Ride Comfort and Suspension Dynamics Analysis of an All-Terrain Vehicle (ATV) using Adams

Prakash Raju Sridharraju | prakashr@student.chalmers.se

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Introduction

This project investigates the vertical ride dynamics of an All-Terrain Vehicle (ATV) using multibody simulation in MSC Adams. Time-domain, frequency-domain, and modal analyses were performed to identify dominant vibration modes affecting driver comfort. Suspension parameter studies were conducted by varying damper and spring characteristics to evaluate ride comfort, stability, and wheel-hop behaviour on stochastic and bump road profiles.

Vehicle Engineering Context

ATVs operate on rough terrain where ride comfort and suspension durability are critical. Identifying resonance frequencies such as heave, pitch, roll, and wheel-hop helps engineers tune suspension parameters to minimise driver vibration and improve vehicle stability.

Keywords: multibody simulation, ATV, ride comfort, modal analysis

A. Time- and frequency-domain analysis

Vertical Ride Comfort Analysis

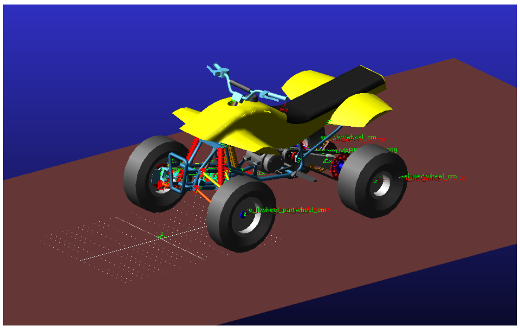


Figure 1: ATV Multibody Model in Adams

The ATV model $terrain\_vehicle3.cmd$ was opened, and a dynamic simulation was run for $100sec$ with a time step of $0.01s$, starting from equilibrium as instructed and the vertical acceleration at the driver's seat was plotted using the measure $frame\_cm\_accZ$.

The FFT shows few clear resonance peaks which can be seen in table-1 which means the vehicle has strong vertical vibrations at these frequencies.

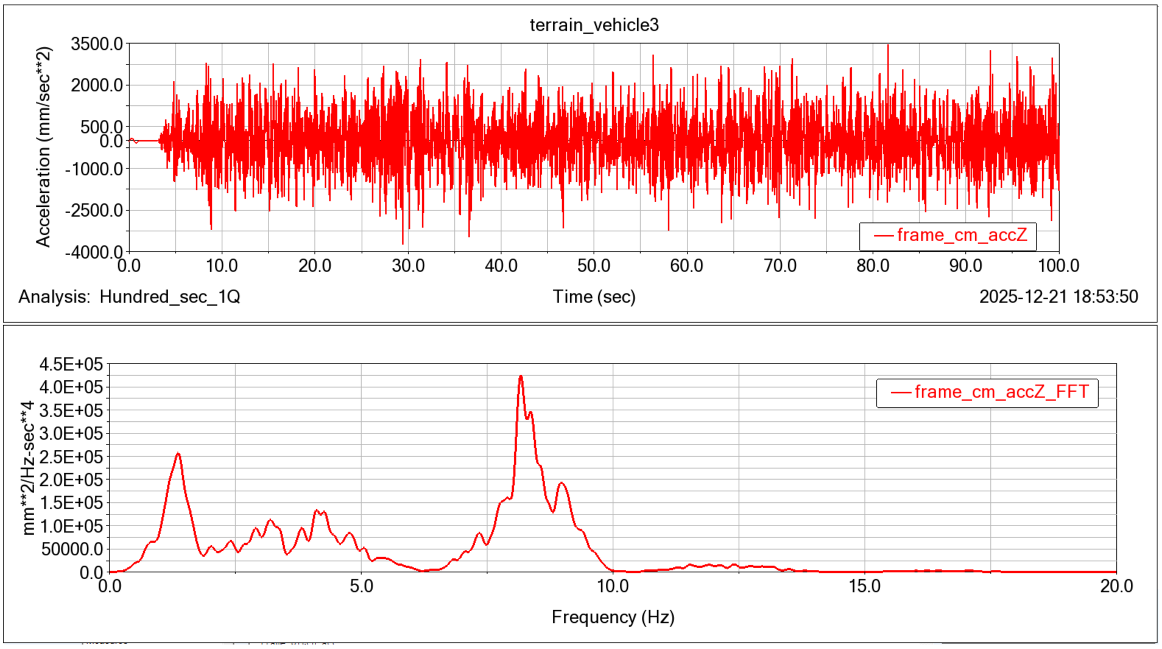


Figure 2: Frame acceleration and FFT plot

S.No	Frequency (Hz)	PSD ($mm^2/Hz.sec^4$)
1	1.355	$2.572 * E^5$
2	3.186	$1.126 * E^5$
3	4.133	$1.344 * E^5$
4	8.166	$4.25 * E^5$
5	8.960	$1.931 * E^5$

Table 1: FFT peaks of Frame vertical acceleration

Modal Analysis of ATV

In the linearised model, five main vibration modes that match the peaks from the FFT. First mode is a pure up-and-down motion of the vehicle frame (heave). Second mode in which the vehicle frame pitches forward and backward. Third mode side till with respect to x-axis (roll) The last two modes are mainly related to the wheels moving up and down, one for the rear wheels (rear hop) and one for the front wheels (front hop). These modes explains each peak is linked to one of these physical motions of the vehicle.

Mode No.	Mode type	Frequency (Hz)
62	Heave	1.44
63	Pitch	2.52
64	Roll	4.61
65	Rear Hop	8.51
66	Front Hop	8.64

Table 2: Eigenmodes of interest

Suspension Frequency Response

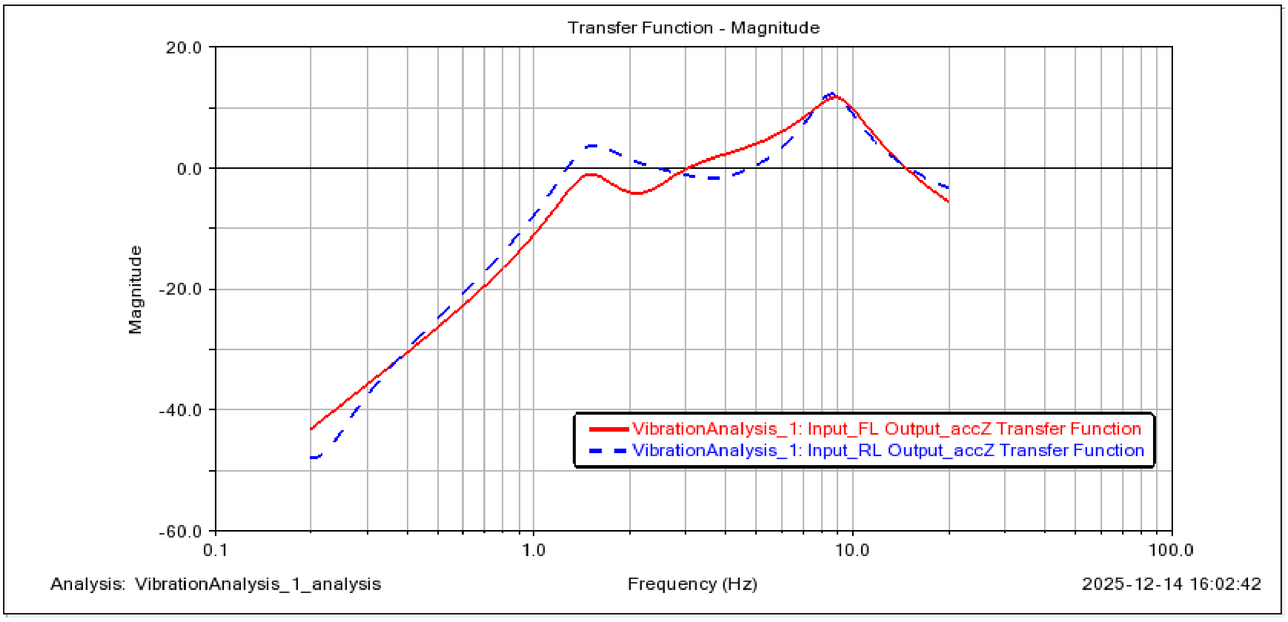
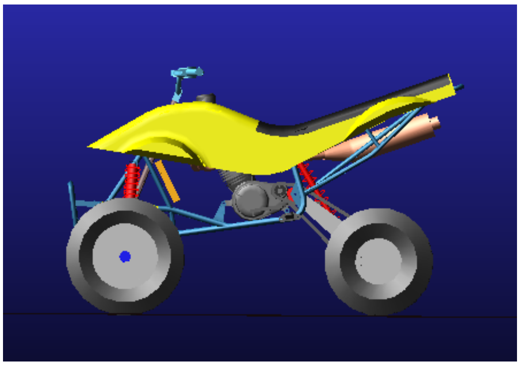


Figure 3: Transfer functions for FL and RL inputs to the vertical acceleration output

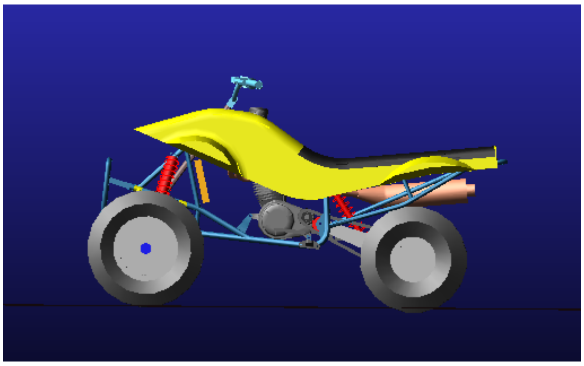
	Sprung		**Unsprung**	
Side	Hz	Mag	Hz	Mag
FL	1.51	-1.05	8.81	11.71
RL	1.54	3.79	8.65	12.2

Table 3: Transfer functions peak values

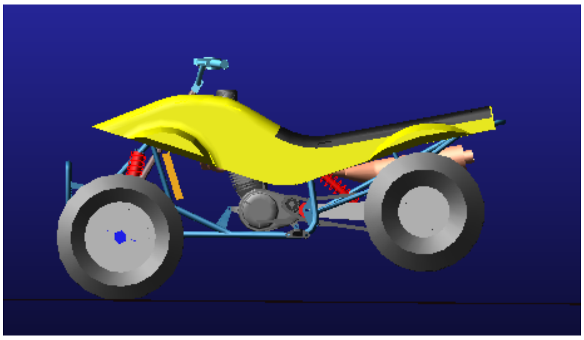
The transfer functions from the front-left and rear-left road inputs to the vertical acceleration at the driver's seat. The curves show how much the driver's acceleration is amplified at different frequencies. From the figure-3 and the table-3, there is a lower-frequency peak, which is the sprung mass motion of the vehicle body, and a higher-frequency peak, which comes from the wheel (unsprung mass) moving up and down. The front-left and rear-left responses look very similar, with almost the same sprung and unsprung natural frequencies.



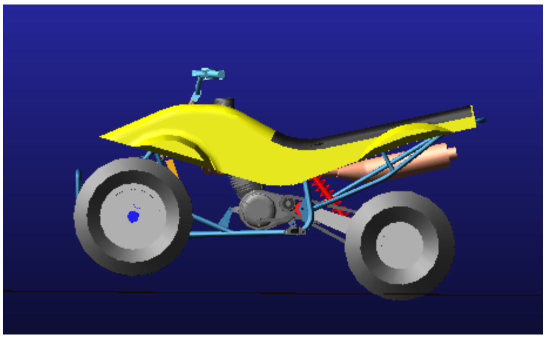
(a) Heave model



(b) Pitch model



(c) Rear wheel hop model



(d) Front wheel hop model

Figure 4: Model shape pictures

The resonance peaks in the transfer-function plots from table-3 approximately match well with the natural modes found in the modal analysis. At low frequency, the first peak corresponds to the sprung-mass heave mode (figure-4a) and the second peak matches the sprung-mass pitch mode (figure-4b). At higher frequency, the peaks match the rear-hop and front-hop wheel modes (figure-4c and 4d), where the rear and front wheel sub-assemblies move up and down relative to the body. When these frequencies are animated, the vehicle motion seen in the mode shapes (heave, pitch, rear hop, front hop, and roll), so the nonlinear FFT results from Question 2, and the forced-vibration transfer functions from Question 4 all shows approximate similarities about the same physical vibrations in the ATV.

Sprung mass heave (62): 1.46 Hz   Sprung mass pitch (63): 2.76 Hz   Vehicle roll (64): 4.62 Hz

Rear hop mass (65): 8.6 Hz   Front hop mass (66): 8.82 Hz

Front Damper Sensitivity Study

When the damping coefficient of the front dampers is reduced, the vehicle response becomes less controlled. Damping is responsible for dissipating vibration energy, so lowering its value allows oscillations to persist for a longer time.

From the frequency-domain results (see Table 4 and Table 6), the main resonance frequencies remain almost unchanged. This is expected, because damping has only a small influence on natural frequencies. However, the amplitude of vibration at resonance increases, especially at the sprung-mass mode. This indicates that the vehicle body experiences larger vertical motions when excited near its natural frequency.

The transfer-function plots clearly show that the lower damping leads to higher peak magnitudes, meaning that road disturbances are transmitted more strongly to the driver. The mode-shape animations confirm this behaviour: the heave and pitch motions become more pronounced and take longer to decay.

Overall, reducing the front damper coefficient makes the ride less comfortable and less stable, even though the dominant vibration modes remain the same.

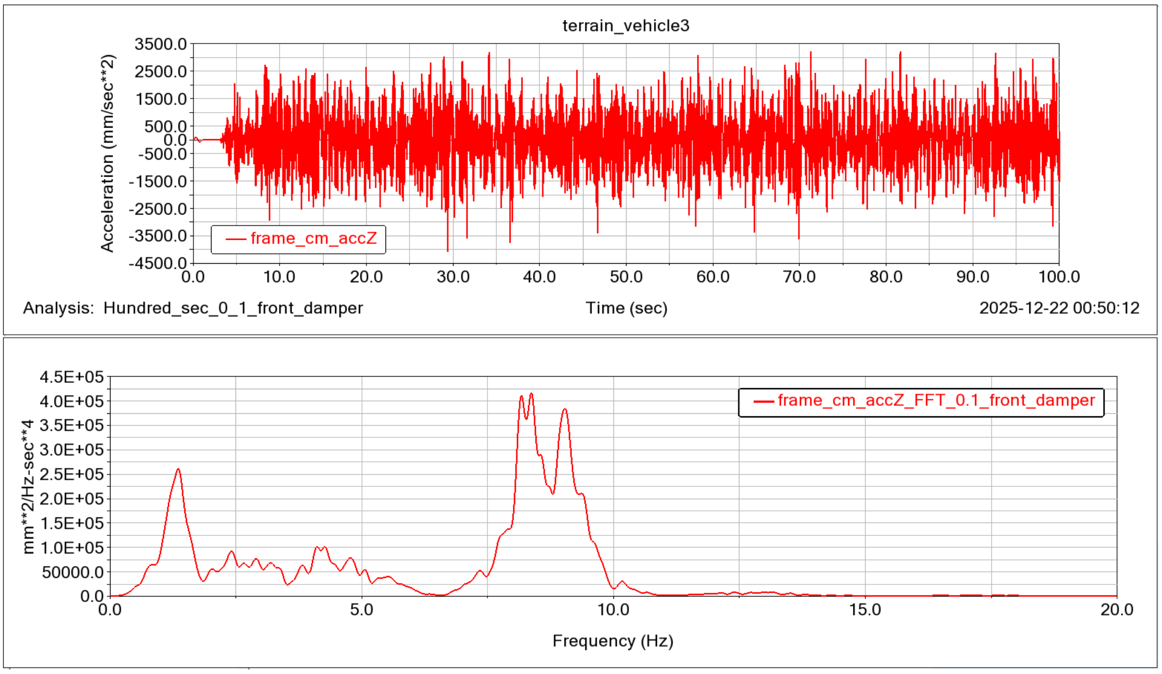


Figure 5: Frame acceleration and FFT plot for $0.1Ns/mm$ front dampers

S.No	Frequency (Hz)	PSD (mm^2/Hz·s^4)
1	1.35	2.615×10^5
2	2.40	9.258×10^4
3	4.10	1.008×10^5
4	8.36	4.169×10^5
5	9.03	3.84×10^5

Table 4: FFT peaks of frame vertical acceleration for 0.1 Ns/mm front dampers

Mode No.	Mode Type	Frequency (Hz)
64	Heave	1.47
65	Pitch	2.61
66	Roll	4.64
67	Rear Hop	8.57
68	Front Hop	9.04

Table 5: Eigenmodes of interest for 0.1 Ns/mm front dampers

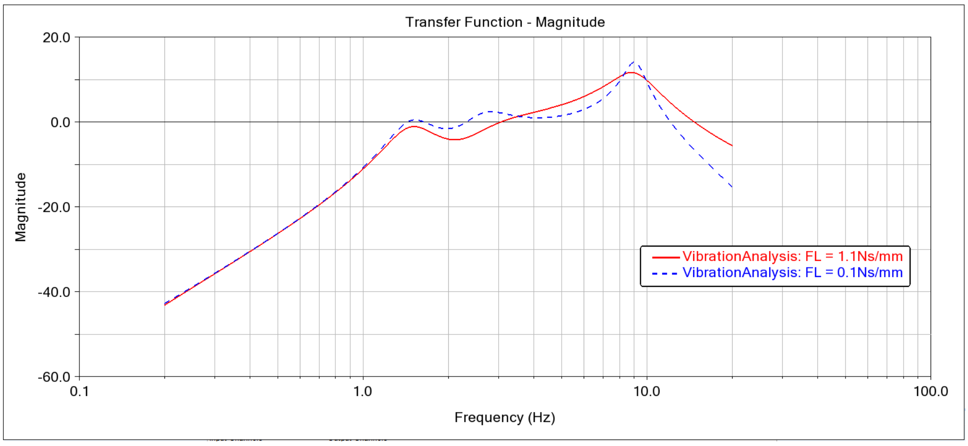


Figure 6: transfer function for new and old damper values (front left)

Dampers front	Sprung		Unsprung	
	Hz	Mag	Hz	Mag
1.1 (Ns/mm)	1.51	-1.05	8.81	11.71
0.1 (Ns/mm)	1.51	0.51	9.05	14.128

Table 6: Transfer functions peak values for new and old dampers

Rear Suspension Stiffness Study

The rear suspension spring stiffness is increased while keeping the reduced front damping. Changing stiffness directly affects the natural frequencies of the system, especially those associated with the rear suspension.

From the FFT and modal analysis results (see Table 7 and Table 8), the rear-related modes shift to higher frequencies. This is because a stiffer spring resists deformation, causing the system to oscillate faster. The sprung-mass modes are also affected, but to a smaller extent.

The transfer-function comparison (see Table 9 and Figure 8) shows that the frequency at which resonance occurs moves upward, and the system becomes more sensitive in a narrower frequency band. The animations of the mode shapes reveal that the rear of the vehicle becomes more rigid, with reduced displacement but higher acceleration levels.

This configuration improves resistance to large rear suspension deflections but can increase vibration transmission at higher frequencies, which may negatively affect ride comfort on rough terrain.

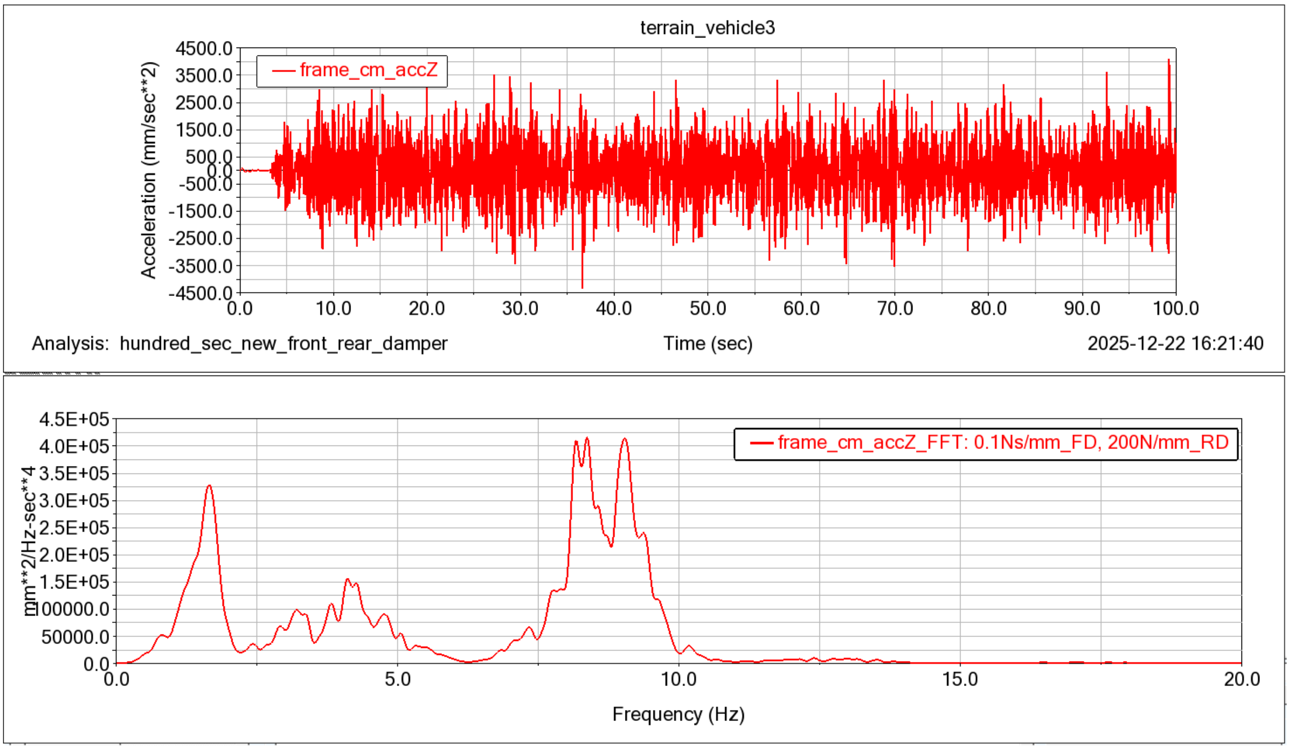


Figure 7: Frame acceleration and FFT plot for $0.1Ns/mm$ front dampers and $200N/mm$ rear damper

S.No	Frequency (Hz)	PSD ($mm^2/Hz{\cdot}s^4$)
1	1.64	3.288×10^5
2	3.21	9.855×10^4
3	4.11	1.562×10^5
4	8.36	4.161×10^5
5	9.03	4.149×10^5

Table 7: FFT peaks of frame vertical acceleration for 0.1 Ns/mm front dampers and 200 N/mm rear damper

Mode No.	Mode Type	Frequency (Hz)
64	Heave	1.8
65	Pitch	2.81
66	Roll	3.05
67	Rear Hop	8.94
68	Front Hop	9.06

Table 8: Eigenmodes of interest for 0.1 Ns/mm front and 200 N/mm rear dampers

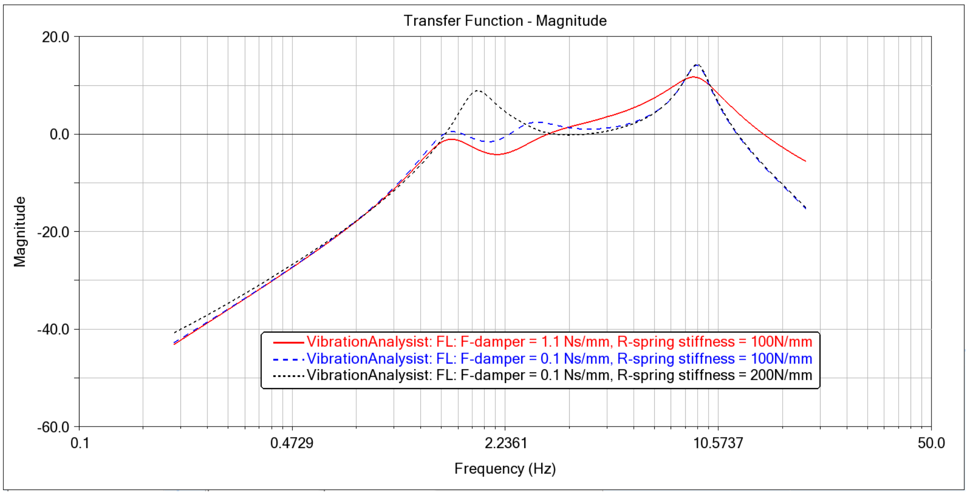


Figure 8: Transfer functions for default, 0.1Ns/mm front and 200N/mm rear dampers

Dampers	Sprung		Unsprung	
	Hz	Mag	Hz	Mag
FD: 0.1 (Ns/mm) **, RD: 100** (N/mm)	1.51	0.51	9.05	14.128
FD: 0.1 (Ns/mm) **, RD: 200** (N/mm)	1.82	8.84	9.05	14.324

Table 9: Transfer functions peak values for 0.1Ns/mm front and 200N/mm rear dampers

Observations

Time-domain and frequency-domain analyses provide complementary insights into vehicle behaviour.

Time-domain analysis is useful for understanding the actual physical response over time, such as how the vehicle reacts to road inputs, how oscillations decay, and how peak accelerations develop. It closely represents what a driver would physically experience and is essential for evaluating transient behaviour.

Frequency-domain analysis, on the other hand, is powerful for identifying dominant vibration modes and resonance frequencies. It allows a clear separation of different physical phenomena (such as heave, pitch, and wheel hop) and makes it easier to study how design changes affect specific modes.

The frequency-domain results helped identify the key resonance frequencies and match them with mode shapes, while the time-domain simulations showed how these modes influence ride comfort in practice. Using both approaches together provides a more complete understanding of the vehicle's behaviour.

B. Rear damper analysis

Damper force study

With the default linear damper (RD:100N/mm, 5Ns/mm, FD: 0.1Ns/mm), the relationship between damper force and swing-arm angle is relatively predictable, so the suspension response follows a consistent pattern when the vehicle runs on the stochastic road.

From the plots in Figure 9, it can be seen that the swing-arm angle varies smoothly while the damper force fluctuates around a stable range, which indicates that the rear suspension is neither too soft nor too stiff in this condition.

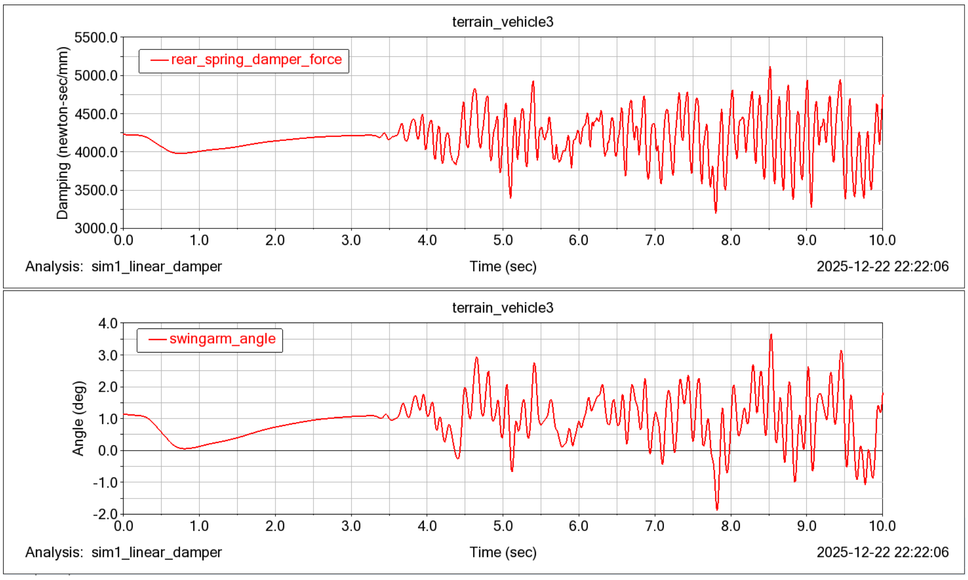


Figure 9: Rear damper force and swing arm angle, plots for linear damper

Nonlinear Damper Performance Study

When the linear rear damper is replaced with the first measured nonlinear damper (see its force–velocity curve in Figure 10), the relationship between swing-arm angle and damper force becomes more complex. The force–angle plot in Figure 11 shows that for some parts of the motion the damper is softer and for other parts it is stiffer, so the force does not increase linearly with motion as before.

Compared to Question 9, the swing-arm angle tends to vary slightly more, and the damper force traces show regions where the damper either allows more movement or resists motion more strongly, depending on the operating range. Overall, this nonlinear damper changes how force is absorbed: it can improve comfort in some ranges by being softer, but it can also lead to sharper force changes in other ranges, which may be felt as more abrupt responses to certain road irregularities.

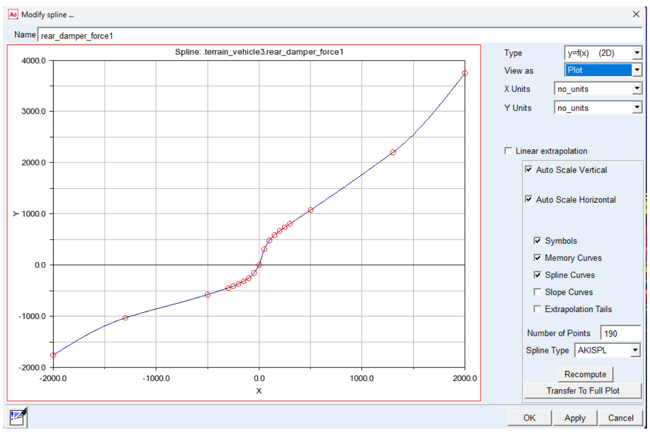


Figure 10: Nonlinear damper-1 curve

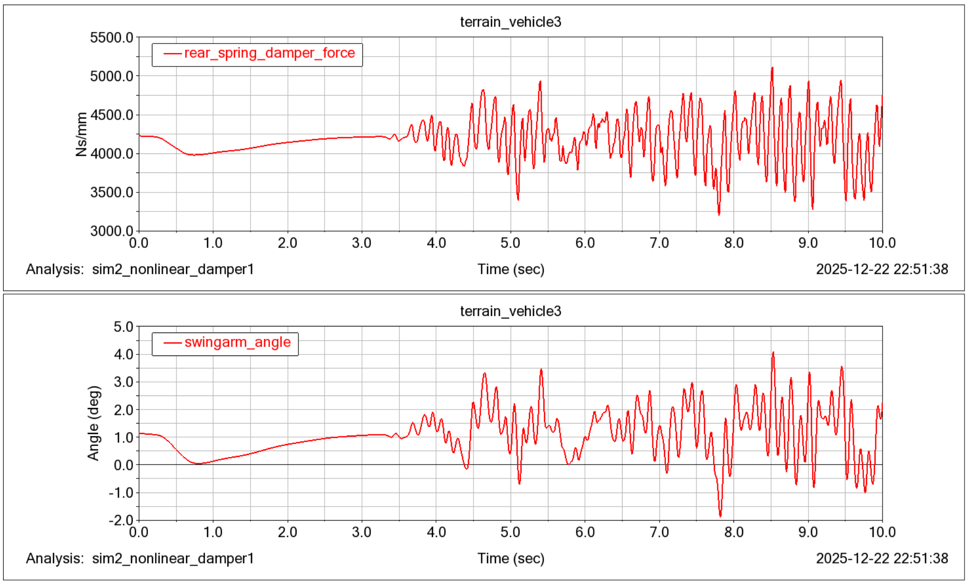


Figure 11: Rear damper force and swing arm angle, plots for nonlinear damper

On the bump road, the suspension is subjected to a more amplified excitation, so differences between linear and nonlinear damping become clearer. With the linear damper, Figure 12 shows that the swing-arm angle response to the bump is more symmetric and the damper force, swing arm angle remains regular, indicating a predictable damping action during compression and rebound.

With nonlinear damper 1 (Figure 13), the bump excitation produces a different pattern: the swing-arm angle can reach slightly larger, and the damper force, swing arm angle shows that the damper reacts differently in compression and rebound or at higher velocities. This means that under a sudden bump the nonlinear damper redistributes the damping force, it may soften the initial impact but can also concentrate higher forces in certain parts of the stroke, which affects both comfort and the way the rear wheel follows the bump profile.

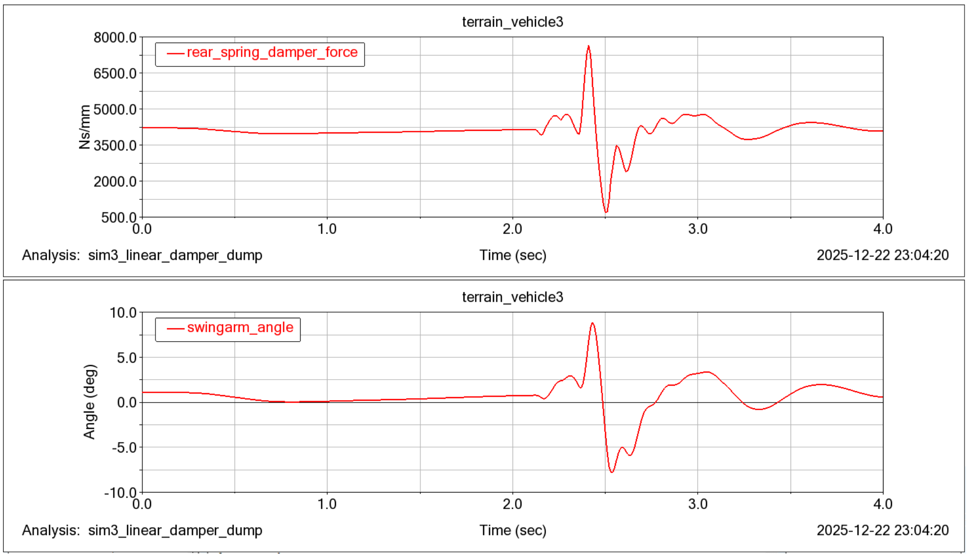


Figure 12: Rear damper force and swing arm angle, plots for linear damper in bump road

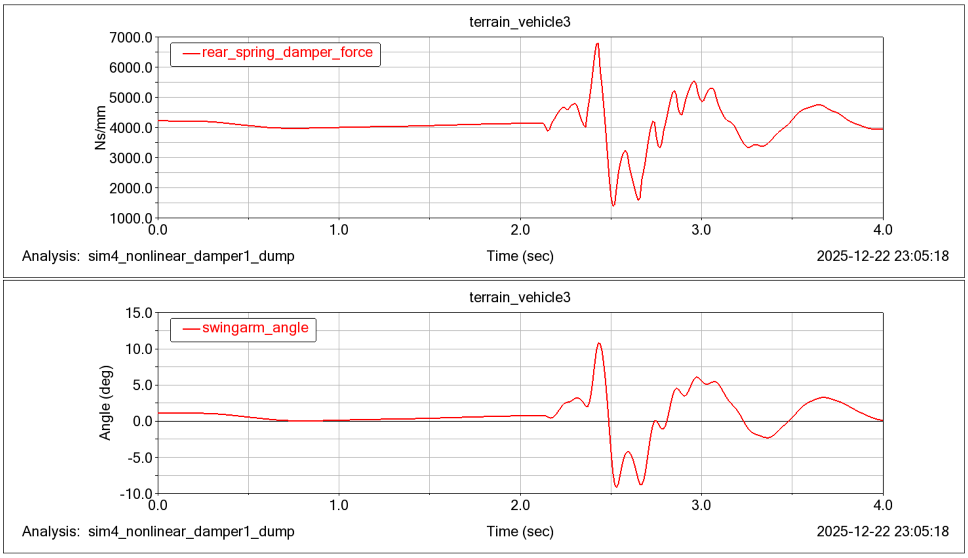


Figure 13: Rear damper force and swing arm angle, plots for nonlinear damper1 in bump road

the second nonlinear damper (Figure 14) adds another level of variation in how the rear suspension responds, both on the straight stochastic road and on the bump road. The plots of damper force, swing arm angle for this damper (Figures 15 and 16) show yet another shape, compared with the default linear damper and nonlinear damper 1, the swing-arm motion and damper forces are distributed differently over the travel.

The overall trends can be understood using the RMS and average values in Tables 10 and 11, which summarise the suspension behaviour for all three damper cases under straight and bump conditions. Without focusing on exact numbers, the key observations are:

- The damper forces remain of similar overall level for all three configurations, meaning the basic load level in the rear suspension does not change drastically between the damper models.
- The swing-arm angle statistics change more noticeably, showing that the different dampers primarily affect how much and how fast the suspension moves rather than the absolute force level.

In other words, the damper choice mainly tunes the character of the rear suspension motion:

- The linear damper gives a smoother and more predictable response on both the straight and bump roads.
- Nonlinear damper 1 lets the suspension move in a less uniform way, so it can feel softer in some parts of the stroke and sharper in others.
- Nonlinear damper 2 adjusts this balance again and slightly changes how both random road inputs and the bump are filtered, as seen from the RMS/average values and the plots.

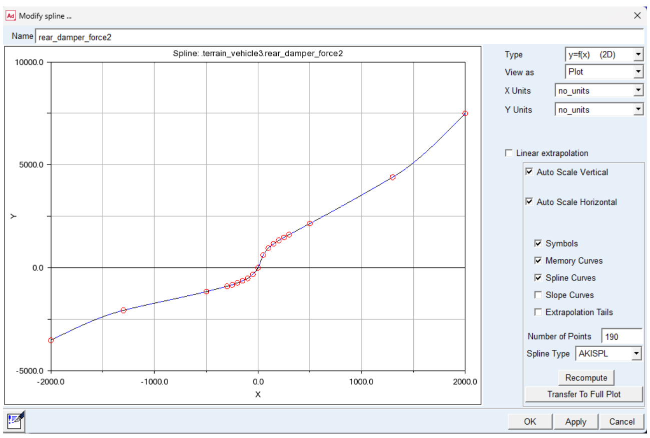


Figure 14: Nonlinear damper-2 curve

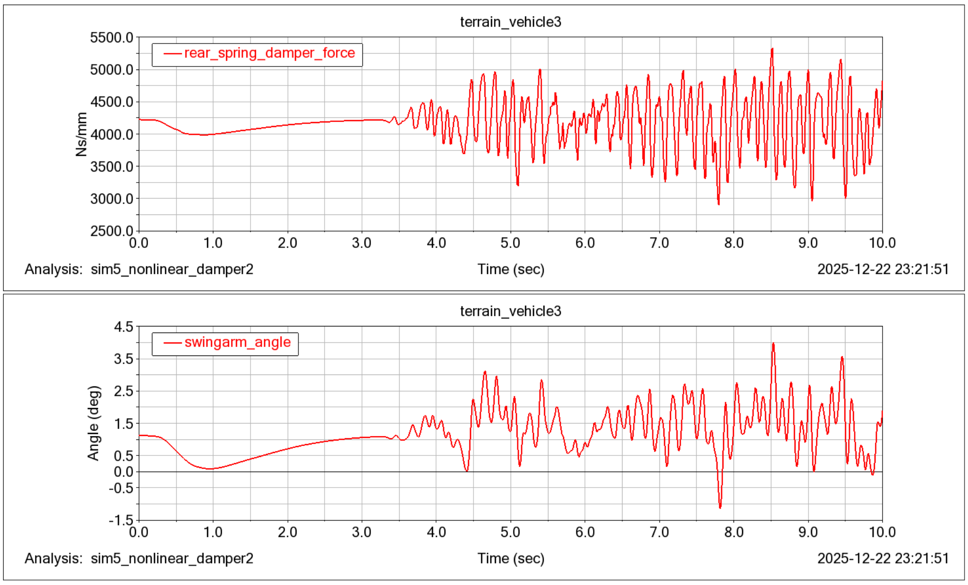


Figure 15: Rear damper force and swing arm angle, plots for nonlinear damper-2

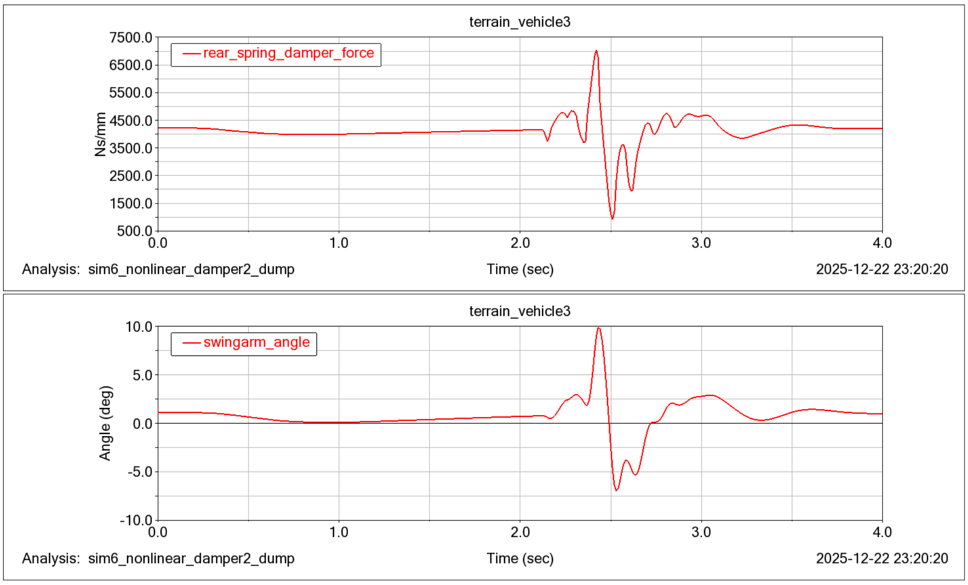


Figure 16: Rear damper force and swing arm angle, plots for nonlinear damper-2 in bump road

	Default		**Damper 1**		**Damper 2**	
RMS values	Straight	Bump	Straight	Bump	Straight	Bump
Damper force (Ns/mm)	4189.87	4165.65	4183.23	4160.85	4191.03	4156.24
Swing Arm Angle (deg)	1.17	2	1.36	2.68	1.37	1.93

Table 10: RMS values for different damper configurations under straight and bump conditions

	Default		**Damper 1**		**Damper 2**	
AVG values	Straight	Bump	Straight	Bump	Straight	Bump
Damper force (Ns/mm)	4178.5	4123.59	4173.87	4113.56	4174.94	4121.76
Swing Arm Angle (deg)	0.9	0.66	1.07	0.77	1.17	0.77

Table 11: AVG values for different damper configurations under straight and bump conditions

Key Findings

- Identified dominant ATV vibration modes: heave (1.4 Hz), pitch (2.5 Hz), roll (4.6 Hz), and wheel-hop (8–9 Hz) affecting driver comfort.
- Reduced front damping increased vibration amplitude without changing natural frequencies, reducing ride comfort.
- Increasing rear suspension stiffness shifted rear-hop frequencies higher, improving deflection resistance but increasing high-frequency vibration transmission.
- Nonlinear rear dampers altered force–motion characteristics, showing trade-offs between comfort and stability on stochastic and bump roads.