Motion Analysis of a Wheeled Mobile Driving Simulator for Urban Traffic Situations

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Abstract *– The simulation of urban traffic has higher system requirements than that of motorway traffic for driving simulators. The motion information characteristic of urban traffic easily exceeds the motion envelope available for state of the art driving simulators. Augmenting translational motion by rail systems results in increasing moving mass and insufficient system dynamics. The presented wheeled mobile driving simulator (WMDS) moves on powered and active steerable wheels and solves the announced core problems. A feasibility analysis is conducted concerning energy, power and friction demand for a real world test drive on an urban traffic circuit. The presented results prove that a friction limited propulsion system, like the WMDS concept is suitable for performing driving simulation. Energy and power demand are also feasible with regard to state of the art battery technology. The promising new approach allows the combination of a large motion envelope with high system dynamics into a high fidelity driving simulator (DS).*

Key words: *"ideal" motion cueing, wheeled mobile driving simulator, urban traffic simulation.*

Introduction

Driving simulators (DS) are an established development tool in the automotive industry. The versatile areas of application all profit from a high degree of reproducibility and safety. One area of application is the development and evaluation of Advanced Driver Assistance Systems (ADAS). For the analysis of safety critical situations concerning driver behavior and human machine interaction, DS constitute the only adequate tools. For the last decade, the development of ADAS was mainly focused on assistance systems for motorway traffic. ADAS such as, Adaptive Cruise Control or Lane Departure Warning and Lane Keeping Support were developed. Advancements in technology combined with gained knowledge of the development of ADAS led to an increase in effort with focus on urban traffic situations. The main reasons for urban traffic accidents are human errors in common traffic situations such as, turning, U-turning, drive-away and reversing [DES1], [DES2], [DES3]. The upcoming demands for ADAS, with respect to urban traffic situations, result in increasing DS requirements, concerning motion envelope and system dynamics. The increasing requirements arise from the decreased velocity level in urban traffic situations, resulting in higher traction forces. Hence, the average magnitude of the horizontal acceleration vector is higher for urban traffic than for motorway traffic. In addition to the addressed increase of dynamic behavior, urban traffic is characterized by more complex interaction between the driver and his surroundings, including pedestrians, crossing vehicles, traffic signs/lights and lighting conditions. This fact requires further tasks for urban traffic simulation which are not addressed in this paper. DS are usually based on hexapod systems that provide 6 degrees of freedom (DOF). Real world vehicle motion also consists of 6 DOF, however surge (X-direction), sway (Y-direction) and yaw (rotation about Z-axis, ψ) have unlimited stroke, whereas pitch (rotation about Y-axis), roll (rotation about X-axis) and heave (Z-direction) show high frequent behavior with strongly limited stroke. The motion simulation of the unlimited vehicle surge, sway and yaw easily exceeds the physical limits of hexapod systems, imposed by the workspace. On the contrary, the limited pitch, roll and heave motion are adequately simulatable by hexapod systems. In order to fulfill the increasing requirements caused by urban traffic simulation, supplementary actuated subsystems have to be added to the hexapod — additional DOF are the consequence. Modern-day DS show up to 10 DOF and have reached remarkable quality in simulating real world driving experiences. These improvements incur great costs to system complexity and increasing moving mass of about 80 t [Cla1]. Along with the redundant DOF, the main reason for high moving mass results from the rail systems used to connect the tilt system to the ground. This connection is necessary to resolve the strongly limited translational motion envelope of hexapods. As a result, a linkage between moving mass and motion envelope is caused, thus, limiting motion envelope and system dynamics.

Alternative Concept

General Idea

Wheeled Mobile Driving Simulators (WMDS) solve the core problem of the linkage between moving mass and motion envelope. The main idea is based on the assumption that a system, whose propulsion is limited by friction forces, must be capable of simulating the dynamics of vehicles that are also limited by tire friction forces. The idea of a WMDS is addressed by a patent held by BMW AG [Don1] and a reference by Slob [Slo1]. The references do not deal with feasibility of WMDS. The most important questions are not yet answered and the basic assumption must still be analyzed. In order to analyze the basic assumption, the expected mass reduction and the electric power supply, the following system assembly is considered (Fig. 1).

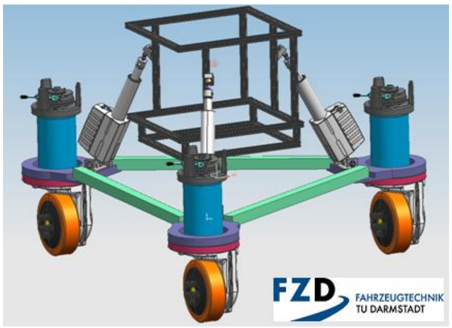

Fig. 1. New approach - WMDS assembly

The presented work is based on a specific assembly of a WMDS showing two subsystems – a wheeled mobile platform and a tilt system (Fig. 1). Each subsystem provides 3 DOF. The wheeled mobile platform includes three powered and active steerable wheels, allowing simulation in 3 unlimited DOF of vehicle motion: surge (*X*), sway (*Y*) and yaw (ψ). On top of the platform a tripod system is mounted providing cabin tilt and heave. Tilting the driver is necessary for two reasons. First, translational acceleration can be simulated by cabin tilt due to false perception of gravity. Second, the limited vehicle motion has to be performed – pitch, roll and heave. The required tilt motions show different characteristics. The tilt motion for acceleration simulation is of low frequent and high stroke, whereas the aforementioned vehicle motions are of high frequent and little stroke. Therefore, a decoupling is intended to reduce the overall moving mass by optimizing the properties of the separate tilt systems. The depicted tilt system in Fig. 1 is optimized to tilt about the driver's head position and is used for rotating the whole cabin in order to simulate translational acceleration only. The vehicle pitch, roll and heave motion is not considered further at the current level of research. It is planned to perform these motions by a separate system that only tilts the driver and his vehicle mockup about a point below the driver seat. The additional system has to be installed between the driver seat and cabin structure and also must be optimized for the high frequent motion showing little stroke.

The design shows system immanent stroke, matching real world vehicle motion. Hence, the system is meant to be a lightweight concept. It seems to be promising for next generation of high fidelity DS that allow simulation of a wide range of traffic situations and high system dynamics.

Wheeled Mobile Driving Simulator Characteristics

The dimensions of the cabin tilt system that are necessary for calculating dynamic actuator load are listed in Table 1 and named accordingly to Fig. 2.

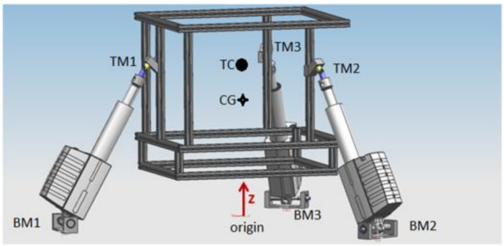

Fig. 2. Design of cabin tilt system

Table 1. Specifications of cabin tilt system

Description	Symbol	Cylindrical coordinates		
		***r* /m**	**φ /rad**	***Z* /m**
Tilt center for initial condition	TC	0	0	2,15
Center of gravity of cabin for initial condition	CG	0	0	1,65
Top-mount of linear actuator for initial position	TM1	1,43	0	2,15
	TM2	1,52	2,09	2,15
	TM3	1,52	4,19	2,15
Bottom-mount of linear actuator for initial position	BM1	2,6	0	0
	BM2	2,6	2,09	0
	BM3	2,6	4,19	0

The estimation of the WMDS mass properties is based on the PSA Peugeot-Citroën DS [Cha1]. The cell weight repartition of the PSA DS is adapted to the WMDS. A worst and best case estimation is assumed. The worst case results mainly from a conservative point of view which utilizes state of the art visual systems. The best case results on an outlook of what could be reached in the near future when using enhanced head mounted displays and virtual reality. The honeycomb structure is the heaviest component of the cabin and has to provide sufficient stiffness for moving its first natural frequency above the operating frequency bandwidth. Hence, its mass decreases with reduced cabin mass and dimensions. Fewer fixation devices and bushings are required due to the tripod system concept. The body shell and vehicle standard equipment is reduced to its minimum. The technological progress in the field of visual systems is expected to compensate leaking impression due to the reduction of the vehicle mockup. Further components listed in Table 2 are transferred unchanged. In total the cabin shows an expected mass of about 310 to 550 kg, depending on the degree of virtualization.

Table 2. Mass repartition of cabin

Major components of cabin	PSA [Cha1] /kg	WMDS /kg	
		worst case	best case
Visual system (projectors etc.) including retro vision	30	30	10
Composite honeycomb structure	250	210	150
Fixation device, bushings	40	30	30
Vehicle cab (body shell)	160	100	0
Vehicle standard equipment (dashboards, seats, etc.)	150	100	40
Acoustic reduction material	30	30	30
Passive force feedback system	30	30	30
Steering wheel feedback system	20	20	20
Cabin Mass	**710**	**550**	**310**

A tripod system, comprising three linear actuators, is used for performing cabin tilt. In order to establish a non-ambiguous relation between actuator stroke and cabin tilt, the top-mounts are designed as cardan joints (2 DOF) while the bottom-mounts show one rotational DOF only. The design leads to a 3 DOF motion system, where translational and rotational motion of the tilt center is coupled by the joint constrains. The unsolicited translational motion for the presented system design shows acceleration values that are about one dimension below human perception thresholds. As a result, the coupled motion does not have to be considered further. The actuators are mounted between the base frame and the cabin structure. The components of the tilt system are supposed to amount to a mass of about 400 kg. The wheeled platform carries the tilt system, a base frame, energy supply and three drive units, each consisting of a tire, a steering motor, a gearbox and a drive motor. The used gearbox descends from forklift trucks and allows both motors to be mounted to the base frame [ZF1]. For this reason, necessary connections (cable, cooling, etc.) are possible without additional effort. In total, the first draft of the wheeled platform is expected to show a mass of about 1250 kg, as shown in Table 3.

Table 3. Mass repartition of wheeled platform

Major components of the wheeled platform	Component mass /kg
Tripod tilt system	400
Base frame	200
Energy supply (electric vehicle battery: 16 kWh) [Kna1]	250
Drive units ($M_{0,\mathrm{Drive}}$ 90 Nm; $M_{0,\mathrm{Steering}}$ 6,5 Nm)	300
Cooling, power electronics, etc.	100
Wheeled platform mass	**1250**

The total system mass of the WMDS, consisting of the cabin, the wheeled platform and a $Q_{.95}$ male person (100 kg) [DIN1], is expected to be in a range between 1660 and 1900 kg, depending on the degree of virtualization.

Methodology

The stated questions concerning feasibility of a WMDS for urban traffic require property identification of urban traffic. Therefore, a representative urban traffic circuit is identified and used for test drives to measure representative motion information (Chapter: *Test Drive*). To answer the feasibility question, detailed information about the WMDS motion is necessary. Hence, the measured target acceleration profiles, from the test drives, have to be transformed into a target trajectory of the WMDS. A trajectory generation is therefore required. Due to the solved linkage between moving mass and motion envelope, an “ideal” motion cueing algorithm (MCA) is used that doesn’t consider translational workspace limitations (Chapter: *Motion cueing Algorithm*).

Based on the moving mass estimation and the generated trajectories, physical relations are applied to determine friction, power and energy demand for high fidelity urban traffic simulation (Chapter: *Calculation of Power, Energy and Friction Demand*).

Implementation

Motion Cueing Algorithm

The “ideal” MCA (Fig. 3) calculates the target DS states that are necessary to perform the target motion information input. The basic idea is to allow as much acceleration simulation from tilt as possible, while considering human perception thresholds for cabin tilt. Therefore, a 2nd order low-pass filter (LPF) is used to determine the change of translational acceleration performed by cabin tilt. The parameterization of the used LPF has to fit human perception thresholds and is tuned for the target acceleration profiles of the discussed test drives on the urban traffic circuit. References show thresholds in a range of 0,05 to 0,2 rad/s for rotational rate and rotational acceleration from 0,03 to 0,1 rad/s² [Dur1], [McC1], [Dob1], [Dag1], [Cla1]. The used thresholds for rotational rate are 0,1 rad/s and 0,1 rad/s² for rotational acceleration. Furthermore, a rate limiter is implemented that limits the LPF output rate if human perception thresholds are harmed.

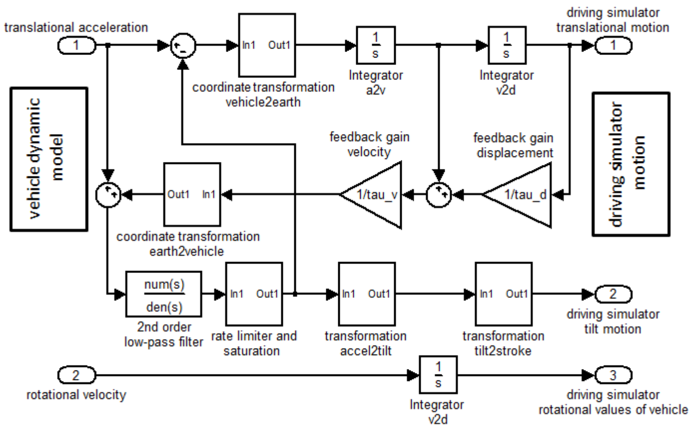

Fig. 3. "Ideal" motion cueing algorithm

The acceleration realized by cabin tilt is subtracted from the target acceleration and leads to the part of acceleration that has to be performed by translational motion. With the described method, an "ideal" motion simulation is reached, considering the usual adverse effects due to tilt. To determine the simulator trajectory, the translational part of acceleration is transformed from vehicle fixed to earth fixed coordinates and then integrated twice. The washout is implemented as a feedback gain of DS velocity ($1/\tau_v$) and displacement from origin ($1/\tau_d$). The stability analysis refers to the loop gain, derived from Eq. 1 and gives a lower boundary for the product of the two feedback gains. The parameter *c* is used to tune the stability margin in view of the Nyquist criterion. According to the analytical limit, the gains are tuned iteratively in order to reduce the required motion envelope for the introduced urban traffic circuit. The physical limitation of cabin tilt is recognized by a saturation block that avoids simulating more than 0,4 *g* translational acceleration by tilt.

$$T = \left(\frac{1+j\omega\tau_d}{(j\omega)^2\tau_d\tau_v}\right)\cdot \mathrm{LPF} \qquad (1)$$

$$T = \left(\frac{1+j\omega\tau_d}{(j\omega)^2\tau_d\tau_v}\right)\frac{1}{1+aj\omega+b(j\omega)^2} = -c, c\le 1, \tau_v \ge 0, \tau_d \ge 0$$

Test Drive

The urban traffic circuit is based on an analysis of 130 randomly chosen traffic situations situated in Darmstadt, Germany. The analysis determines the relative frequency of traffic situation categories: crossroads, T-crossroads, bends, lane splitting and merge. The characteristic properties of these categories are analyzed: lane width, number of lanes, speed limit, curvature, distance of road splitting/merging and priority signs. The most representative ones are assembled to make up the Darmstadt urban traffic circuit, considering the relative frequency. In total, 25 representative traffic situations are selected and fitted into a road map. The connection of the relevant situations is determined by *Google Maps* and is slightly adapted in order to improve practicability. The urban traffic circuit is 21 km long and takes about one hour per turn. In total, four urban test drives are analyzed.

The 229 km long motorway traffic circuit used refers to Filzek [Fil1] and represents German motorways. It takes about two hours per turn. Because of its length, it is split into three parts and evaluated separately. Motorway exits and ramps are not part of the circuit. In total, six parts of the motorway test drives are conducted and evaluated.

The test drives are performed in a VW Golf VI R (equipped car with 2 passengers has 1780 kg mass; 199 kW; all wheel drive) which is equipped with a *GeneSys ADMA G* [Gen1] measurement unit, logging data at a rate of 100 Hz.

All test drives are made during the day but off-hour. Each test drive is driven by a different driver. The driver base is not representative of the general public. All drivers are familiar with the urban traffic circuit, resulting in more dynamic driving behaviour and leading to higher requirements for the conducted feasibility analysis, than a normal driver would cause.

Calculation of Power, Energy and Friction Demand

Energy and Power Demand

At the current level of research, aspects like visual systems and data transfer are not executed. The estimation of energy and power demand is focused on motion of the WMDS only. The considered motions consist of translation, yaw and cabin tilt.

For the translational part, the acceleration and velocity profile of the wheeled platform is required. The necessary information is generated by the introduced MCA. For the best case scenario we assume a maximum recuperation power of 29 kW, as it is used for quick charging batteries of electric vehicles [Kna1] and an efficiency coefficient of 0,8 for battery power to wheel hub torque and vice versa. For the worst case scenario a maximum recuperation power of 4 kW is assumed [Kna1]. The calculations are performed in accordance with Eq. 2 and 3.

$$P_{trans} = a_{trans}\cdot v_{trans}\cdot m_{total} \qquad (2)$$

$$E_{trans} = \int P_{in}\,dt;$$

$$P_{in} = \begin{cases} P_{trans}\cdot\frac{1}{\eta}, P_{trans} > 0; \quad \eta = 0{,}8 \\ P_{trans}\cdot\eta,\ P_{trans,min} \le P_{trans} \le 0 \\ P_{trans,min};\ P_{trans} \le P_{trans,min} = \begin{cases} -4\mathrm{kW,worst\ case} \\ -29\mathrm{kW,best\ case}\end{cases}\end{cases} \qquad (3)$$

The yaw rotation causes rotational power and energy demand. Therefore the moment of inertia (about the vertical z-axis) is required and is assumed to be a homogeneous cylinder, with the estimated system mass from chapter *Wheeled Mobile Driving Simulator Characteristics* (Eq. 4). The rotational acceleration profile is derived from the logged rotational velocity of the test drives. The measured velocity profile is filtered by a 1st order LPF with a frequency limit of 5 Hz in order to focus on typical vehicle dynamics and eliminating noise. The energy demand is calculated according to Eq. 3 by substituting P_{trans} with P_{yaw} from Eq. 5.

$$\Theta_z = \tfrac{1}{2} \cdot m_{total} \cdot r_{drive}^{\;2}, r_{drive} = 2{,}83\text{m} \qquad (4)$$

$$P_{yaw} = \ddot{\psi} \cdot \dot{\psi} \cdot \Theta_z \qquad (5)$$

The power demand for cabin tilt is calculated by the stroke velocity profile and the dynamic axial force of each linear actuator. The energy demand is calculated accordingly to Eq. 3 by substituting P_{trans} with P_{tilt} from Eq. 6. The profiles of the stroke velocities are derived by the used MCA. The part of acceleration simulation that is performed by cabin tilt corresponds to a cabin orientation profile, which is transformed into stroke profiles of each linear actuator. The calculation of dynamic actuator forces is based on the equilibrium of momentum and forces. The degree of concretization does not reach precise actuator level. The power demand for carrying the cabin mass is expected to be about 100 W and is negligible [Moo1]. The method presented does not consider energy required to hold load.

$$P_{tilt_i} = v_{stroke_i} \cdot F_{axial_i} \qquad (6)$$

Friction Demand

Friction demand for a WMDS consists of three parts – target translational acceleration, centripetal acceleration due to cornering and target yaw acceleration. The target translational acceleration is calculated by the MCA.

$$\mu_{trans} = \frac{|a_{trans}|}{g} \qquad (7)$$

The centripetal acceleration is also based on the MCA output but has to be extracted separately from the velocity vector of the WMDS. Therefore, the course angle rate is substituted by yaw and side slip angle rate according to Eq. 8.

$$\mu_{centripetal} = \frac{\left|v \cdot \left(\dot{\psi} + \dot{\beta}\right)\right|}{g} = \frac{|v| \cdot |\dot{v}|}{g} \qquad (8)$$

The friction demand for target yaw acceleration is estimated using Eq. 9 with regard to the estimated moment of inertia from Eq. 4.

$$\mu_{yaw} = \frac{1}{m_{total} \cdot g} \frac{|\ddot{\psi}| \cdot \Theta_z}{r_{drive}}; r_{drive} = 2{,}83\text{m} \qquad (9)$$

The superposition of friction demand at each tire is not determined yet, but a comparison of the different parts is conducted in the following chapter. Furthermore, a worst case analysis is made by superposing all three parts, based on their magnitudes without considering the direction of each.

Results

Urban Traffic vs. Motorway Traffic

The plots of the cumulative distribution function (CDF) in Fig. 5 show translational acceleration, velocity and displacement demand of the two drive cycles discussed in chapter *Test Drive*. The workspace difference is demonstrated by a XY-plot (Fig. 4) of the trajectories determined by the used MCA for exemplary representatives of the drive cycles.

The CDF-plots prove that urban traffic shows higher characteristic motion properties than motorway traffic and verify the assumption made in the introduction, also proving an increase in energy demand. Because of the large workspace demand, an analysis of energy demand is not conducted at this point and will be treated in the following chapter (*MCA Parameterization*). It must be emphasized that even the unscaled motion simulation shows a translational acceleration demand of less than 8 m/s^2, underlining the basic assumption of the feasibility of a WMDS with friction limited transmission of traction forces. A closer look onto friction demand is also conducted in the following chapter.

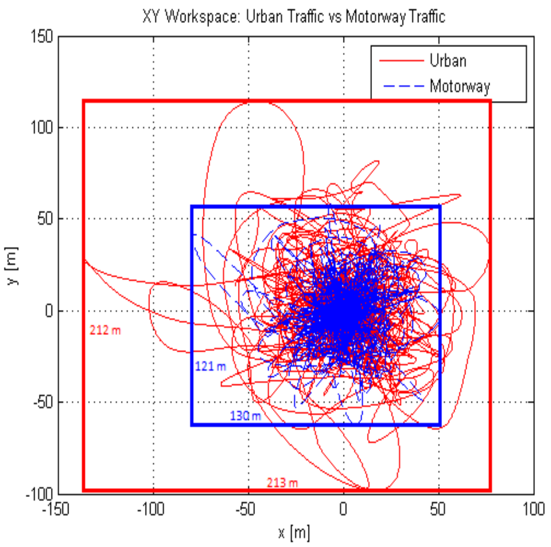

Fig. 4. Comparison of motorway and urban traffic concerning workspace demand

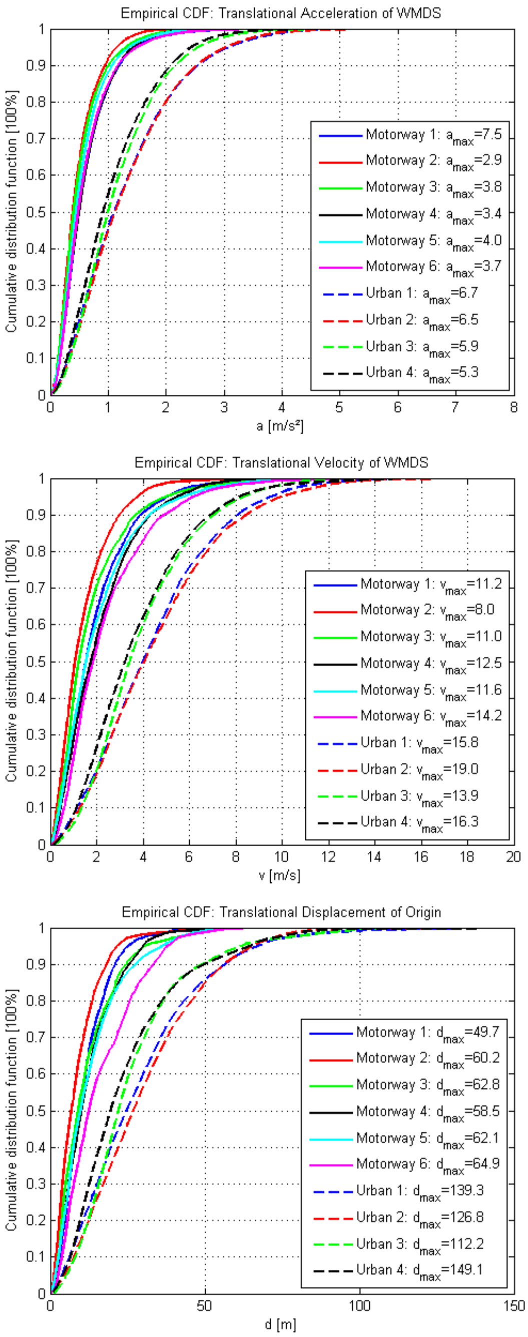

Fig. 5. Comparison of motorway and urban traffic concerning acceleration, velocity and displacement demand (scale factor=1)

MCA Parameterization for Urban Traffic Simulation

Because of the large workspace demand for unscaled driving simulation of sustained acceleration, as it is in urban traffic situations, scale factors that reduce the target horizontal acceleration have to be considered. This approach is common in the field of motion simulation [Gre1], [Cha1]. It must be emphasized that maintaining an unscaled simulation is still possible for specially developed maneuvers fitting DS properties. The influence of scaling horizontal acceleration is shown for three MCA parameterizations listed in Table 4. The parameterization is tuned iteratively for the test drives considering human perception thresholds.

Table 4. MCA parameterization for scale factors

Scale factor	LPF cut-off frequency (ω_{LPF} /s^{-1})	LPF damping ratio (ς_{LPF} /s^{-1})	Feedback gain (τ_v/s)	Feedback gain (τ_d/s)
1	0,559	1,59	4,49	19,77
0,7	0,709	1,202		
0,5	0,834	0,751		

Table 5 shows the influence of the scale factors on power, acceleration, displacement and velocity of the WMDS. The results represent the arithmetical average ($Ø_{urban}$) of the 50 %, 90 % and 100 % quantile for the four urban test drives conducted. The workspace demand strongly depends on the scale factors. For the conducted simulation it is recommendable to reduce target values but now the degree of reduction is only limited by the room and not by moving mass or required system dynamics anymore.

Table 5. Simulation results – acceleration, velocity, displacement and power demand for urban traffic

Scale factor		$Ø_{urban}$ |P| /kW		$Ø_{urban}$ |a| /m/s²	$Ø_{urban}$ |d| /m	$Ø_{urban}$ |v| /m/s
		best case	worst case			
1	$Q_{.5}$	3,5	4,0	1,0	23,2	3,7
	$Q_{.9}$	14,9	17,0	2,3	52,9	7,6
	$Q_{1.0}$	101,1	115,7	6,1	131,8	16,2
0,7	$Q_{.5}$	1,0	1,1	0,6	10,0	1,7
	$Q_{.9}$	4,9	5,6	1,5	23,8	3,7
	$Q_{1.0}$	31,6	36,2	4,0	52,8	8,3
0,5	$Q_{.5}$	0,27	0,31	0,36	4,5	0,88
	$Q_{.9}$	1,72	1,97	1,05	11,0	1.92
	$Q_{1.0}$	12,21	13,97	2,75	26,5	4.56

Regarding WMDS motion, Table 6 contains the average power demand (P_{mean}) for three MCA parameterizations with respect to the urban test drives. The scale factors are only applied to horizontal accelerations, therefore energy demand for yaw motion is independent from scale factors. The distinction of best and worst case scenario is only applied to translational motion because of the energy demand drop of about one dimension for yaw and tilt motion. Specifically the best case scenario shows promising values for energy demand. The order of magnitude is feasible by modern electric vehicle battery systems. The influence of moving mass and the requirement of high recuperation power is highlighted by the comparison of the two cases.

Table 6. Simulation results – energy demand for urban traffic

Scale factor	Øurban $P_{mean_{trans}}$ / kW		Øurban $P_{mean_{yaw}}$ / kW	Øurban $P_{mean_{tilt}}$ / kW	Øurban $\sum P_{mean_i}$ / kW	
	best case	worst case	worst case	worst case	best case	worst case
1	1,4	3,02		0,032	1,44	3,06
0,7	0,43	0,66	0,01	0,029	0,47	0,7
0,5	0,14	0,17		0,028	0,18	0,21

The components of friction demand for performing the urban test drives by the WMDS are shown in Table 7. While translational and centripetal components show similar magnitude, the yaw component is of minor importance. The final superposition of all components, considering magnitude and direction, cannot be performed yet, but even the worst case analysis, superposing only magnitude of all friction components at each time step, shows that a maximum friction coefficient of μ_{max}=1,19 is required to perform an „ideal“ urban driving simulation that can be recognized by suitable tire-ground pairing. For scaled simulation, the maximum value drops to about μ_{max}=0,7 proving the basic assumption that friction limited DS are capable to perform typical driving simulation for normal driver. The worst case analysis also shows that the $Q_{1.0}$ values of the different friction components do not occur simultaneously.

Table 7. Simulation results – friction demand for urban traffic

Scale factor			Øurban $\lvert\mu_{trans}\rvert$	Øurban $\lvert\mu_{centripetal}\rvert$	Øurban $\lvert\mu_{yaw}\rvert$	Øurban $\sum \lvert\mu_i\rvert$
	1	$Q_{.5}$	0,11	0,05	0,002	0,17
		$Q_{.9}$	0,24	0,17	0,018	0,4
		$Q_{1.0}$	0,62	0,59	0,176	1,19
	0,7	$Q_{.5}$	0,06	0,03	0,002	0,1
		$Q_{.9}$	0,16	0,11	0,018	0,26
		$Q_{1.0}$	0,41	0,34	0,176	0,72
	0,5	$Q_{.5}$	0,04	0,02	0,002	0,06
		$Q_{.9}$	0,11	0,07	0,018	0,18
		$Q_{1.0}$	0,28	0,27	0,176	0,56

Conclusion

The results of the paper show the advantage of the WMDS approach compared to state of the art DS. Even the worst case assumption shows that friction demand, power supply and mass reduction are feasible by WMDS. WMDS are capable to perform a wide range of traffic situations while maintaining system dynamics. It has to be stressed that the field of application of the new WMDS embraces state of the art practice and augments to untapped fields, like high fidelity urban traffic situations. The developed “ideal” motion cueing algorithm shows the necessary workspace in relation to the algorithm parameterization that affects the quality of real world driving experience simulation and the deducible system requirements such as DS velocity and acceleration. The parameterization of the low-pass filter and the adjusted feedback gains have a major effect on the workspace requirement. Further research will show the potential for improvement. The calculated energy and friction demand proves the introduced basic assumption of friction limited DS feasibility. Scaling target motion information is advisable for most simulations but only owes to the room and not to moving mass and required system dynamics anymore.

The presented results allow the aggregation of a large motion envelope with high system dynamics into a high fidelity DS that combines properties of high dynamic simulators, like the one by Daimler AG [Zee1], with the workspace of the Toyota [Mur1] and NADS systems [Cla1].

Concerning the best case assumption, the presented WMDS concept shows promising advantages concerning further improvements that are valuable for system development and validation processes in automotive industry.

Further WMDS behaviour has to be analyzed. Additional DS components like the visual system, data transfer, air conditioning of the cabin and the introduced additional motion system for performing vehicle pitch, roll and heave motion have to be considered. Also a suitable safety concept has to be developed that allows a safe usage of the autonomously moving motion base, even for system failures. Furthermore conventional vehicle dynamic behaviour is inherited causing known problems such as, wheel load change, resulting in increase of control effort to establish smooth DS motion. Greater WMDS analysis has to be conducted to answer advanced open questions.

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Formula Symbols

Symbol	Unit	Description
a	m/s²	acceleration
β	rad	Side slip angle
c	1	Parameter to tune stability margin of loop gain
$Ø_{urban}$	1	Arithmetical average of the urban test drive results
d	m	Displacement
E	J	Energy
η	1	Efficiency factor
M_0	Nm	Holding torque
m_{Cabin}	kg	Mass of Cabin
m_{total}	kg	Mass of total system
ν	rad	Course angle
μ	1	Friction coefficient
P	kW	Power
ψ	rad	Yaw angle; rotation about vertical z-axis
Q	1	Quantile
r_{drive}	m	Distance from center of base frame to drive unit center; distance from z-axis to wheel contact point
Θ_z	kgm²	Moment of inertia of whole system about vertical z-axis
τ_d	1	Feedback gain for WMDS displacement of origin
τ_v	1	Feedback gain for WMDS velocity
v	m/s	Velocity
ω_{LPF}	Hz	Cut-off frequency
ς_{LPF}	1	Damping ratio