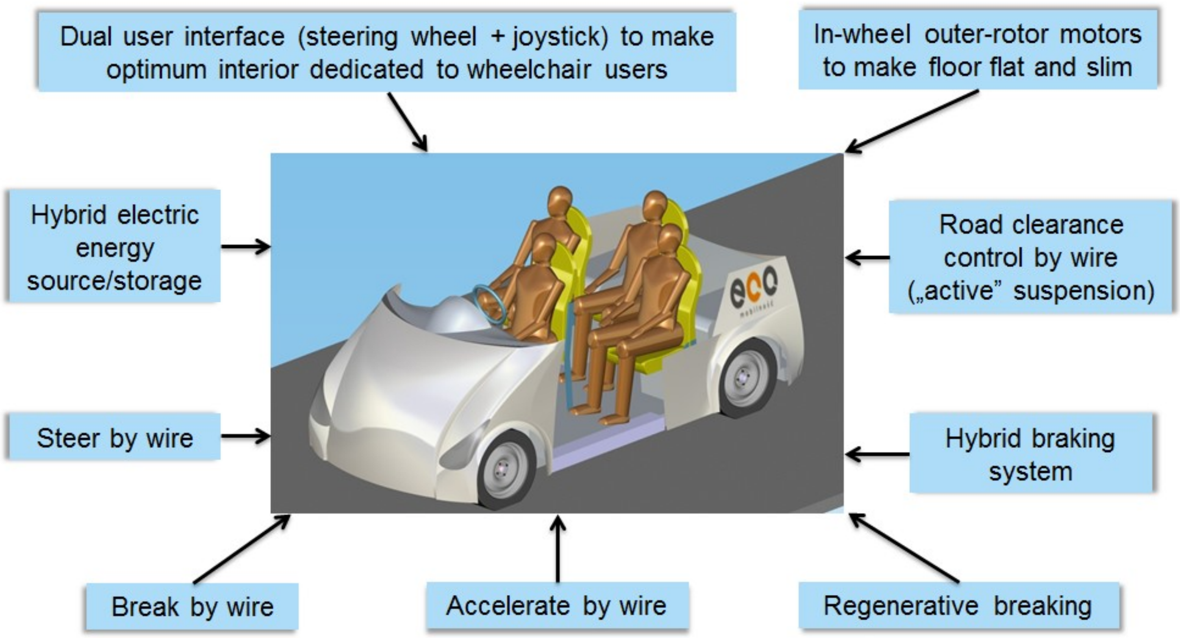

Figure 1 - General characteristics of the presented car

Due to this paper's limited scope, it will focus on the following issues:
- vehicle's general electrical structure
- vehicle's computer simulation (regarding a disable driver)
- ergonomic analysis.

ELECTRIC POWERTRAIN WITH HYBRID ENERGY STORAGE

Lithium-ion batteries are the vehicle's energy source. It is beneficial from several points of view to support the electrochemical battery with ultra-capacitor storage. The usefulness of this approach has been demonstrated e.g. in (Michalczuk et al, 2012). Even state-of-the-art battery technology will be adversely affected in low temperatures and high current cycling. Sub-zero temperatures significantly limit available power and high current cycling drastically shortens a battery's life and as the vehicle is mainly intended for city use then this will have to be addressed. Frequent braking/accelerating cycles put considerable demand on the battery than typical suburban driving and as traffic congestion in city centres increases during rush hour then a typical drive cycle will include many more frequent braking events than standard test cycles. At the same time, regenerative braking assisted by ultra-capacitor storage instead of using lithium batteries is more efficient and results in greater distance covered on one charge. After extensive numerical studies it was decided to hybridise the vehicle's energy storage. The topology of the power converter is shown in 0.

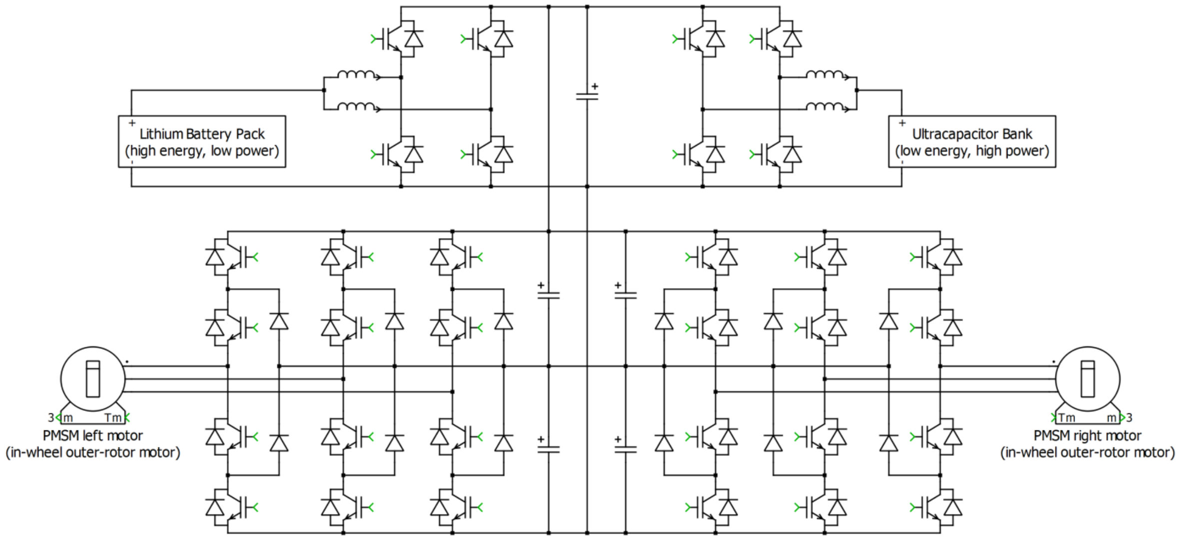

Figure 2 - Topology of the converter

The converter includes two 2-level 2-leg DC/DC interleaved converters connected to the common DC-bus and interfacing both storages (batteries and ultra-capacitors). Two 3-level 3-leg DC/AC drive inverters feed two in-wheel outer-rotor PMSMs (permanent magnet synchronous motors). The number of levels has been chosen as a compromise between the cost and complexity of the system and the quality of the output voltage waveform. Key parameters of the powertrain are presented in (Grzesiak et al., 2012).
The experimental control system will include units for independent speed control, average speed control and no speed control (the driver acting as the speed controller, i.e. deactivated digital control loop for speed).

Steer-by-wire

The car will be equipped with a steer-by-wire (SBW) system. In many standardised tests (e.g. in the double-lane-change maneuver - "moose test") the dynamics of the position control system for the steering motor could affect results. If, e.g. a communication bus is used to command the angular position to the SBW controller, this delay occurs in the open-loop and cannot be compensated. An example of performance deterioration for CAN delays from 5ms up to 100ms is given in (Fang et al., 2008). A continuous model of such a system might appear as in figure 3 if no

CAN communication bus is involved in the feedback paths. To maintain a zero steady state error, a proportional controller is sufficient because of integral action present in the plant.

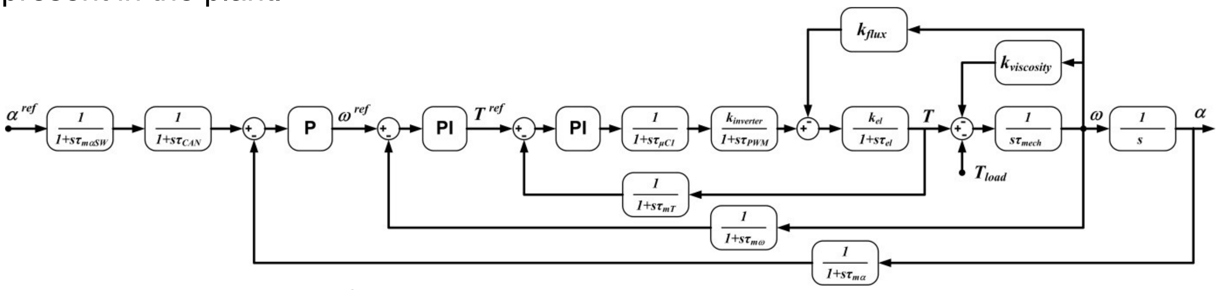

Figure 3 - A continuous model of position control system with delays introduced, inter alia, by communication bus (τCAN), measurement interfaces and signal conditioning (τmT, τm□, τmα), PWM modulator (τPWM) and microcontroller (τμC1) – a steer-by-wire motor.

It is common to modify the basic cascaded control scheme from figure 3 to meet the requirement of precise reference tracking and good disturbance rejection. More elaborated schemes like the one shown in figure 4 are usually employed (Paul, 2005).

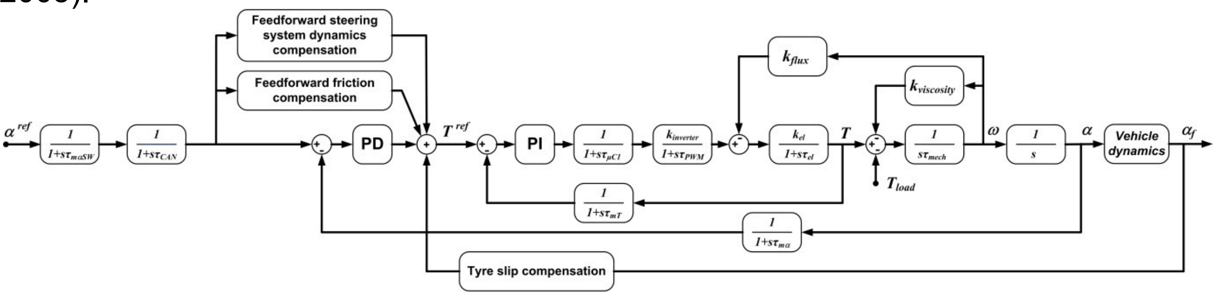

Figure 4 - A continuous model of SBW position control system with delays introduced, inter alia, by communication bus (τCAN), measurement interfaces and signal conditioning (τmT, τmα, τmα), PWM modulator (τPWM) and microcontroller (τμC1) – with feedforward controller and tire slip compensation.

In this paper all delays related to a SBW system has been modeled as single first order inertia with time constant equal to 10ms and accuracy of tracking in transient states is assumed to be within the range of 0.5^0.

SIMULATION STUDIES OF THE DRIVER-VEHICLE-ROAD SYSTEM

This chapter presents simulation studies of the driver-vehicle-road system and the research will address the double lane change maneuver (“moose test”).
Research in the driving process task is divided into the kinematics and dynamics analysis. The purpose of testing the vehicle’s kinematics, prior to testing the its dynamics, is to estimate the minimum velocity values in turning the steering wheel, resulting in a positive test result. The dynamics simulation studies are more important and in the case of our car, their purpose is to select a suitable structure for the electronic differential system of the drive motors as well as either examining the need

to use an anti-roll bar with McPherson strut suspension or the steering force needed to turn the wheels.
Currently, the “moose test” conditions are standardised in most countries but as an example it could be the German standard (FTA, 2009). In literature one can also find publications containing the results of experimental studies on the analysis of the selected driver behaviour during the “moose test”, i.e. an analysis of the angular speed of the steering wheel (Breuer, 1998).
However, the authors of this paper are unaware of any publications describing the results of driving tests carried out for disabled drivers. However, it is known that drivers with lower limb disabilities may have minimal difficulty in gripping the steering wheel. Therefore, simulation studies should determine whether it is reasonable to limit the turning range of the car’s steering wheel. Limiting the turning range results in a lower angular speed in the steering wheel required in the “moose test”. It is assumed that a disabled driver will not be able to turn the steering wheel with the same speed as an able-bodied driver.

Analysis methods of the steering velocity of front wheels in the double lane change maneuver – kinematics analysis

Kinematics studies can be performed with relevant and well-known assumptions that simplify the description of the car’s movement, where the Ackermann’s formula is used. With these assumptions, the analysis of the movement trajectory is purely geometrical and helps to objectively determine the minimum required steering velocity of the wheels in the “moose test”. However, the results obtained do not provide a complete picture of the course of the car’s trajectory. For this vehicle the calculations were performed assuming the parameters: wheelbase L=2.33m the distance of the gravitational centre of the vehicle’s body from the centre of the front wheel axis LPA = 1.521m, maximum steering angle of the front wheels α_{max} = 35 °/s. Road dimensions for the lane change manoeuver correspond with the initial phase of the passing in the Moose Test: AC = 13.5 m and CD = 4m (FTA, 2009, Breuer, 1998). The calculation results of the kinematics analysis are shown in figure 5.

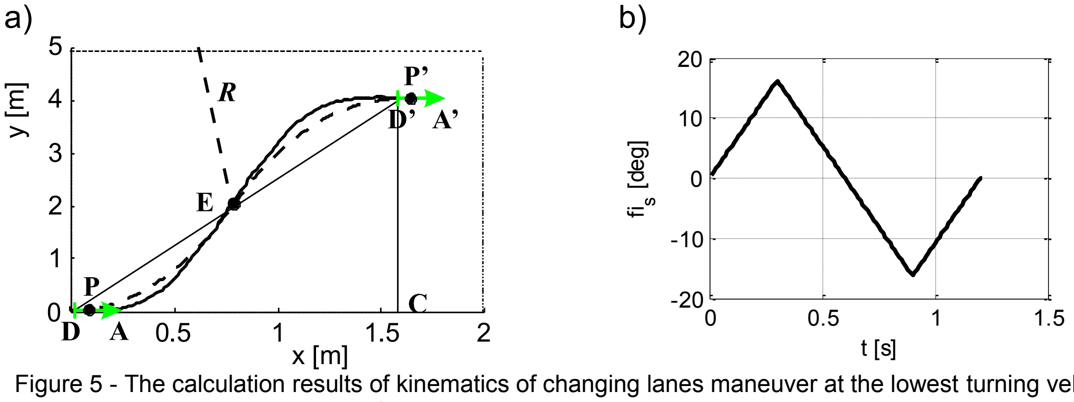

Figure 5 - The calculation results of kinematics of changing lanes maneuver at the lowest turning velocity of the front wheels: a) trajectory, b) turning angle.

The dotted line DD' consists of two arcs with R as a constant radius and E as the midpoint of the two curves. The curvature radius is R=16.7m. The black line describes the trajectory of the vehicle's gravity-point centre achieved when the turning velocity is constant and the ride does not include a movement phase with a constant steering angle. The flow diagram of the steering angle is shown in 5b. The resulting required minimum velocity of turning the front wheel is υ_w = 54 ^{0}/s. It should be noted that under both model-test and real test drive conditions it will be very difficult for the prototype vehicle to perform a lane-changing manoeuver in a way that the manoeuver is perfectly aligned with the lane axis. A number of factors are attributed to this, including driver behaviour, vehicle drift as well as how the steer-by-wire system works. Any deviation in the final manoeuver from its assumed perfect alignment status results in a change in the velocity of the steering wheels. This may also contribute to a negative assessment of the manoeuver performed.

The driver model

In order to retain this paper's more manageable scope, the emphasis here is on the control aspect of the human driver and its subsequent conceptual or computer-based modelling. The function of the steering angle adjuster is performed by the "driver model" and its structure is shown in figure 6. The input value is a set value of the vehicle wheels' steering angle, resulting from an analysis of the geometric manoeuver (figure 5). The adjustable steering angle relative to the chassis is subject to a time delay due to complicated human reactions, which according to various authors could include; the length of time of the signal's reception or a decision-making time lapse coupled with the motor's reaction time length (Grabarek, 2002). In the presented model, the decision-making duration is described indirectly as the distance travelled from the time of passing through the set point of beginning the manoeuver (point A – figure 5) until the beginning of the reaction (therefore, there is no time delay in the structure of the driver model). Whereas the time of motility (turning the steering wheel through a certain angle) is related to the maximum steering wheel turning velocity υ_k. According to studies (Breuer, 1998) the velocity of turning the steering wheel for the tests carried out in driving simulators produces values of 300 - 1200 ^{0}/s ranges, when the test drive only achieves 100 – 300 ^{0}/s. This limitation may be included in the Rate Limiter block. An inertial element (block of First Order Inertia) describes the dynamics of the steer-by-wire system. The Saturation block describes the design limitations of the adjustable steering angle of vehicle's front wheels. The *PID controller* block is used for modelling the driver's reaction to any deviation in the car's trajectory from the desired "perfect" trajectory – figure 6. The task of the block is to steer the vehicle back to the desired trajectory.

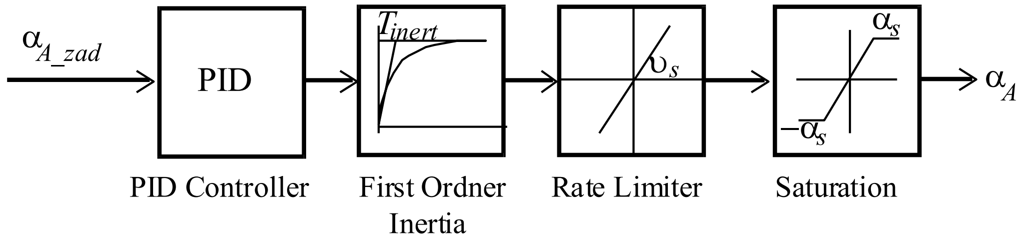

Figure 6 - The model structure of the steering system
Vehicle dynamics simulation during the double lane change maneuver - “moose test”

The dynamic model of the examined object consists of partial models: the mechanical system, the electronic differential system for controlling the rotational velocity of the drive wheels, steer-by-wire system to change the driving direction and the steering system (the model described above).
The mechanical system model is developed in accordance with the principles of MBS (Blundell, 2004). The model has been developed in a Matlab–SimMechanics environment. The structure of the model is described with 0 : a) a system of rigid bodies, b) McPherson strut suspension (front). Figure 4b with indicated: C1-C2-S1-F – rocker arm (red), H-L-F-K – wheel carrier (blue), G, H – McPherson strut (light blue), M-K – steering gear bar (magenta). S1-S2 – anti-roll bar (green line).

a) b)

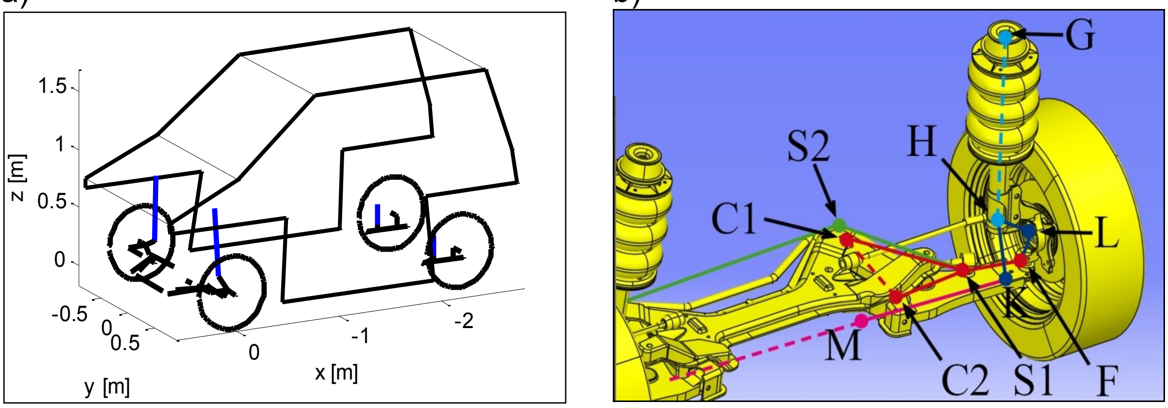

Figure 7 - Model structure of a) system of rigid bodies of a car model, b) McPherson strut suspension

To present the dynamic properties of the tyres the TNO Delft Tyre package has been used (TNO Automotive, 2008) with the tyre parameters defined with the library file 'TNO_car205_60R15.tir'.
The solution of installing an electronic differential of propulsion motors (Gair, 2004), greatly affects the dynamic properties of the vehicle. Preliminary studies of dynamics simulations of the car model have shown that it is worth taking into account the following structures of motor regulatory systems: operating on speed difference for speed control or with the regulator of an average velocity in point A of the front axle of the car. The structures of the regulating systems are shown in figure 8.
In the control system shown in figure 8, the motor dynamics is described with the first-order inertial element assuming that the time constant of the engine response is $T_{inertial\_s}$ = 5 ms. Visual simulation of the car model driving along the “moose test”

route is shown in Figure 9. The vehicle is estimated to be travelling at a constant driving velocity v_{sA} = 50 km/hr, with the propulsion system operating. Simulation results are presented in figure 10.

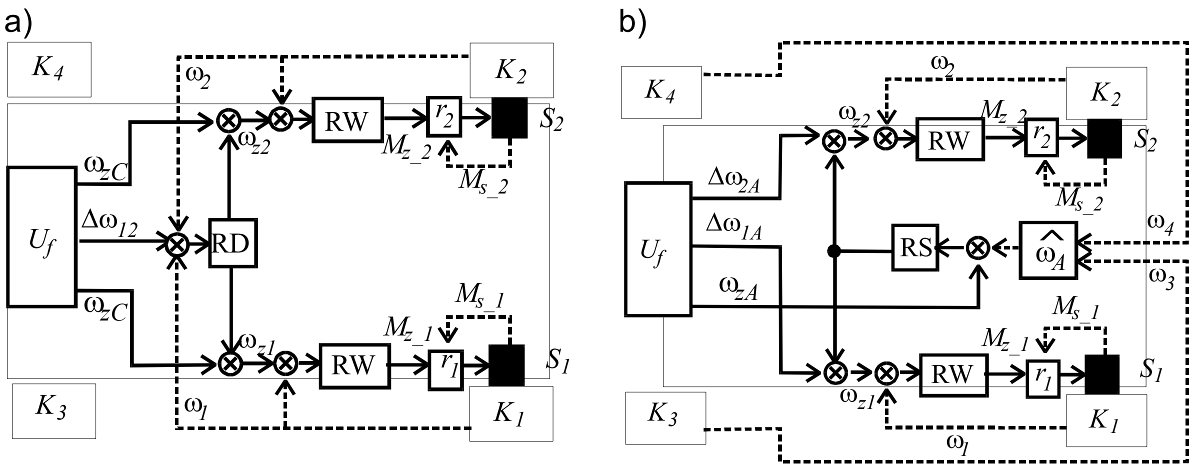

Figure 8 - Block diagrams of regulating systems with velocity setting considered for applications in the designed ECO car: a) with the infliction of a velocity difference, d) with the regulator of an average velocity in point A of the front axle of the car. Marked: ***RW*** – velocity control regulator, ***RD*** – velocity difference regulator, ***RS*** – average velocity regulator, S_1, S_2 – electric traction motors, r_1, r_2 – regulators of traction motors, ω_{zA} – the preset angular velocity of the replacement wheel at the point A of the front axle center, $\Delta\omega_{1A}$, $\Delta\omega_{2A}$ – the preset difference of angular velocity between the replacement wheel and the first wheel, - and the second wheel, ω_{z1}, ω_{z2} – preset angular velocity of motors, $\Delta\omega_{12}$ – preset velocity difference.

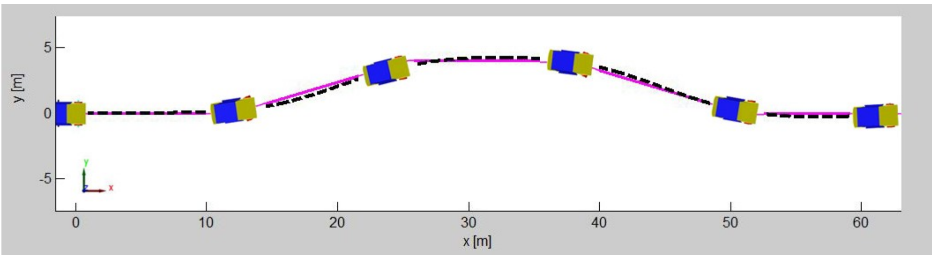

Figure 9 - Visualization of simulation. The positions of the car for the moments 0, 1, 2, 3, 4, 5 s.

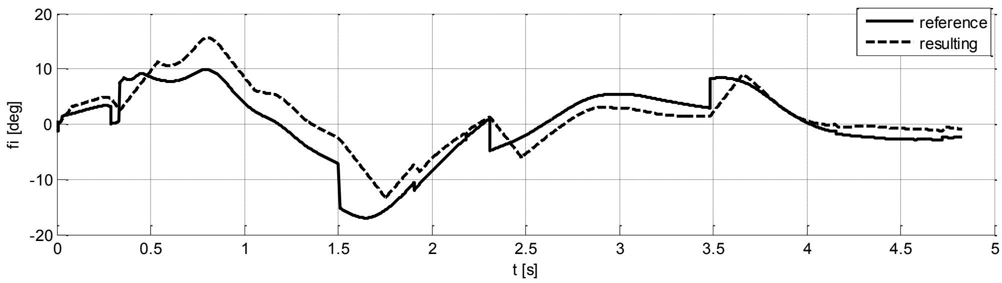

Figure 10 - The results of driving simulation in the “moose test”. Front wheel steering angles of the driver model: Input (“reference”) and output (“resulting”)

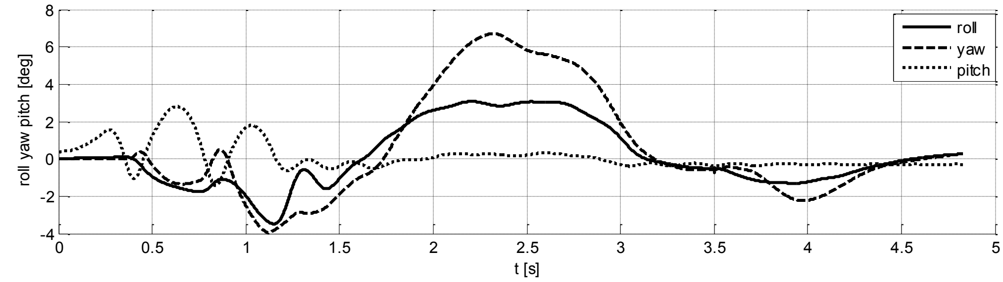

Figure 11 - The results of driving simulation in the “moose test”. Tilt angle of body chassis in the local coordinate system: roll – yaw - pitch

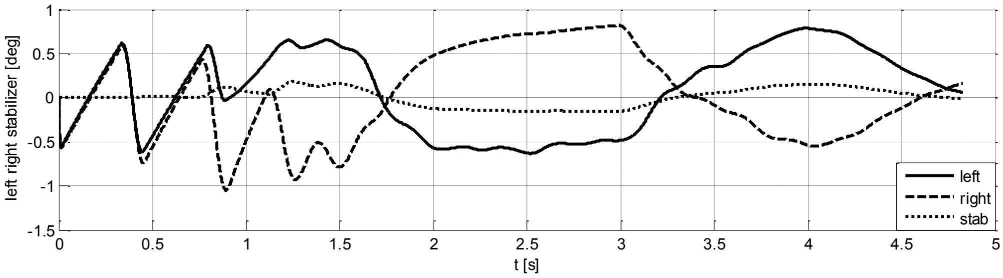

Figure 12 - The results of driving simulation in the “moose test”. The deflection angle of control arms and the twist angle of the anti-roll bar.

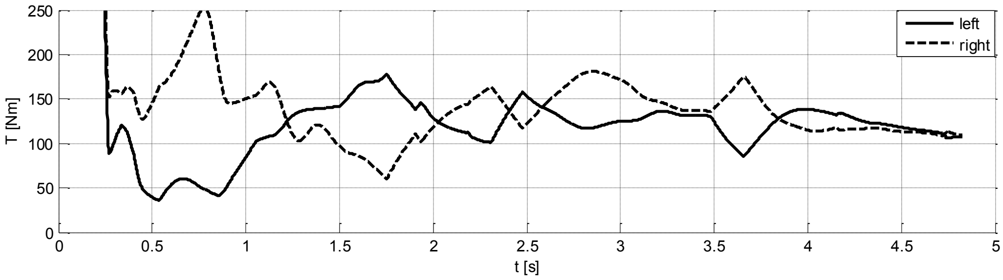

Figure 13 - The results of driving simulation in the “moose test”. Motor torques.

Figure 11 shows the deflection angle of vehicle’s chassis. The yaw angle is the resultant angle in the drifting of the vehicle’s centre of gravity relative to the OZ axis of the local chassis system. In order to demonstrate the workflow of the McPherson strut suspension, which is shown in figure 7, a deflection angle of rocker arms and a steering angle of the stabiliser have been determined. Figure 12 shows the simulation test results of the variables. At the start of the simulation a galloping motion of the car model occurs, which is related to the inaccurate specified initial conditions of the simulation. It can also be observed that when the car makes a turn, the control arm deflections are compensated by a twist of the compliant element of the anti-roll bar. Figure 13 shows the applied torque of the propulsion system traction motors Ms1 and Ms2, resulting from the operation of the adjustment system (figure 8a). As a result of this adjustment, the outer wheels are driven by a torque higher than the internal wheels (the first turn is to the left).

Analysis of the required steering velocity of the front wheels in the moose test

The above simulation results show that the vehicle is able to pass the “moose test”. As stated above, it is assumed that the driver may, to some degree, have difficulties gripping the steering wheel, thus the analysis results of the turning velocity of the wheels should provide an answer to the question of how to adjust the maximum turning range of the steering wheel in the vehicle.

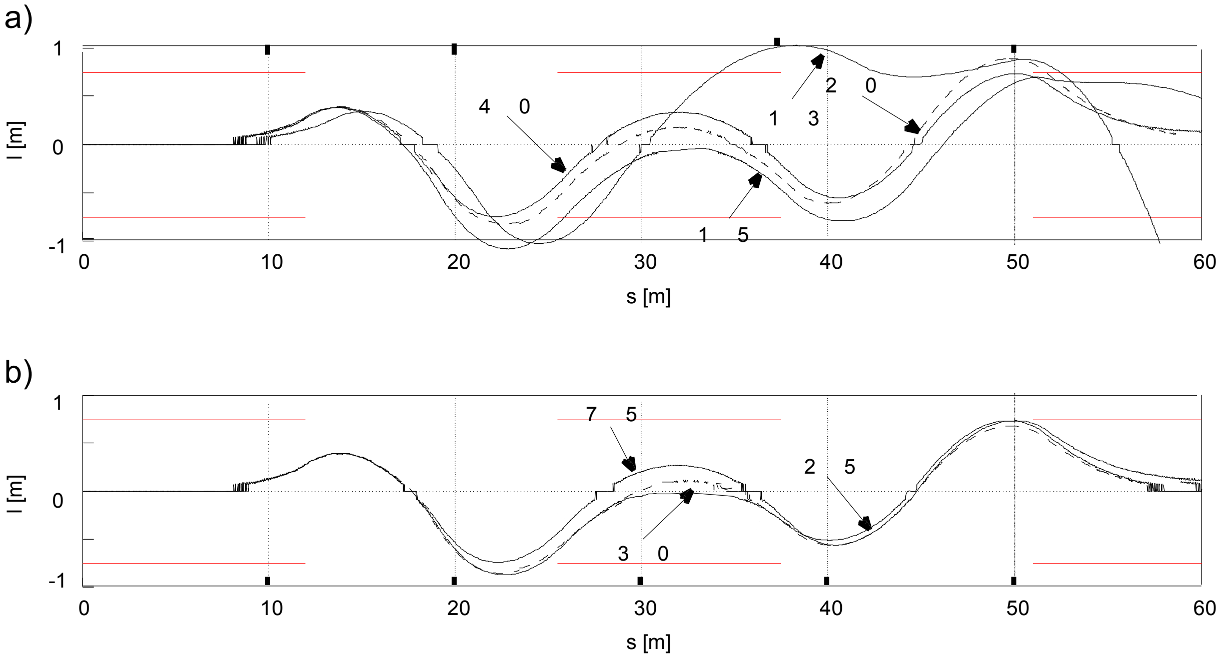

Figure 14 - The results of driving simulation in the “moose test”. Distance of the gravity center point of the car model from the road axis, for different preset values of the maximum speed of turning the steering wheel.

Figure 14 shows the vehicle’s trajectory simulation results of the dynamics model. The “13” line indicates the distance of the vehicle’s central gravitational point from the road axis, in the case of limitation to the maximum velocity of turning the front wheels to the value υ_s = 13 ^{0}/s. In this case it failed to keep the vehicle within the required distance range from the road axis. The “15” line shows the velocity υ_s = 15 ^{0}/s. At this turning velocity, the trajectory model exceeds the assumed lane width. This situation is also repeated for higher velocities υ_s = 20 ^{0}/s (“20”), υ_s = 25 ^{0}/s (“25”), υ_s = 30 ^{0}/s (“30”). In two cases, the model driven by a virtual driver fits within the demarcated lane, for the velocity υ_s = 40^0/s – “40” and υ_s = 75 ^{0}/s – “75”. The results show that for a steering velocity of at least υ_s = 40 ^{0}/s is required the positive test performance. For a maximum steering angle of the front wheels α_{max} = 35 ^{0}/s, to be achieved through a 1.25 rotation of the steering wheel, the wheel would have to be turned at a velocity of at least υ_k = 514 ^{0}/s, and for a 0.5 rotation, just 206 ^{0}/s.

ECO-CAR FUNCTIONALITY AND ERGONOMICS

The important issues of the Eco-Car design process are the solutions sought regarding its functionality and ergonomics, particularly as the vehicle should be user-friendly and efficient for both able-bodied and disabled (incl. wheelchair) users. The targeted user factor pre-empted and influenced the vehicle's ergonomic design. This group is prevalent in developed societies with a diversified age structure and a significant elderly population. The scope of physical disabilities is not only the result of disease or injury (which is irrespective of age) but also due to a wide range of diseases specifically affecting older people. It was also assumed that a person with limited mobility will drive from the vantage position of their own wheelchair. This factor necessitated adequate space for boarding and positioning the wheelchair within the cabin. The vehicle is also equipped with a special securing system to prevent the wheelchair from moving around inside the cabin. Allowing wheelchair users to be drivers of the vehicle required a specific system for folding and manoeuvring the front seat when necessary. The mechanism and accompanying microprocessor control system is an important innovation for the authors of this paper. A wheelchair can enter the vehicle either from the rear (with the aid of a special ramp) or from either side of the cabin. Using an air suspension system the car's floor plate is lowered to 12cm above the ground which allows easy and trouble-free access according to tests conducted with wheelchair users and who are further assisted with ergonomically located handles. For electric wheelchair users, access to the cabin is only through the rear of the car. Figure 15 presents the full access functionality of the vehicle.

A characteristic feature that distinguishes the Eco-Car's design from that of typical passenger cars is seat height. When designing the cabin space the needs of users of reduced height (C_5 women in the driver's seat and C_{95} men sitting in the highest wheelchairs) were fully taken into account. In accordance with the legal standards, the height of the wheelchair ranges from 40cm to 52cm. The height of the driver's seat is set to 40cm resulting in maximum difference in seat and wheelchair height of 12cm. The cabin is designed in such a way that the driver's seat has no longitudinal adjustment and due to its height, neither any vertical adjustment. Both a wheelchair-user driver and an able-bodied driver in the driver's seat occupy the same area in the cabin's spatial structure. This is also shown in the panel design which, with both longitudinal and vertical adjustment features, can be set to facilitate drivers of all sizes. Another issue is the construction of wheelchairs with low backs. When travelling by car, and especially when driving requires the head and back to be supported. What the wheelchair does not provide for the disabled driver, must be accommodated another way, and in the case of the Eco-Car, the backrest from the folded driver's seat is used. It also comes with seatbelts which can be used to secure a wheelchair.

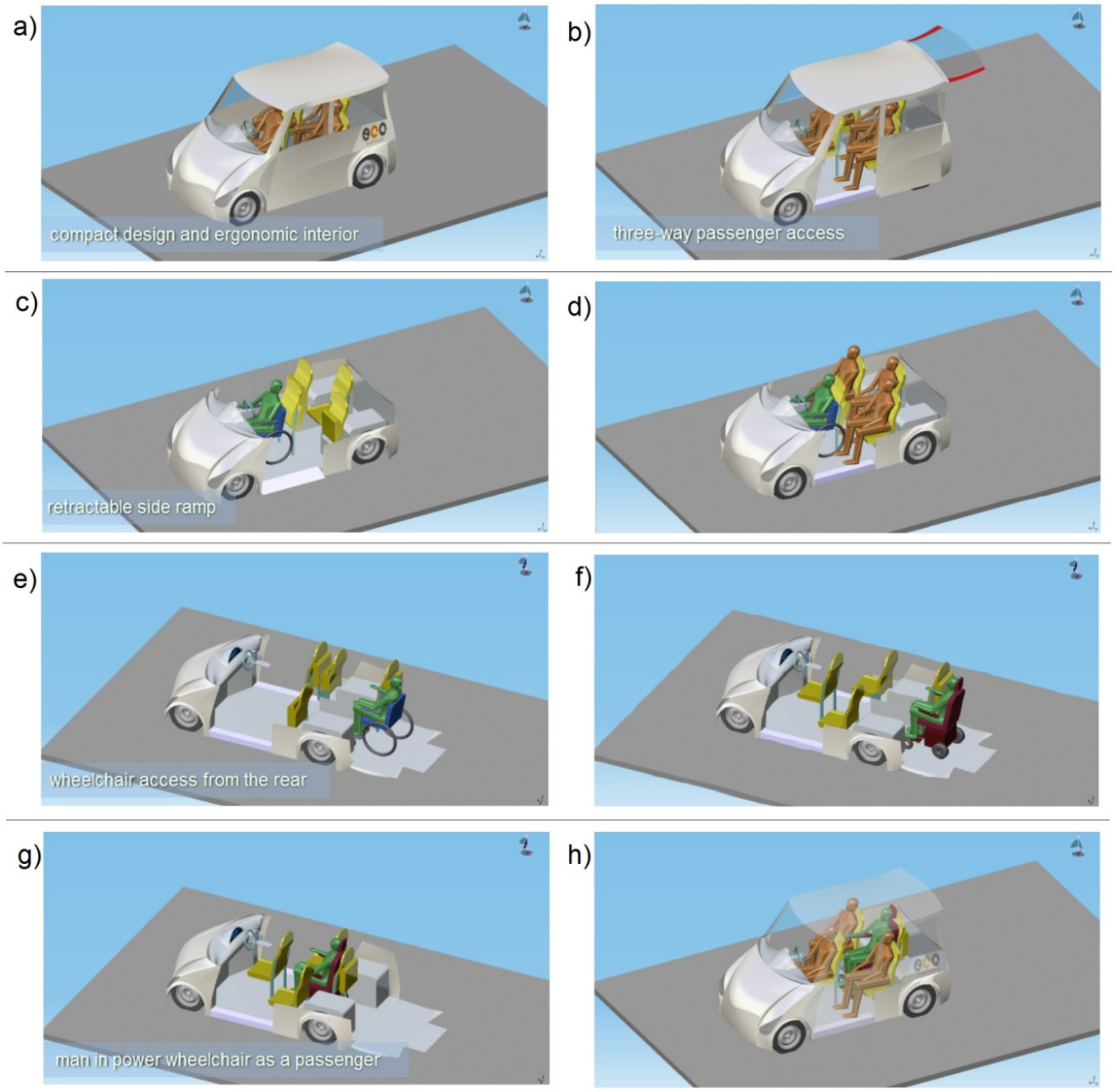

Figure 15 - Visualisation of eco-car functionality: a) general vehicle structure, b) possibilities of entering to the vehicle, c), d) a placement of disabled person (driver) and location of four people (including one disabled person), e), f) entering of a manual wheelchair and an electric wheelchair through the rear ramp, g), h) locating one passenger in the electric wheelchair and the remaining three non-disabled passengers.

This additional use required the backrest to be specially designed to enable the wheelchair to move and be accurately aligned despite the wheelchair's projecting shelf and handles. Figure 16 shows the position of the driver in a wheelchair, supported with the folded seat for a non-disabled driver.

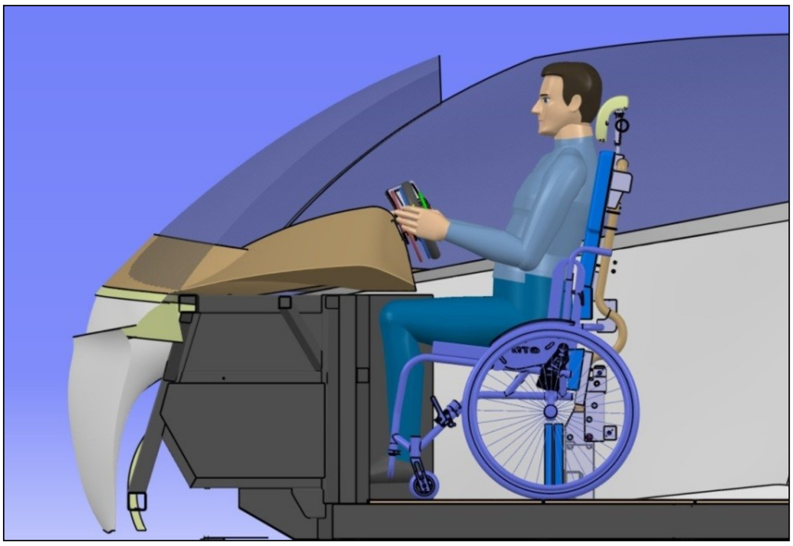

Figure 16 - Location of a driver in an active wheelchair

The dashboard (encasing steering, signaling and control equipment) can be freely adjusted with a remote control by any user occupying the driver's seat, either able-bodied or I a wheelchair. The steering wheel becomes an important issue as its construction must allow the driver to perform all essential control work using only the upper limbs while its handling must be equally intuitive for all the users. Figure 17 shows one of the design versions of the multifunctional steering wheel.

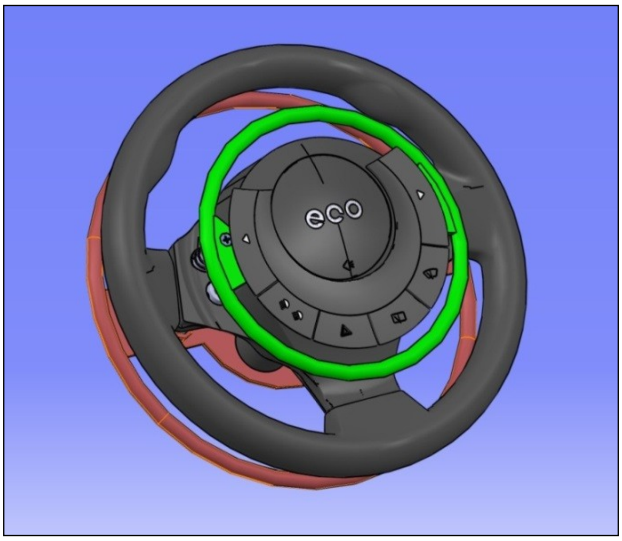

Figure 17 - Circular multifunctional steering wheel

Figure 18 presents a visualisation of one of the dashboard/steering wheel positions based on user requirements.

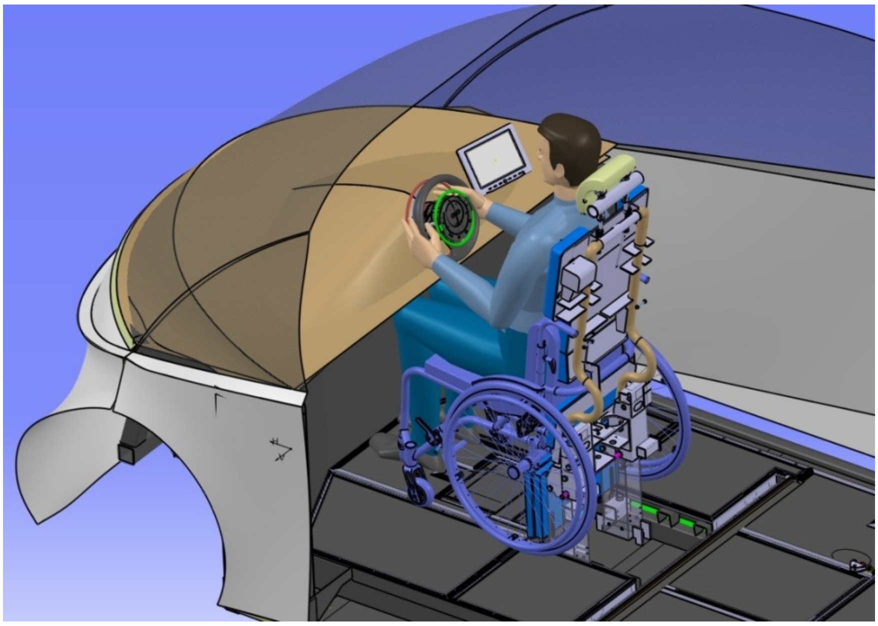

Figure 18 - Visualisation of panel position with the steering wheel, adjusted to the driver's anthropometric characteristic

CONCLUSIONS

This paper presents a proposal for an innovative electric vehicle solution, adapted for use by both able-bodied and disabled people in wheelchairs. The proposed solution is currently at the prototype construction stage. Some of the elements: drive-by-wire, electronic steering wheel system and adjustable dashboard, electric drive system using a lithium-ion battery and super-capacitor have already successfully passed laboratory tests. The inventors are convinced this proposed solution can be applied in future environmentally-friendly and aware cities. Cycle rental schemes have already proved phenomenally popular and can be easily complimented by an equivalent eco-car rental option, which is both human and very green.

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