Frequency response and latency analysis of a driving simulator for chassis development and vehicle dynamics evaluation

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Abstract *– Driving simulators come in various sizes and are used for car development, driver training and much more; but can a mid-size simulator be used for subjective evaluation of vehicle dynamics in chassis development of normal cars? To answer this question a frequency response and latency analysis is conducted to find the performance characteristics of a Cruden simulator built up at AUDI AG. Describing functions, crosstalk and signal-to-noise ratios for the motion platform are evaluated in 6DoF, based on methods described in the AGARD-144 Advisory Report [Lea79]. For all 6DoF the system is limited in bandwidth by the first eigenfrequency, outside the frequency range of interest. Two different methods are used to evaluate frequency response for the force-feedback steering wheel, showing results with- and without human physics in the dynamic control loop. Visual latency is higher than the dynamic threshold for the motion platform and control loader, but still less than 30 ms using the fastest settings. There is not much room for improvement of the rendering software, as two-thirds of the visual latency comes from the projectors. These results show that the platform is suitable for subjective evaluation of vehicle dynamics ranging from 0-10 Hz.*

Keywords: *Frequency response, latency, eigenfrequency, bandwidth, AGARD-AR-144*

Introduction

Driving simulators have become an important development tool in the automotive industry. Professional motorsport teams use simulators to tune the car setup, develop race strategies and assess & train their drivers in a controlled environment. Car manufacturers use them to reduce vehicle development time and cost. Cruden simulators have successfully been used for vehicle development for many years at various customers. Using a simulator for the development of driver assistance systems or for testing human machine interfaces involves different demands compared to using the simulator for the evaluation and development of vehicle dynamics. Big simulators provide a larger range of movement compared to smaller simulators, but are generally heavier and slower. Recent publications already show the usefulness of big simulators, like the Stuttgart driving simulator, in the concept phase of vehicle dynamics development [Bau14]. The question in this research is: Are the dynamic characteristics of a mid-size simulator for subjective evaluation of vehicle dynamics in chassis development of normal cars suitable?

Delays between driver input and simulator response can reduce simulator effectiveness or even be the cause of simulator sickness [Joh05]. It is therefore the aim of simulator engineers to reduce these delays in the visual, haptic and vestibular feedback channels to a minimum. Professional test drivers evaluate car characteristics by assessing the vehicle response to various driver inputs (e.g. step steering, ISO lane change). The driver assesses the handling quality of the vehicle by the time delays between different vehicle states, using steering as input and yaw-rate or lateral acceleration as reaction of the vehicle [Hei02]. While the lateral acceleration is presented to the driver by the motion system, yaw-rate is mainly perceived with the eyes and thus presented to the driver by the visual system [Tom06]. If a test driver is driving a curve on a simulator and the visuals have a different delay compared to the motion platform, the sensory information does not match and the driver cannot rate the dynamics of the vehicle. It is therefore crucial that the latencies of all parts of the simulator are well known and can be shaped to be in sync. Additionally, test drivers also rate the magnitude of different vehicle states to evaluate the characteristics of a vehicle. If for example a vehicle is driven at the limit in a curve the driver can clearly feel this limit by a drop in steering torque as well as an increase in side slip, both caused by grip loss of the tires. While the steering torque is presented to the driver by the control loader, the side slip and lateral behaviour is cued by the motion platform. To provide the driver in a simulator with realistic stimuli, it is necessary to know the dynamic response of the platform and control loader. The platform will be used to evaluate vehicle dynamics ranging from 0 Hz to frequencies considerably below 10 Hz [Als94, Mit14], so the focus is on these frequencies.

Figure 1. Photo of the Cruden simulator at AUDI AG

This paper describes the objective results obtained from an extensive performance analysis of a Cruden simulator at AUDI AG. Prior articles e.g. [Str07] or [Nie08] evaluated the performance of single simulator components such as the motion base or the visuals. In this paper, motion, steering and visuals are analysed and compared. The motion platform as well as the control loader are subject to tests to evaluate the magnitude and phase response for the frequencies of interest and identify the bandwidth and eigenfrequencies. Additionally a full latency analysis is performed for all parts of the simulator, revealing latency bottlenecks and potential room for improvement. The simulator at hand (Figure 1) consists of a six degrees of freedom (6DoF) Hexapod providing vestibular and haptic feedback to the driver. A control loader is used to create fast and accurate haptic feedback on the steering wheel. The control loader provides a peak torque of 30 Nm and maximum continuous torque of 9.6 Nm. The visuals are projected by five 120 Hz WUXGA DLP projectors on a circular screen mounted on the floor. The screen is 8 meters in diameter, creating a 210 degrees horizontal field of view. Auditory feedback is provided by three surrounding speakers and a subwoofer fixed under the driver seat. The simulator system architecture is displayed in Figure 2. The system contains a motion cabinet (MCB) with all electronic control hardware for the six platform actuators and the control loader. The real-time PC in this motion cabinet communicates with a computer running the dynamic vehicle model. The model updates every 0.6 ms and UDP communication is at 1000 Hz. Both computers communicate over UDP with a master PC running Cruden simulator software: Racer Pro. The master PC communicates with five visual rendering PC's which render the visuals for the five projectors.

Table 1. Platform excursions for 500 kg payload

	Position	Velocity	Acceleration
Surge (+/-)	0.63 m	0.8 m/s	15 m/s²
Sway (+/-)	0.66 m	0.8 m/s	15 m/s²
Heave (+/-)	0.41 m	0.6 m/s	15 m/s²
Roll (+/-)	29.2 °	35 °/s	600 °/s²
Pitch (+/-)	28.2 °	35 °/s	700 °/s²
Yaw (+/-)	28.7 °	40 °/s	900 °/s²

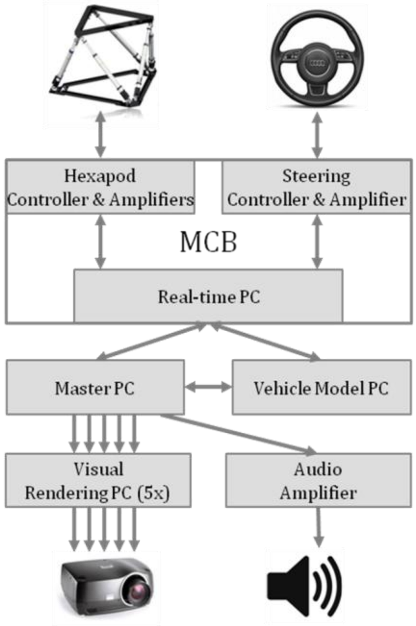

Figure 2. Simulator system architecture motion, control loader, visuals, audio

Methods

Motion frequency response

The performance of the motion platform was analysed based on AGARD-AR-144 [Lea79] and the methodology described in [Nie08, Roz07]. Conducting a frequency response analysis of the motion platform requires measurements of platform acceleration in six degrees of freedom. The setup consists of three triaxial acceleration sensors placed on the bottom of the upper platform, equally distributed on a circle with a radius of 0.5 meter. The acceleration setpoint consists of pure sinusoidal signals between 1 Hz and 10 Hz in steps of 1 Hz. Additional measurements are performed for frequencies from 0.2 to 1 Hz in steps of 0.1 Hz. Each sinusoid is commanded for four seconds or at least eight full cycles with an additional fade in and fade out time of five seconds. The four different amplitudes at each frequency are chosen to be 50%, 25%, 12.5% and 6.25% of the particular system limit at each frequency. The system limits for position, velocity and acceleration can be found in Table 1. Higher amplitudes are avoided to prevent damage to the simulator mock-up, seat and dashboard. To evaluate the bandwidth of the system, additional measurements up to 40 Hz with a constant setpoint of 2 m/s^2 for translation and 1.4 rad/s^2 for rotation were carried out for each DoF. The setpoints are sent directly to the platform, bypassing workspace management and motion cueing. Setpoints can be generated in Simulink on the Vehicle Model PC using UDP input and UDP output blocks. This eliminates the need for software changes to do the performance tests. The sample rate for the measurement system is 1000 Hz.

Describing functions

From the measurements between 1 and 40 Hz the describing function $G(j\omega_i)$ at each frequency is calculated by dividing the FFT coefficients of the measured output $Y(j\omega_i)$ and the setpoint $X(j\omega_i)$.

$$G(j\omega_i) = \frac{Y(j\omega_i)}{X(j\omega_i)} \tag{1}$$

With this routine describing functions for the driven DoF as well as the non driven DoFs can be found. The latter are also known as crosstalk and should be kept as low as possible.

Signal-to-noise ratio

In a second step the different noise levels at each grid point are determined. The measured acceleration output of the motion platform always consists of desired output plus undesired output in the form of noise. The average power in a frequency interval σ_i^2 is described by the FFT coefficients x(i) in that frequency interval:

$$\sigma_i^2 = \frac{2}{N} \sum_{i=1}^{N/2} |x(i)|^2 \tag{2}$$

where N is the number of samples in the FFT. Relating the power of the fundamental output to the overall acceleration noise leads to signal-to-noise ratios SNR(k) for all measured combinations of frequencies and amplitudes:

$$SNR(k) = \frac{\sigma_k^2}{\left(\sum_{i=1}^{N/2} \sigma_i^2\right) - \sigma_k^2} \tag{3}$$

These SNRs can then be used to determine boundaries of the motion range with SNR values higher than a specific value.

Control loading frequency response

The control loading measurement procedure is different from the motion measurements because the control loader requires a torque setpoint instead of an acceleration setpoint. The torque is measured using a loadcell inside the control loader. This torque can only be achieved if the steering wheel is blocked by something which generates a reaction force. Two different methods to block the steering wheel are investigated independently. The simulator contains an adjustable steering column with several linkage parts between control loader and steering wheel. However, for the frequency response tests in this paper the steering wheel was mounted directly to the control loader. The influence of the steering column will be investigated in a later stage of development.

Method 1 - Human

The first method uses a human holding the steering wheel and thereby becoming part of the dynamics in the control loop. A constant torque set point of 3 Nm is used to make sure the torque always acts in one direction. This eliminates the influence of mechanical play, should there be any in the system. On top of this constant torque the system is excited with sinusoidal and random input signals.

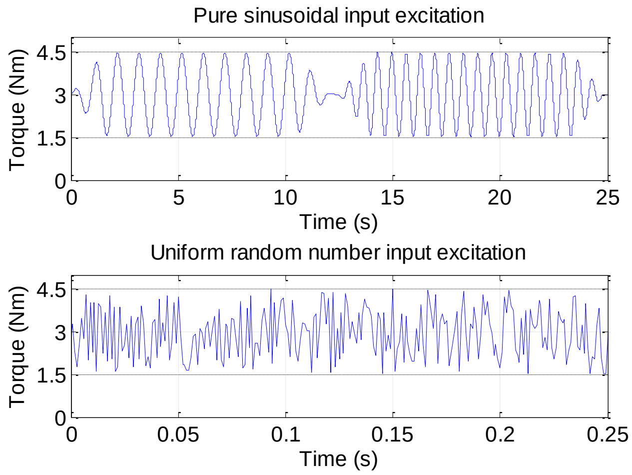

Figure 3. Example of two different input signals

The sinusoidal type is similar to the motion experiments and uses a pure sinusoidal excitation with amplitude 1.5 Nm for each frequency between 1 and 100 Hz. The random type of excitation consists of uniform random numbers between -1.5 and 1.5 Nm. Uniform random numbers are used instead of the more common Gaussian distribution to prevent excitation outside the 1.5 - 4.5 Nm range and maintain a flat power spectrum. This random signal is applied for 5 minutes while the test subject holds the steering wheel in the centre position. Examples for the two individual input signals are shown in Figure 3. To show reproducibility the experiment is carried out three times with both input signals. Additionally the experiment is repeated three times with the random noise input signal with a small female and large male test subject respectively. Although this is not a proper statistical analysis it provides some insight in the robustness of the results over humans with different body characteristics.

Method 2 - Mechanical bar

For the second method the steering wheel is blocked mechanically with a bar between the spokes. The idea of this method is to eliminate the human part of the control loop and find objective results for just the control loader. Again two different types of excitation are used on top of the constant 3 Nm torque setpoint. The experiments are carried out three times to check reproducibility of the results.

Latency

Latency research can be performed by applying step inputs to the system and evaluating the response. The latency is determined by logging both the measured response and the generated trigger signal with the same data logger. The latencies in this paper are the system's full end-to-end latencies from the moment the driver generates an input to the system until the driver receives feedback. The step input for motion is equal to 1 m/s^2 or 1 rad/s^2 and the response is measured using the three triaxial accelerometers. The control loader step response is evaluated using a step from 1 to 3 Nm and the response is measured with the loadcell inside the control loader. For the visual latency the trigger applies a sudden 90 degrees step in vehicle camera angle generating a completely new image. The response time is measured using a photocell detecting the sudden change in visuals at the centre of the middle projector. The scene is chosen such that, for a 90 degrees step in vehicle camera angle, the photocell measures a transition from full white to full black. All sensors are basic elements without embedded DSPs and are assumed to have negligible latency. The measurement setup makes sure that sensor latency can only increase measured values.

$$G(s) = \frac{1}{\tau s + 1} e^{-\tau_d s} \quad (4)$$

The measured latency can be expressed as a combination of deadtime τ_d and rise time τ with deadtime as the time it takes until the first response, and rise time as the time it takes from the initial response to reach 63% of the setpoint. These two together form the dynamic threshold value [Roz07].

Results

In the following sections the research results are given for each experiment. The first section describes the frequency response results for the motion platform. In addition to the pure 6DoF frequency response results, crosstalk and signal-to-noise ratios are presented. What follows are the results for the control loader frequency response measured with two different input excitations and in a fixed setup as well as with human test subjects. The last section shows latency results for motion, control loading and visuals with extensive results for different visual settings.

Motion frequency response

Describing functions

Figure 4 shows the motion frequency response for all six degrees of freedom. The first eigenfrequencies are found in sway and surge direction, at 16 and 22 Hz respectively.

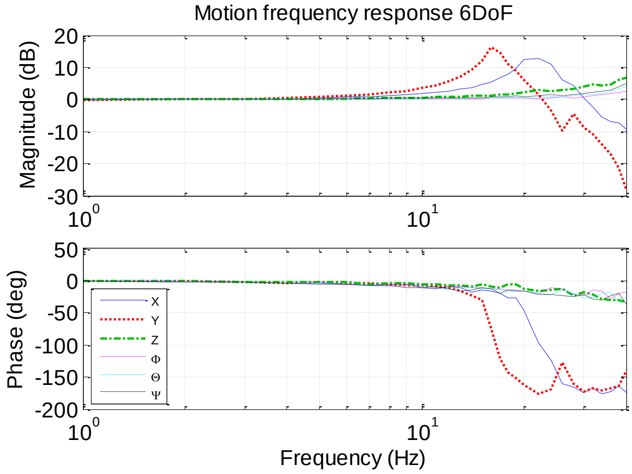

Figure 4. Motion frequency response 6DoF

The -3 dB point is found to be more than 20 Hz. Essentially the system is bandwidth limited at the eigenfrequency for the various degrees of freedom. The eigenfrequency for heave and rotational degrees of freedom is more than 40 Hz. For all directions of movement the magnitude is between 0 and 1 dB for frequencies below 5 Hz. At 5 Hz the maximum phase delay is less than 5 degrees and at 10 Hz the phase delay is just below 10 degrees. The phase-lag increases rapidly for surge and sway when the system reaches the specific eigenfrequencies for these directions.

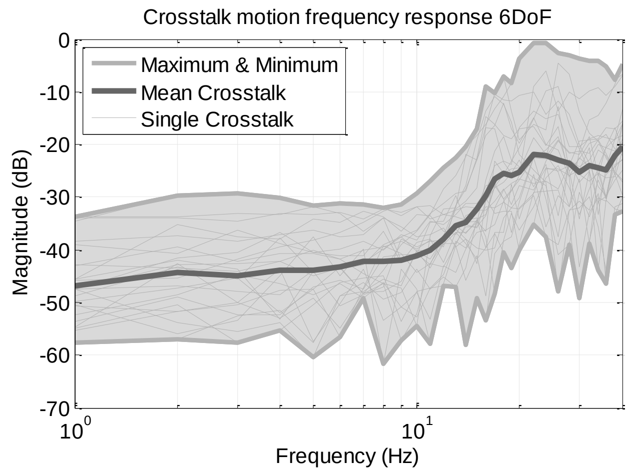

Figure 5. Crosstalk motion frequency response 6DoF

The transfer functions between driven and non-driven degrees of freedom are shown in Figure 5. The crosstalks between roll and sway and between pitch and surge were not included in this figure because the sensors were mounted at a nonzero height with respect to the axis of rotation. Thus, for these relations, excitation signals and crosstalk cannot be distinguished. Between one and 10 Hz the maximum magnitude is equal to -29 dB which is only 3.5%. The calculated mean crosstalk is in the range of -47 to -43 dB (0.5% to 0.7%). Assumed that part of this crosstalk is caused by placement errors and sensor noise, the actual crosstalk is most likely even smaller. However, the amount of crosstalk is much higher near the first eigenfrequencies, especially when the platform is excited in surge or sway direction.

Signal-to-noise ratio

For surge and roll acceleration the signal-to-noise level contour is plotted in Figure 6 and 7 respectively. The best area in terms of signal-to-noise is found between 2-5 Hz for surge at higher acceleration amplitudes. The lower bound is related to measurement noise from the acceleration sensors. The upper bound is related to the first eigenfrequency and at some point the signal-to-noise ratio is limited by the maximum platform capabilities in terms of bandwidth. The plots for pitch and heave show a similar result. Highest signal-to-noise ratio for these measurements is ~630 and is found at ~4 Hz.

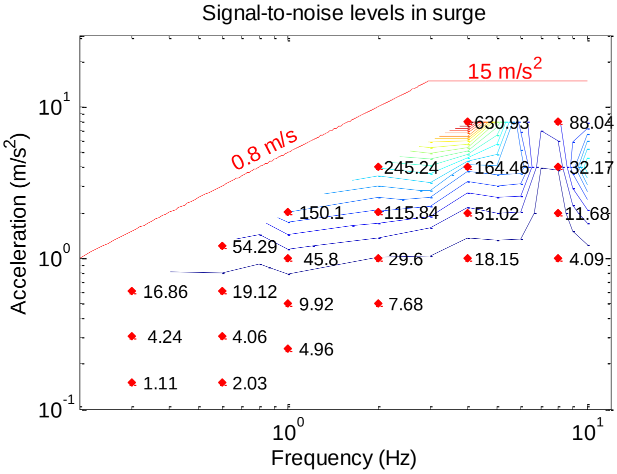

Figure 6. Signal-to-noise levels in surge

The signal-to-noise contour plot for roll acceleration shows the most promising area for measurements between 2-10 Hz when the platform becomes acceleration limited instead of velocity limited. The upper bound will again be related to the first eigenfrequency, but is not visible in this plot as the first eigenfrequency was found to be much higher. The results found for pitch and yaw are almost identical to the result found for roll. Highest signal-to-noise ratio for these measurements is ~150 and is found again at ~4 Hz.

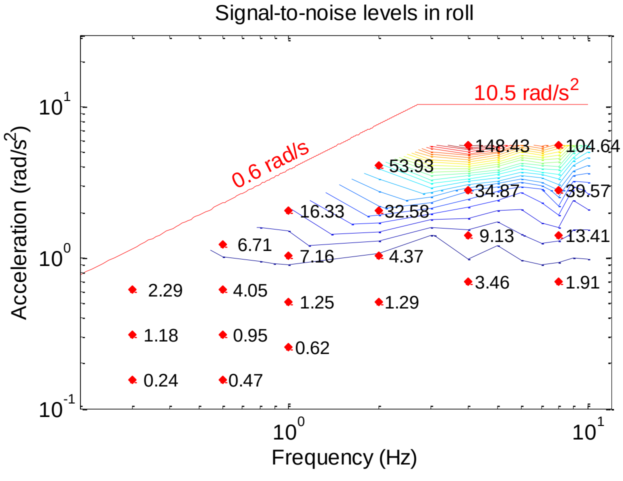

Figure 7. Signal-to-noise levels in roll

Control loading frequency response

Results 1 - Human

Figure 8 shows the frequency response plot for the control loading force loop with a human holding the steering wheel. The result for the two different input excitations described before show similar behaviour.

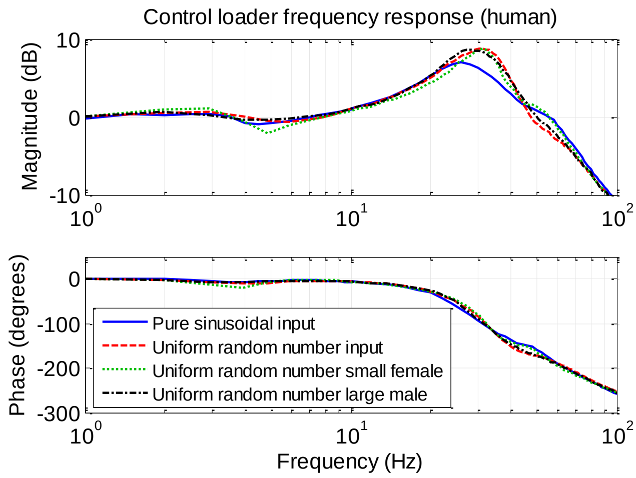

Figure 8. Control loader frequency response (human)

A bandwidth between 50 and 60 Hz is achieved and the first eigenfrequency is found to be approximately 30 Hz. The "dip" between 1 and 10 Hz is an interesting artifact because the magnitude drops below zero dB which is not expected. The transfer function for the small female shows a less damped response between 1 and 10 Hz compared to the large male response but the magnitude stays above the -3 dB threshold for frequencies between 1 and 10 Hz. The phase-lag at 10 Hz is only 5-8 degrees which decreases to 25-30 degrees at 20 Hz and 75-100 degrees at 30 Hz.

Results 2 - Mechanical bar

Instead of using a human to block the steering wheel, another option is to mechanically block the steering wheel. The force loop frequency response for this setup is shown in Figure 9. The result for both excitation signal methods is again similar, except between 14-35 Hz. Here the "uniform random number" method should be trusted, because for the "pure frequency" method the torque actually changed direction and the steering wheel came loose from the mechanical bar as it is only blocked in one direction.

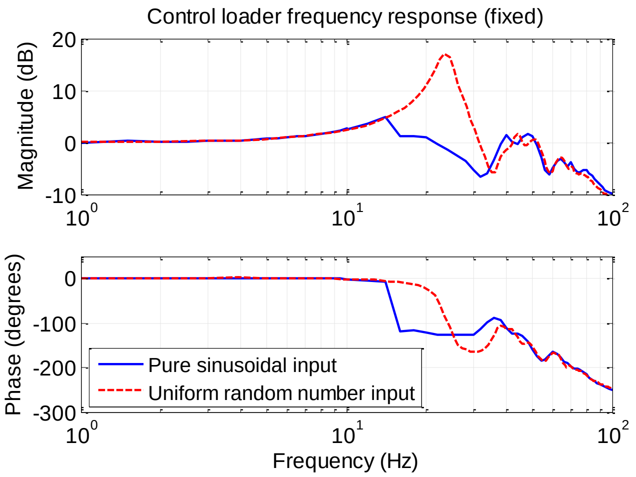

Figure 9. Control loader frequency response (fixed)

The -3 dB crossing point is found at a lower frequency and is now found at ~33 Hz. The first eigenfrequency is also found at a lower frequency, 24 Hz instead of 30 Hz. However, notice that the "dip" between 1 and 10 Hz is now completely gone. This supports the hypothesis that the "dip" is a human artefact and is caused by the human part of the dynamic control loop. The phase-lag is 2 degrees at 10 Hz and 25 degrees at 20 Hz after which the phase drops rapidly because of the eigenfrequency.

Latency

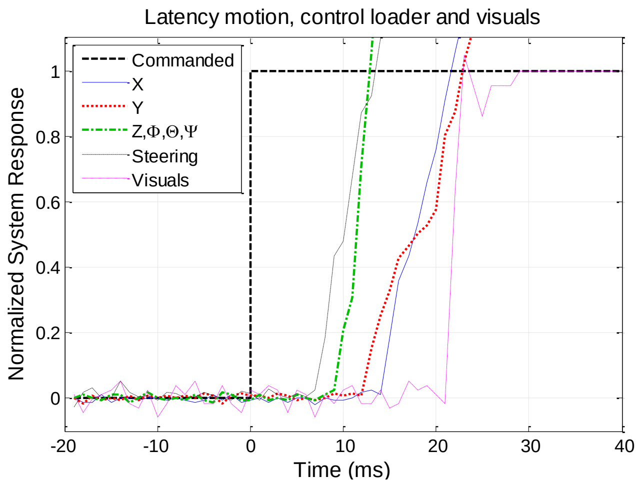

Figure 10. Latency motion, control loader and visuals

Figure 10 shows the latency results for motion, control loading and visuals. Only 7 ms are needed for the control loader to start moving, and after ~10 ms 63% of the setpoint is reached. For the motion platform the step response is measured for all 6DoF independently. The rise time for heave, roll, pitch and yaw is similar and ~2 ms. This makes sense when looking at the describing functions found before. The dynamic threshold for surge and sway movements is higher because the bandwidth is smaller in these directions. An interesting observation is the fact that surge and sway not only have a higher rise time, but also the deadtime is found to be higher.

As expected the visuals have the highest latency, but this is still less than 30 ms in the fastest setup. The visuals are projected by five Barco projectors with ~19 ms latency according to the specifications (and 18.2 ms as verified with a Leo Bodnar visual latency tester). This means that the total rendering time including transport delays is only ~10 ms. The small peaks measured between -20 and 20 ms seem to be related to the 120 Hz or 8 ms refresh rate of the individual three colours (RGB) of the projector. These oscillations are visible when the projector displays a white- or coloured image, but not when a black image is projected.

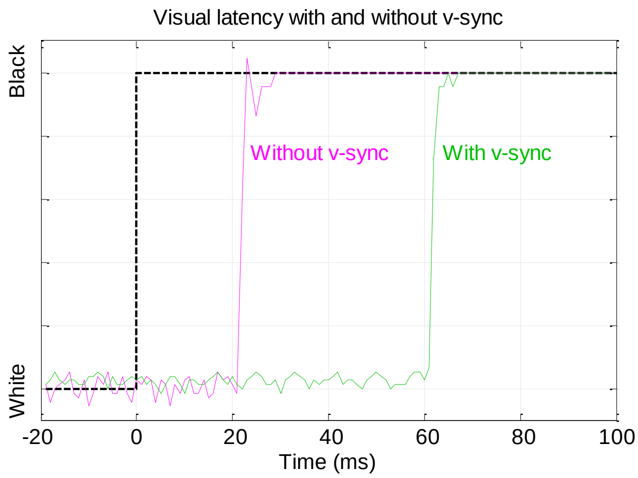

Figure 11. Visual latency with- and without v-sync

Unfortunately the total delay is more than doubled when vertical synchronization is used, see Figure 11. The same result was found by [Str07] using a similar NVIDIA Quadro graphics card. Since image quality is nearly free of tearing without vertical synchronization, this option is not required with these projectors. Without v-sync the number of rendered fps was equal to ~256 and with v-sync this is limited to 120 as the projectors run at 120 Hz.

Discussion

In this study the performance of a Cruden simulator for subjective evaluation of vehicle dynamics at AUDI AG was performed. Describing functions, crosstalk and signal-to-noise ratios were measured for the motion platform in 6DoF. Two methods to measure control loader frequency response were analysed and for both methods two types of input excitation were used and compared. Latency analysis was performed for motion, control loader and visuals using step inputs. The results show satisfactory performance characteristics in all areas, but there is always room for improvement.

Eigenfrequencies for the motion platform are found to be 16 Hz in sway direction and 22 Hz in surge direction with corresponding magnitudes of 16 and 13 dB. For optimal use of the platform it would be desirable to lower the frequency response magnitude at these eigenfrequencies, to effectively increase the usable frequency range right until the bandwidth value. To shape the magnitude of the transfer functions of the motion system it would be possible to implement an inverse dynamic controller as suggested by [Fis09]. However, these type of modifications can increase phase-lag and have to be used with care to avoid singularities which can cause system instability. Crosstalk also becomes an important topic when using the platform closer to the eigenfrequency. A different measurement setup could be adopted to obtain better results for crosstalk, avoiding the influence of sensor placement. The tests should be repeated with correct height of the centres of rotation to be able to quantify crosstalk between pitch and surge and roll and sway. Note that hysteresis and reversal bump effects are not part of the performance analysis in this paper. Because the actuators are electric and not hydraulic, these effects are assumed to be negligible for the application of this particular motion platform. However, this could still be a topic for future research. Future work is planned on measuring the response for a standard Cruden top platform, to investigate the influence of the custom car mock-up parts mounted at AUDI AG.

The control loader results described in this paper show bandwidth values of more than 30 Hz both with- and without the human in the loop. Two methods were investigated and compared. In general the random noise method is preferred as this method is quicker, provides similar results and post-processing is easier as it is not required to cut out the specific sections for each frequency. Additionally there is no risk of too high amplifications causing the torque to change direction resulting in an incorrect torque measurement. The effect of the adjustable steering column for this particular simulator is not included in the results in this paper. The influence of this mechanical linkage will be investigated at a later stage. The implementation of a dynamic controller as suggested by [Fis09] could also be an interesting topic for future research.

In terms of latency the bottleneck was found to be in the visual system, although this latency is still less than 30 ms. Yet there is nearly no possibility of improvement as most of the delay is the result of the projectors. One possible improvement would be the implementation of predictive filters for the visual system as proposed by [McF86, Ste13]. Before doing so one should compare the measured delay of each simulator subsystem with the perceived delay for each subsystem using human perception models. Synchronization performance of the different visualization channels for the five projectors is also an interesting topic for future research.

Audio was not analysed in this project so far. Engine sound, wind noise and tyre squeal as a noise related to the chassis are important for driver immersion and simulator fidelity. The quality and latency of the audio channels is one of the next topics for investigation.

Conclusions

The motion platform shows satisfactory characteristics in the frequency range of interest, with a magnitude between 0 and 1 dB for 0-10 Hz, less than 5 degrees phase-lag at 5 Hz and only 10 degrees phase-lag at 10 Hz. Eigenfrequencies are found at 16 and 22 Hz for sway and surge direction respectively which effectively limits the bandwidth in these directions to these frequencies. The bandwidth for heave, roll, pitch and yaw acceleration was outside the measurement range and is more than 40 Hz. Maximum crosstalk between 1-10 Hz is only 3.5% but this increases significantly near the eigenfrequencies. Part of this crosstalk is assumed to be related to sensor placement.

The control loader achieves a high bandwidth of more than 50 Hz with a human in the loop. However, the useable range is limited to approximately 30 Hz by the first eigenfrequency. A small dip in magnitude was found between 0-10 Hz with a human in the loop. This dip is not visible in the results for the fixed setup using a mechanical bar between the spokes to block the steering wheel. Thus, this is caused by the human body characteristics that are part of the dynamics in the control loop. Correction is not needed as the human is also part of the control loop when driving a normal car.

The dynamic threshold for control loader as well as heave, roll, pitch and yaw for the motion platform is found to be only ~10 ms. This is the end-to-end latency from the driver generating an input until the driver receiving feedback. The response is very fast compared to other motion platforms such as [Nie08, Gra01, Roz07]. Latencies for surge and sway are a bit higher, but still below 20 ms. Visual end-to-end latency is found to be just below 30 ms in the fastest setup, but almost doubles when using v-sync. Almost 2/3 of this latency (18-19 ms) is due to the projectors so there is not much room for improvement of the rendering in terms of latency. The total time for rendering plus transport delays is only ~10 ms. Image quality is similar and still nearly free of tearing without v-sync, so the decision has been made to switch v-sync off for this particular application.

Findings of this research project show that the dynamic characteristics of this mid-size simulator are suitable for subjective evaluation of vehicle dynamics in the chassis development of normal cars.

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