Strain and acceleration measurement and analysis for vehicle seat response reconstruction

Dr. Michiel Heyns Pr.Eng.

T: +27 12 664-7604
C: +27 82 445-0510
mheyns@investmech.com
Introduction

• Objective:
  – To reduce weighted root-mean-square seat response of the driver and crew seats to specified limit:
  • Limited time - only 1 month
  • Limited resources – only four measurement opportunities on the Gerotek test track
  • Could only change damping of the vehicle suspension – only 3 days for this phase

• Needed an approach using
  – Analytical modelling – optimize damper characteristics
  – Experimental response measurements
  – Laboratory testing and seat characteristic refinement
Method

Best damping
- Non-linear time and state dependant transient equations – solve by fixed step Fourth Order Runge-Kutta
- Response simulation with pseudo-random theoretical road profile
- Measure vehicle responses
- System ID and road profile reconstruction
- Verify road profile accuracy by calculating responses and compare
- Damping factor sensitivity analysis – optimal damping factor
- Verify on the test track

Seat weighted RMS
- Select test track
- Seat and seat mount response measurements
- Test rig assembly
- Reconstruct seat mount vibration in the laboratory
- Iterative testing and seat modification to obtain required seat Weighted RMS
The position of the centre of gravity was determined as follows:
1. Horizontally – from weights measured at rear and front wheels
2. Vertically – estimated from relative weight and CoG’s of the components

Effect of modifications on CoG determined in the same way
Tyre model

Used stiffness and damping coefficient
10% damping factor was assumed
Measured data was used
Linear stiffness coefficient and linear damping coefficient was assumed
Effect of stiction during characterization was assumed negligible

Note, in this case tyre stiffness >> suspension stiffness
Non-linear time and state dependant modelling

\[ \sum M_x = F_{S2}L_F - F_{S4}L_R + F_{S1}L_F - F_{S3}L_R = I_{xx,B} \ddot{\theta}_{x,B} \]

\[ \sum M_y = F_{S2}L_1 + F_{S4}L_2 - F_{S1}L_1 - F_{S3}L_2 = I_{yy,B} \ddot{\theta}_{y,B} \]

\[ \sum F_z = F_{S1} + F_{S2} + F_{S3} + F_{S4} = m_B \ddot{z}_B \]
Axle mathematical model

\[ F_{T3} = (z_3 - T_{3z})k_T + (\dot{z}_3 - \dot{T}_{3z})c_T \]

\[ F_{T4} = (z_4 - T_{4z})k_T + (\dot{z}_4 - \dot{T}_{4z})c_T \]

\[ T_{3z} = A_{2z} - \theta_y T_{3x} \]

\[ T_{4z} = A_{2z} + \theta_y T_{4x} \]

\[ \sum M_y = F_{T4}L_2 - F_{T3}L_2 + A_{2s1}F_{S3} - A_{2s2}F_{S4} = I_{yy,R} \ddot{\theta}_y,R^{\text{Velocity [m/s]}} \]

\[ \sum F_z = F_{T3} + F_{T4} - F_{S3} - F_{S4} = m_R \ddot{z}_R \]
Damper model

- Rebound/Compression force ratio = typically 1 to 3
  - R/C ration = 1 typical for off-road
  - R/C ration = 3 typical for sport sedan vehicles
  - In this case \( \frac{15030}{6450} = 2.33 \)
Reference damping factor

- Use average of compression and rebound
- Damping factor for the reference vehicle $\xi = 0.27$

$$\xi = \frac{18500}{2 \sqrt{360000 \times \frac{13300}{4}}} = 0.27$$

Question: What is a typical damping factor for off-road vehicles?
Answer:
Trade-off between ride comfort, road holding and road handling, you must compromise, cannot have all
Low value at High Speed and High value at Low speed
Need more as road roughness increase
Race cars need good handling: $\xi = 0.65 - 0.75$
Passenger cars maximized for ride comfort: $\xi \approx 0.25$
Strain gauges & accelerometers

Instrumentation to record acceleration input and response of crew seat

Strain gauges to measure suspension force
Hardware used

• Data Logger:
  – Description: Somat eDaq-Lite
  – Sampling frequency = 2,000 Hz
  – Anti-aliasing:
    • Linear Phase
    • Cut-off frequency = 667 Hz
    • Output data type: 32 Bit Float
• ARX = AutoRegressive model with external input
• Part of PCMatlab System Identification toolbox
  – Model Characterization: \( th=\text{arx}([\text{Response Input}], NN) \)
    • Mathematics: 
      \[
      y(t) + a_1 y(t - 1) + a_2 y(t - 2) + \cdots + a_{na} y(t - na) = b_1 u(t) + b_2 u(t - 1) + \cdots + b_{nb} u(t - nb + 1) + e(t)
      \]
      • The function solves the \( a \) & \( b \) coefficients
      • \( e(t) \) is external noise
  – Simulation: \( \text{Response}=\text{idsim(Input, th)} \)
ARX forward system identification was used: 
\[ \text{th} = \text{arx}([y \_\text{temp} \ u \_\text{id}], \text{NN}); \]
Check accuracy by reconstructing acceleration responses used: 
\[ y \_\text{id} = \text{idsim}(u \_\text{id}, \text{th}); \]
The mode order was \( [N_{\text{input terms}}, N_{\text{output terms}}, N_{\text{delay}}] = [1,1,0] \)
Reverse-inverse ARX model to reconstruct road input
Correlation coefficient = 0.87

Only a slight shift in mean of signal required to give good perception of fit
Road profile reconstructed from recorded accelerations

Calculated road profile $z(x)$ Measured $\ddot{z}$

Measured acceleration used to calculate road profiles: $y_{idRIm}=idsim(u_{idRIm},thRI)$;
The function does not use time step or sampling frequency – time step in reconstructed data = time step of response data used, 200 Hz in this case
Simulations to Weighted RMS

- Use road profile to calculate responses
- ISO 2631 running weighted RMS
- Repeat for damping magnification factor varied from 0.2 to 2.4 in steps of 0.2
  - That is the compression and rebound damper characteristics were multiplied with this factor
  - This will indicate in which direction to adjust the dampers

Tips that enabled 55 simulations of 120s each in 15 hour span on ONE Dell Notebook Computer:
- Read data into memory and limit disk operations to the minimum
- Ensure that all computer threads are used – split algorithms if necessary
- Declare result matrix sizes before starting the simulation
- Limit array values used in interpolations to find instantaneous road profile displacement and velocity
Damping factor sensitivity

Results indicated an increase of 60% (factor 1.6) on damping factor. Damping factor was changed to $1.6 \times 0.27 = 0.43$ (43%).

Response optimal point
Designed to give mounting points exactly as in the vehicle

No natural modes in the operating frequency range

40 kN servo-hydraulic actuator excites the super structure
Objective – reconstruct measured seat frame acceleration

Instrumentation to verify drive signal

This is the measured acceleration, that is also the desired response of the seat frame on the servo-hydraulic actuator
Why High-pass filter – Sine wave?

Non-zero mean causes the trend

Note the non-zero mean

Solution: High-pass filter signals

Servo-hydraulic actuator in displacement control $\rightarrow Drv = HPF \left[ \int_{t=0}^{T} \left[ HPF \int_{t=0}^{T} \ddot{x} \, dt \right] \, dt \right]$
High-pass filter effect on random signals

It is essential to remove the low-frequency “drifts” from the integrated signals.

Unfiltered:
\[ d1x = \text{cumtrapz}(t, d2x); \]

\[ x = \text{cumtrapz}(t, d1x); \]

Filtered:
\[ d1x = \text{filtfilt}(B, A, \text{cumtrapz}(t, d2x)); \]

\[ x = \text{filtfilt}(B, A, \text{cumtrapz}(t, d1x)); \]

Mathematical descriptions:
\[
\text{d1x} = \text{cumtrapz}(t, \text{d2x}); \\
\text{x} = \text{cumtrapz}(t, \text{d1x}); \\
\text{Fs} = 200; \\
\text{HPFCutoff} = 1; \\
[B, A] = \text{butter}(8, \text{HPFCutoff}/(\text{Fs}/2), \text{’high’}); \\
\text{d1x} = \text{filtfilt}(B, A, \text{cumtrapz}(t, \text{d2x})); \\
\text{x} = \text{filtfilt}(B, A, \text{cumtrapz}(t, \text{d1x}));
\]
Test rig (DAC – Hydraulics – Servo-valve – Inertias – AAF – ADC):

- Non-linear frequency dependant responses
  - Natural frequencies
  - Inertia in the oil supply system, etc.
- Therefore, $Signal_{Wanted} \neq Signal_{Drive}$
- Solution:
  - Compensate in the time or frequency domains
  - In this case, frequency domain is sufficient
How was this done

- \( \text{Drive}_i = \text{DesRes} \)
- Measure rig Response \( \text{RigRes} \)
- \( F(f) = \frac{\text{FFT(DesRes)}}{\text{FFT(RigRes)}} \)
- \( \text{Drive}_{i+1} = \text{Re}(\text{IFFT}[F(f) \times \text{FFT(Drive}_i)]) \)
- Iterate until: \[ \frac{|\text{FFT(DesRes)}|}{|\text{FFT(RigRes)}|} \geq 90\% \]

The result is an acceleration drive signal adjusted for the rig frequency response
The objective is to have accurate reconstruction of the dominant peaks in the spectrum.

Motion sickness: 0.1 – 0.5 Hz
Health, comfort, perception: 0.5 Hz – 80 Hz

Measured RMS = 2.04
Rig RMS = 2.05
Driver Seat
32 km/h Rally Track Full Load

Transmissibility function driver seat response/vehicle

Effect of damping on magnitude at resonance

Spectrum of Driver Seat Response acceleration

Spectrum of Driver Seat Input

Spectrum of Driver Seat Response acceleration

Spectrum of Driver Seat Input

Transmissibility more than 1 at f < 6 Hz
Seat resonates
Solution: Increase damping

This seat filters high frequency content
Crew Seat
32 km/h Rally Track Full Load

Note how this seat acceleration follows that of the vehicle body over the frequency spectrum of the time signal.

Seat response is in phase with vehicle
Seat is stiff due to mounting straps
Solution: Replace straps with elastic material
Final remarks

• Servo-hydraulic test rig now used to minimise transmissibility to the seat
  – Adding elasticity to isolate
  – Seat layout design changes

• This process major contribution
  – Enable quick (3 Days) for damper selection for minimum seat weighted RMS response
  – Test rig that enables continuous, cheap, repeatable, reliable reconstruction of vehicle responses at the seat mounts
  – Laboratory verification of seat designs to meet client objectives