Exploration of Steering Wheel Angle Based Workload Measures in Relationship to Steering Feel Evaluation.

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ABSTRACT

This paper discusses the driver steering task for different steering manoeuvres, and for varying vehicle design characteristics. Two steering manoeuvres are selected, a single lane change and a u-turn. We have adjusted the vehicle design characteristic in the following two ways, by reducing the torsion-bar stiffness in the steering system and by reducing the understeer gradient of the vehicle, respectively. In both cases, it is expected that the driver will have to increase the steering effort to maintain the same steering performance, i.e. he will experience a higher workload.
Steering performance and workload will be discussed in terms of the TLC (Time to Line Crossing), the HFA (High Frequency Area) and SRR (Steering Reversal Rate). These indicators have been explored experimentally before as potential primary task measures in subjective tests, to assess steering task performance for different settings of a steer-by-wire control system.
A number of questions arise from such experimental analysis. First of all, these closed-loop tests are carried out with real drivers, with an unknown variation in their driving skills, making the outcome of these tests hard to interpret. Closed loop simulation analyses often apply driver models, claiming to incorporate the major features of real driver response,. This suggests a minimum demand for the indicators introduced above, that they also make sense for such driver models in a simulation environment. This statement is supported by recent simulation research by Sakai et. al. in [4], who observed increased driver workload to prevent yawrate oscillations in case of a reduced torsion bar stiffness in the steering system. Second, a driver is responding to the steering and vehicle response characteristics, as observed through the steering wheel and the vehicle yaw rate behaviour. One would therefore expect a relationship between the workload- en performance indicators, and these response characteristics. These relationships are discussed in this paper in terms of equivalent spring- and damper values, as introduced by Misaji et. al in [2].

Keywords / Steering Assistance and Control, Driver-Vehicle Interface

1. INTRODUCTION

Developments in intelligent steering support systems, such as EPS and steer-by-wire in general, have renewed the interest in assessment methods to judge steering performance and to suggest the optimal steering system settings. The classical objective methods, based on reference manoeuvres such as step steering input and random steering input only describe the overall vehicle performance and do not account for the vehicle-driver interface. These methods are therefore insufficient to judge advanced steering system performance. Other methods have been proposed based on the driver steering corrections to follow a certain path, where one may count the number of significant corrections (steering reversal rate) or consider the energy content of the high frequency part of the steering signal power spectral density (high frequency area). These approaches have in common that the driver is taken in the loop, and are assumed to be related to mental workload, i.e. the perception of the driver to have sufficient skills to overcome critical steering circumstances. These indicators have been successfully applied in the assessment of a side stick steering control interface, see [6], where it turned out that both indicators were able to discriminate between different steering column stiffnesses and different power steering gains. These results are confirmed in [4], where it was observed that lower steering column stiffness may lead to larger yaw oscillations, to be compensated by the driver at the cost of a higher workload. However, the results were less clear for combinations of such modifications, which means that the application of such indicators in general is still not well understood.
On the other hand, the experimental evidence is principally based on tests with varying driver characteristics, and a more fundamental approach starting from simplified driver models may improve our understanding. That means that one may consider a driver model as used in [4], accounting for both physiological effects (leading to lag, as well as to delay in response) and control action in response to some anticipated path error (including a proportional response plus a derivative control term, i.e. a lead control term in the steering response) and ask the question whether this

model will show an effect in workload related measures when required to follow a prescribed path such as a single lanechange or a u-turn, for some modified vehicle parameters. Can we observe some behaviour of the driver model, that confirms the experimental findings? The vehicle parameters, to be varied are the steering column stiffness and the understeer gradient.
In addition to these studies, oscillatory input studies are carried out for the same selection of parameter variations, for some selected input frequency. This results in hysteresis plots of steering torque vs. steering angle and yaw rate vs. steering wheel angle.
An interesting approach to interpret results based on hysteresis contours was presented in [2] at the previous AVEC '02 symposium, and applied to analyse on-centre high speed feel with emphasis on the relationship between non-linear vehicle characteristics and steer feel subjective ratings.
This method derives an equivalent linear vibration system based on the hysteresis characteristics with the 'bone-curve' of the hysteresis plots expressed in terms of a power function in the steering wheel input angle. That means that equivalent stiffness and damping characteristics are derived that serve for the interpretation of the differences of steering behaviour and driver feedback for the different sets of design parameters.
Finally, the results of both types of analysis are compared. This will lead to further support for the practical feasibility of certain workload measures, and it will broaden our understanding of these measures in terms of the equivalent characteristics of the steering system.
The analysis in this paper is based on the model in [4] with some modifications, see section 2. In section 3, we will discuss workload & steering performance indicators. The approach in [2] and [5] is shortly addressed in section 4. In section 5, the model is used to track a single lanechange under preview control, and the results will be discussed. Likewise, in section 6, we will treat the u-turn. Finally, in section, 7, conclusions are drawn.

2. THE MODEL

The model as used in this paper is based on [4] including the choice of parameter values, with some modifications, see fig.1. The model consists of a steering mechanism including EPS through a torque motor, accounting for torsional rigidities, viscous and frictional damping in column shaft, torsion bar and vehicle front suspension. Compared to [4], the EPS reduction gear ratio has been reduced with 50 % to reduce the hysteresis in the steering response (which was quite large in [4]), and a small mechanical trail of 0.01 m has been included. Axle characteristics (single track vehicle model) are taken as nonlinear, described through the Magical Formula model, with the vehicle parameters (reference case) listed in Table 1.
The understeer gradient will be varied by increasing the distance from front axle to the vehicle cog, with all the other parameters in table 1 unchanged. The steering

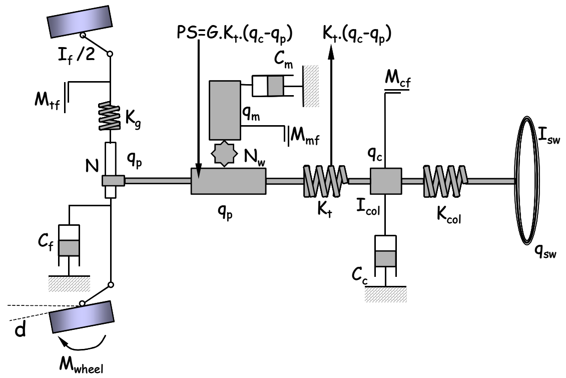

Fig. 1.: Model of the steering system

column stiffness will be modified by reducing the torsion bar stiffness from its reference value of 120 Nm/rad.
The lateral force and aligning torque for the front axle (indicated with an index f) are described as follows:

$$F_f = D_{yf}.\sin[C_y.\arctan(B_{yf}.\alpha_f)] \tag{1}$$

$$M_f = F_f.D_m.\cos[C_m.\arctan(B_m.\alpha_f)] \tag{2}$$

and likewise for the rear axle, with the peak factor D_{y*} equal to the axle load (f, r) and B.C.D equal to the slip-stiffness.

Table 1.: Parameter values vehicle model

vehicle mass	1250 [kg]
front axle slip stiffness	100,000 [N/rad]
rear axle slip stiffness	120,000 [N/rad]
shape factor axles C_y	1.3
stiffness factor B_m	20
stiffness factor C_m	1.2
peak factor D_m	0.025 [Nm]
front axle to cog	1.1 [m]
wheel base	2.8 [m]

The vehicle is assumed to follow a track, where the driver model is looking some preview time (taken as 1 sec.) ahead. The vehicle is driving with a speed of 72 km/hr .The preview path error e_p (see fig. 2) is based on the quadratic extrapolation of the vehicle trajectory, with the error described as the perpendicular distance of this preview point to the reference course. Two lines are indicated in fig. 2, lying at a certain prescribed perpendicular distance to the reference course. These lines are considered to be the boundaries of the lane the vehicle is following, and crossing either one of these lines is interpreted as leaving the road. The driver is responding to the preview path error e_p as follows (cf. [4]:

$$\theta_{sw} = K_p \frac{(T_l s + 1).e^{-0.2s}}{0.1s + 1}.e_p \tag{3}$$

for steering wheel angle θ_{sw}. The parameters in expression (3) may be questioned. The coefficient 0.1

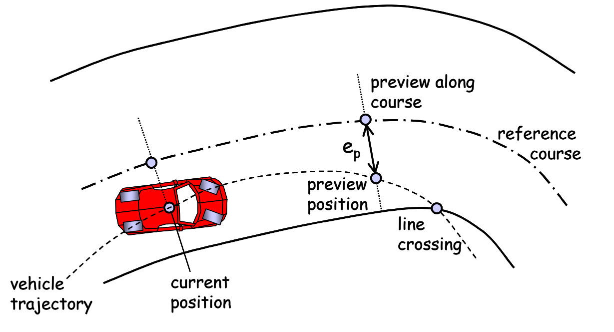

Fig. 2.: Tracking of the reference course

implies the driver steering response to exceed 1 Hz in general. Under normal traffic conditions, drivers tend to keep their corrective response way below 1 Hz corresponding to the control of path error or path angle. Only with increased workload, the response is believed to go up in frequency, which may be due to control with higher order values such as heading rate or path error rate. This last type of (derivative) control is described by the lead-time constant T_l. In the reference situation, the value of T_l is assumed to vanish. Compensation of higher yaw oscillations is mostly done by the driver through derivative control, i.e. by a value for T_l such that the difference in yaw rate response compared to the reference situation is reduced at the cost of higher frequency corrections.

The gain factor K_p is assumed to have the same value as in [4], i.e. $K_p = 0.55$ [rad/m].

In correspondence with [4], with $r_r(t)$ and $r(t)$ the yaw rate in the reference case and for modified model parameters, respectively, the error in yaw rate performance to be reduced is defined as the evaluation index EI:

$$EI = \frac{\int_t [r(t) - r_r(t)]^2 \, dt}{\int_t r_r(t)^2 \, dt} \tag{4}$$

3. WORKLOAD AND PERFORMANCE INDICATORS

The following indicators are introduced:

HFA: Higher Frequency Area.

HFA is derived from the power spectral density function based on the deviation of the steering wheel signal from the running average. Here, it is calculated as the energy in a frequency band f_{cr} - 4.0 Hz divided by the energy in the frequency band 0.0 - f_{cr} Hz. The value f_{cr} should be chosen such that the impact of the design change is well described. For variation in the steering column stiffness, the PSD function tends to change mostly beyond 1.4 Hz. With the vehicle understeer gradient changed, the PSD is dramatically changed as well in the frequency range between 0.8 Hz. and 1.4 Hz. We therefore select f_{cr} as 1.4 Hz and 0.8 Hz for both sensitivity studies, respectively. We note here again that the limit frequency f_{cr} is usually chosen lower, but that would imply to increase the lag time coefficient in (3). We have decided to maintain the value of 0.1 sec.

SRR: Steering Reversal Rate.

SRR is defined as the number of times per second that the direction of the steering wheel movement is reversed through a small angle, here chosen as 0.4°.

There is evidence that these indicators are related to workload. However, some correlation with steering performance cannot be denied either.

A same uncertainty exists with respect to:

TLC: Time to Line Crossing.

Usually, TLC is defined as the time needed to reach the boundary of the lane, if steering wheel and speed are assumed not to change. In this paper, TLC is defined slightly different, as the time passed before leaving the road, based on quadratic extrapolation of the vehicle path, see fig. 2. We will only consider the (critical) TLC-values below 3 sec.

TLC tells us something about the combined position and location of the vehicle with respect to the 'lane boundaries', which may get worse even if the vehicle moves more closely to the lanechange trajectory. In that case, TLC indicates more severe oscillations of the vehicle, which is expected to correlate with the previous indicators.

4. STEER FEEL INDICATORS

Let us consider the steering wheel angle – steering torque hysteresis plot for the reference case at steering angle amplitude of 20° for 1 Hz, see fig. 3.

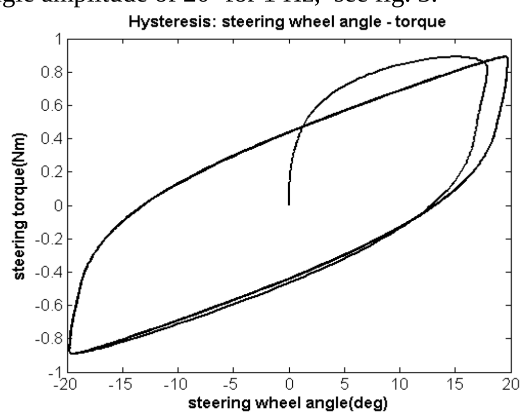

Fig. 3.: Steering wheel angle - Steering torque hysteresis curves at 1 Hz

The hysteresis vibration system, leading to this plot, can be derived from an equivalent linear vibration system according to the following equation:

$$\ddot{\theta} + 2.C_t.\dot{\theta} + K_t.\theta = M.\cos(\omega t) \tag{5}$$

The equivalent damping C_t (related to the area bounded by the hysteresis loop) and stiffness K_t (the 'slope' of the loop) depend on the wheel turning frequency and appear to correlate with subjective assessment of steering feel in terms of damping feel and perceived steering torque gain, see [5]. In addition, the ratio of C and K appears to distinguish between perceived excessive damping (with negative effect on driver control) and high sensitivity to human factors and external factors (and therefore lower controllability).

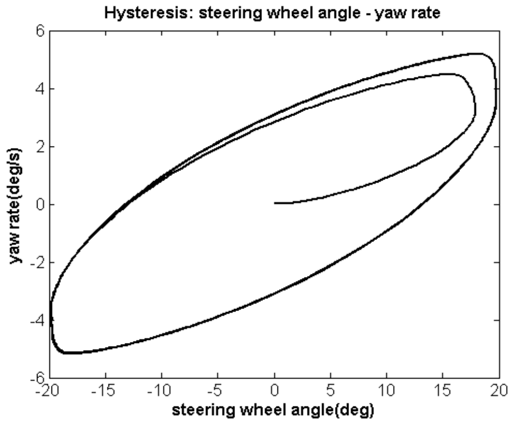

Fig. 4.: Steering wheel angle – Yaw rate hysteresis curves at 1 Hz

Similar conclusions are drawn in [5] with respect to the steering wheel angle – yawrate hysteresis plot, see fig. 4. Here, the damping and stiffness coefficients, denoted as C_r and K_r, correspond to the yaw rate response lag and the yaw rate gain against steering angle. Especially for the spring constants, the largest discrimination between different vehicles was observed for 1 Hz. Further analysis in this paper will therefore be carried out for this frequency and steering wheel amplitude of 20^o.

5. TRACKING THE SINGLE LANECHANGE

The single lane change and the path of the vehicle under reference conditions are shown in figure 5. A slight overshoot of about 0.5 meter is observed. The steering wheel angle power spectral density for reduced steering column stiffness is shown in figure 6. One observes that most of the steering energy is restricted between 0 and 1.4 Hz, due to the selected lag time constant in (3), with some energy for slightly larger frequencies.

With the vehicle moving first to the left, it is to be expected that the TLC with respect to the right boundary is more critical than the TLC with respect to the left boundary, as indicated in fig. 7. The vehicle is first approaching the right road boundary. The driver tends to follow the path smoothly up to the end of the lanechange when TLC to the left boundary becomes critical. The steering corrections by the driver tend to bring the vehicle back in approaching the right boundary again. This suggests introducing the following performance measures:

$$TLC_r \equiv \int_t [3 - \min(3, TLC_{right})]dt \qquad (6)$$

$$TLC_l \equiv \int_t [3 - \min(3, TLC_{left})]dt \qquad (7)$$

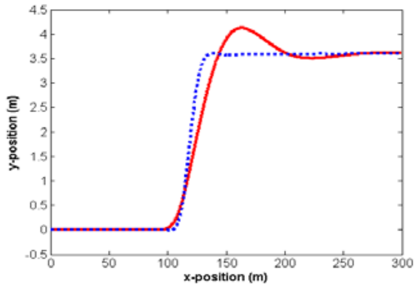

Fig. 5.: Lane change trajectory (dotted) and vehicle path, (solid)

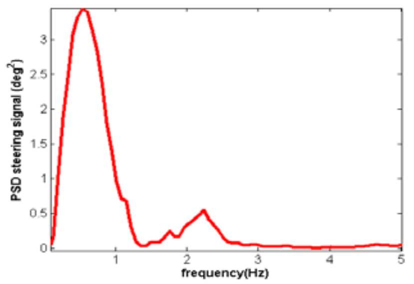

Fig. 6.: PSD, steering wheel angle

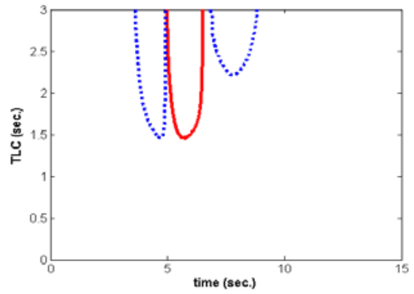

Fig. 7.: Time to Line Crossing, reference case (left: solid, right: dotted)

We shall now vary the model parameters in the following way:

i. torsion bar stiffness = f. 120 Nm/rad, f = 1.0 ... 0.1
ii. understeer gradient = 0.035 … 0.002

The results for decreasing torsion bar stiffness are shown in figures 8. In the top two pictures, the results for nonzero time lead factor T_l are shown as dotted lines. With T_l increased (in the order of 0.02 sec.) to minimize the evaluation index EI cf. (4), TLC is reduced indicating that the performance in terms of oscillations is getting worse. The vehicle is moving more oscillatory, which is also demonstrated by the strong impact of this derivative control on HFA.

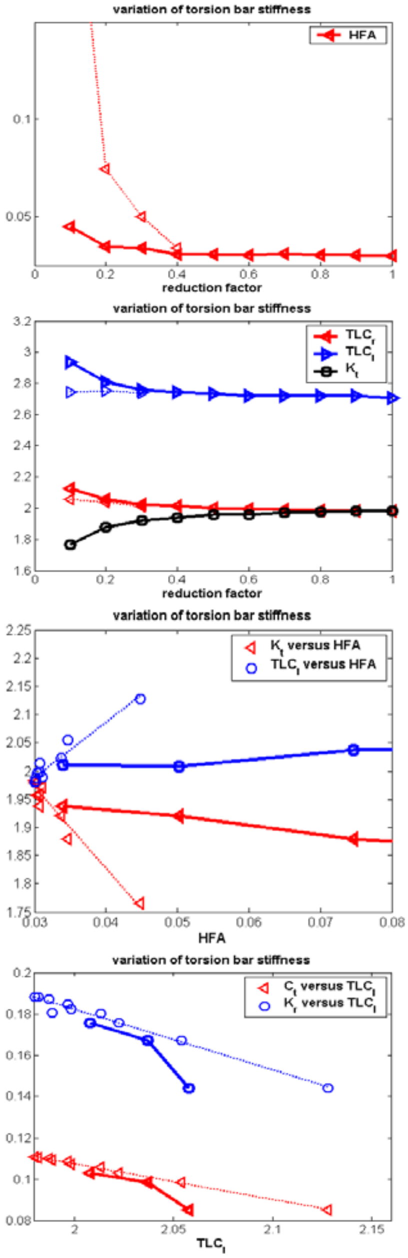

Fig. 8.: Variation of workload-, steering performance- and steer feel indicators for a single lane change, varying torsion bar stiffness

All steer feel indicators are decreased for reduced torsion bar stiffness (including the ration of C and K, not shown here), with the results strongly correlated to the workload and performance indicators. The dotted straight lines in the lower two plots are regression lines. The solid lines are the values for adjusted lead time constant T_l. In terms of the interpretation in section 4, one may state that lower torsion bar stiffness result in lower controllability, lower perceived steering torque gain, lower yaw rate gain and lower yaw rate response lag, leading in a combined way to higher workload and reduced steering performance. This may be compensated by derivative control, leading to even higher workload.

The Steering Reversal Rate (SRR) appeared not to be very discriminating. A slight increase from SRR=0.4 (stiff torsion bar) to SRR=0.533 was observed from the simulations.

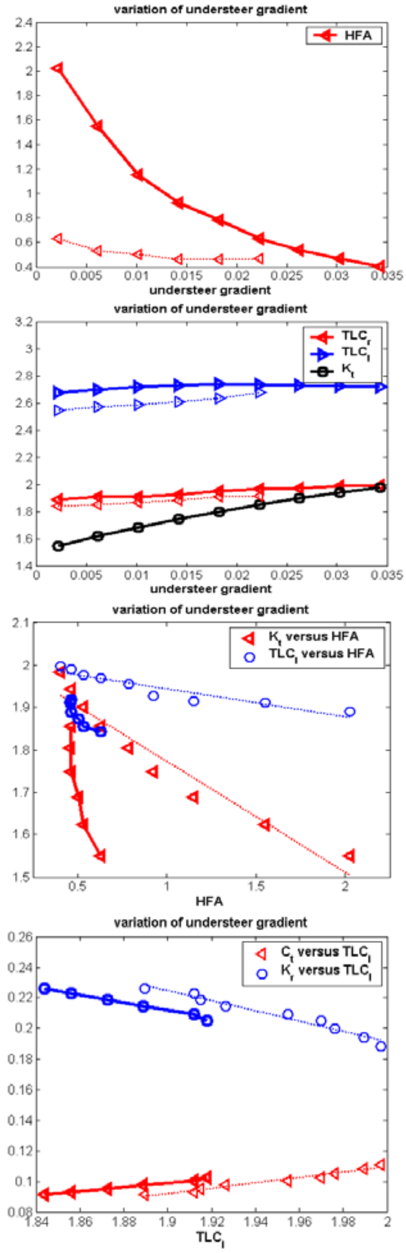

Fig. 9.: Variation of workload-, steering performance- and steer feel indicators for a single lane change, varying understeer gradient

Similar results for varying understeer gradient are shown in figure 9. One observes a strong increase of HFA near neutral steering. At the same time, also the TLC-performance is reduced. For comparison, we have listed the reaction time for a step steer input (steering wheel angle of 20^o) and the steady state yaw gain for the vehicle in table 2, see also [3]. Both gain increase (more 'sportive driving') with reduced understeer, as well as the reaction time, supporting these results.

Nonzero T_l to minimize the evaluation index EI leads to excessive HFA. In fact, for this value of T_l for almost neutral steering (about 0.07 sec.), the driver-vehicle system appears to become close to loosing stability. Some smaller value of T_l is therefore more realistic.

Table 2.: Step steer response results

understeer gradient	0.0343	0.0263	0.0185	0.0102	0.0022
Yawrate gain (s.s.)	4.74	5.14	5.62	6.20	6.92
t_{reac} [sec.]	0.173	0.183	0.199	0.223	0.260

Furthermore, it appears that the steering torque related steer feel factors K_t and C_t increase moving towards neutral steering, whereas the yaw rate related factors K_r and C_r increase. This means again a lower controllability and lower perceived steering torque gain, but a higher yaw rate gain and yaw rate response lag (confirmed by table 2).
The SRR tend to stay small (around 0.3) for understeer gradient exceeding 0.015. For smaller values, SRR rapidly grows to 1.53 for understeer gradient equal to 0.022. Hence, SRR again seems not to be such a reliable indicator, except for extreme variation in vehicle parameters.

6. TRACKING THE U-TURN

We have repeated some calculations for a u-turn (anti-clockwise, with curve radius of 70 meter), with the same model, similar speeds and zero corrective lead time values for T_l. For variation of the steering column stiffness, the resulting TLC-values (right boundary) and HFA-values are shown in figure 10.

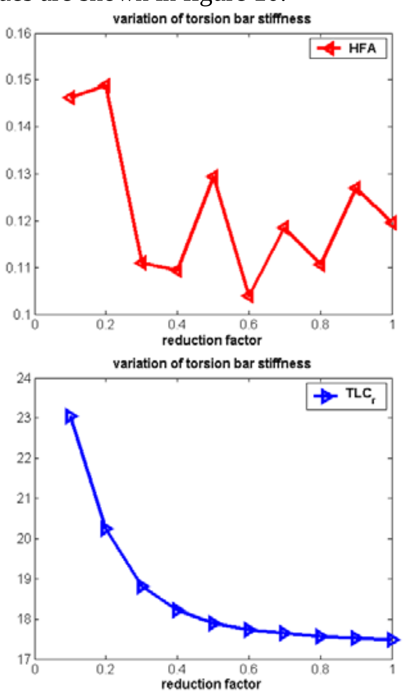

Fig. 10.: Variation of HFA and TLC for a single u-turn, varying torsion bar stiffness

These results show clearly that the u-turn in combination with HFA is not a reliable approach to discriminate between different steering characteristics. On the other hand, TLC-results are qualitatively comparable to the results for the lane change.

7. CONCLUSIONS

We have examined workload and performance indicators, based on a simulation approach. It turns out that, for a single lane change manoeuvre, the High Frequency Area (HFA) and Time to Line Crossing (TLC) are appropriate to discriminate between different design characteristics with respect to both the steering system and the vehicle chassis. HFA is depending very much on the selected limit frequency f_{cr} between the 'reference'-frequency values and the increased levels due to the design modifications. The SRR (Steering Reversal Rate) does not appear to be a suitable indicator for this purpose. Correlation with equivalent damping and stiffness, both with respect to the steering wheel – steering torque hysteresis and the steering wheel – yaw response hysteresis, allows for further interpretation of these indicators. For a u-turn, these conclusions still hold for TLC. HFA appears not to be a robust indicator for different testing manoeuvres.

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