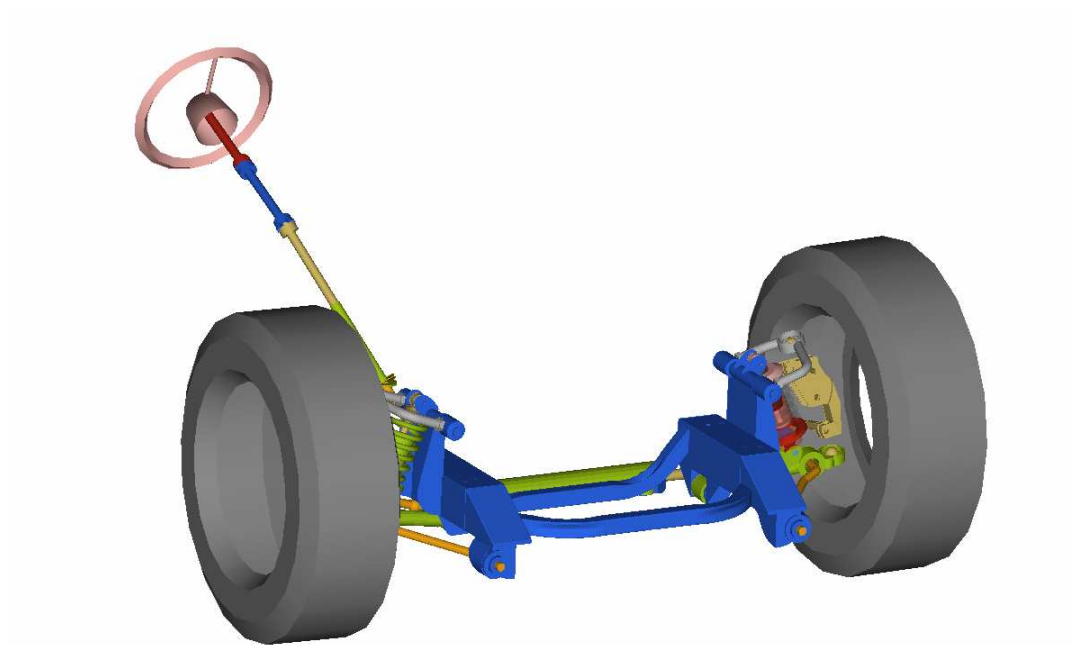Project: **CO0048**

Re: **FX-HR Holden Front End**
- 800kg axle rating
- manufactured after August 2010

Stress Analysis & Geometry Assessment

Prepared for: **V6 Conversions**

Date: **2nd December 2010**

By: **Brett Longhurst**
BEng (Mech) Hons1

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1. SUMMARY

This report outlines a programme of modelling undertaken to assess strength and geometry changes of the front subframe, suspension and steering systems for a bolt in front end kit to suit FX-HR model Holdens. The suspension system has been assessed against the guidelines provided in Vehicle Standards Bulletin 14 (VSB14) of the National Code of Practice for Light Vehicle Construction and Modification (NCOP).

This report provides an assessment of the overall design only, and does not consider the quality of manufacture. It is assumed that all welds are carried out by a suitably qualified person to the relevant Australian Standard. Furthermore, this report is only applicable to the component configuration and variants listed in Section 3.1. Any components or variants of the design not listed in Section 3.1 are not covered by the assessment provided in this report.

Finite Element Analysis (FEA) and Multi Body Dynamics (MBD) are computer simulation techniques that have been used to simulate the suspension system. The following load cases were considered for the assessment of the system, all of which meet or exceed those specified in VSB14:

- Bump Load (4g vertical)
- Braking Load (2g vertical combined with 1.2g longitudinal)
- Overturning Load (2g vertical combined with 2.5g lateral)
- ± 50mm travel from static ride height (geometry assessment only)
- ± 360deg steering wheel rotation (geometry assessment only)

The analysis results indicate that the front end system meets the load requirements set out in VSB14 for the following vehicle configuration limits:

- a maximum front axle weight rating of 800kg
- a maximum track width of 1400mm
- maximum tyre rolling radius of 340mm (equivalent to 245/40R19)

The geometry changes throughout the full range of suspension travel were also assessed and were determined to be satisfactory for this application. These changes in geometry should provide the vehicle with a neutral handling characteristics and a tendency towards understeer. Changes in toe and camber are in the desired directions, within acceptable ranges and maintain a smooth rate of change throughout the range of travel.

Geometry changes throughout the range of steering travel were also assessed and were again determined to be satisfactory for the application.

No potential lockup conditions were identified.

2. ANALYSIS & REPORT OUTLINE

2.1. Purpose

The purpose of this analysis and report is to provide the certification engineer with a general assessment of the front end kit's strength and geometry changes subject to the guidelines set out in VSB14. The certification engineer will then be able to take these analysis results into consideration in conjunction with any other assessment procedures they deem necessary in approving the front end kit for road registration.

This report is not an approval for road registration in itself, and forms only part of an overall assessment process performed by a recognised vehicle certification engineer.

The purpose of using computer simulation techniques such as FEA and MBD in this report, is to provide detailed insight into the performance of a component or system without the expense of physical testing. Simulation also allows for rapid development and design changes, should they be required to meet performance targets, without having to manufacture new parts and perform physical testing on each design iteration. Whilst physical testing can assess the actual performance of the parts being tested, using simulation provides detailed understanding and a high degree of confidence in the performance of the design for a realistic cost to the manufacturer.

Any additional information or analysis required by the certification engineer or any other relevant party, can be requested by contacting Bremar Automotion via the contact details on the report cover page. This includes any items that are stated in this report as being "outside the scope of this analysis", as such items can often be addressed in detail if deemed necessary and appropriate. We are always willing to work in conjunction with the client, engineers and any other relevant parties to achieve a satisfactory outcome for all involved.

2.2. Scope

The scope of this analysis and report is to assess the strength of fabricated or modified components and geometry changes of the overall suspension and steering design. The analysis has been performed based on CAD data supplied by the manufacturer, and no physical inspection of the system has been performed as part of this analysis.

All welds are assumed to be carried out by a suitably qualified welder to the relevant Australian standards. Welded connections are considered to be ideal, and welds are not explicitly modelled as part of this analysis. Modelling of welded connections is considered beyond the scope of this analysis due to the complex nature of welds and their various localised effects on the materials being welded. Without physical test data to correlate detailed weld models to, any attempt to explicitly model welds will usually be inaccurate. Not modelling weld material is also considered to be a conservative approach, due to the fact that the actual part will have more material around welded joints than the FEA model reflects. This will generally result in actual stress levels near welded joints being lower than the stresses predicted by the analysis.

Fatigue has not been addressed in this analysis due to the fact that a physical teardown and inspection has been performed on an identical front end kit that has performed over 135,000km. The report deemed all components to be satisfactory with no evidence of fatigue or overloading. A copy of the report covering this teardown and inspection can be requested from the manufacturer. No fatigue loads are set out in VSB14, so whilst fatigue can be assessed using FEA methods, doing so would require assumed loadings. For this reason, the physical teardown and inspection of a representative unit is considered to be a more realistic assessment of fatigue and durability in this case.

Ball joints and rose joints were modelled as ideal joints and were not assessed for strength as part of this analysis. The lower ball joint is used in a different manner than its original application on the Commodore's McPherson strut arrangement and is subjected to different loads, however physical testing of the ball joint has been performed and has been deemed suitable for this application. A copy of this test report can be requested from the manufacturer.

2.3. Some Comments on Multi Body Dynamics

Multi Body Dynamics is a computational tool that enables the analyst to model the motions and forces within a mechanism under a variety of loadcases. This gives a much better understanding of the performance of the mechanism as a whole, than simple hand calculations may provide. The analysis assumes that all bodies are rigid with no flex in them unless specifically defined. Similarly, all joints are assumed to be ideal with no friction or compliance in them unless specifically defined.

Although MBD allows detailed modelling and calculations to be performed, the analysis is only as good as the data and assumptions upon which it is based. It relies on accurate geometry, properties of flexible joints and bushes where required, support conditions and loadcases to produce an accurate outcome. Typically the geometry can be well defined, and flexible joint and bush properties can be obtained from a variety of sources, however physical testing of specimens of the actual bushes to be used is preferred and will provide the highest degree of accuracy. Representation of the correct support conditions provides the model with accurate constraint data required to ensure the correct transmission of loads and motion through the mechanism.

MBD models should be correlated with physical test data during initial stages of the analysis where possible, to ensure the accuracy of analysis results. No analysis results should be implemented into production without sufficient physical testing of the proposed design having been completed to ensure its suitability for purpose. This is especially true for components that may compromise the safety of the end user or any other party. Bremar Automotion takes no responsibility for the physical testing of analysed components/systems to determine their suitability for use, unless agreed and explicitly expressed in writing.

2.4. Methods Used

Altair Hypermesh was used to create Finite Element Analysis (FEA) models of the system from 3D CAD models supplied by V6 Conversions. Tube and RHS sections were modelled using 2D shell elements, while thread inserts, rod ends and various other components were modelled using 3D solid elements. Typical element size was 3-5mm.

Altair MotionView was used to assemble the MBD model, create force and motion inputs and the required output requests. The model was created in 3D, with flexible bodies used for all suspension arms and subframe components. Using flexible bodies in an MBD model was chosen for this analysis, as this method provides many benefits over a static FEA analysis. These benefits include the fact that no assumptions need to be made about loads or constraints on individual components, and stress results can be output over the whole range of travel, rather than a single static position.

Some components were also analysed using non linear FEA analysis, once the material yield strength was exceeded to give a more accurate representation of stresses and strains beyond the yield strength. A basic bi-linear material model was used in the non linear analyses, and all load magnitudes and directions were extracted directly from the MBD model.

Altair MotionSolve was used as the solver for the mechanism analysis. Since the loadcases are defined as being static, all analyses were run as static or quasi static, where mass and inertial effects of the components is not considered.

3. MODEL GEOMETRY

3.1. Model Components

The suspension geometry for this front end is a unequal length wishbone configuration with a two piece lower control arm and rear mounted steering rack. The MBD model consisted of the following flexible bodies, as shown in colour in Figure 1 below:

   a) Upper Control Arm

   b) Subframe

   c) Tension Rod

   d) Lower Control Arm

Each of these items is described in more detail in Sub Sections 3.1.1 to 3.1.4.

The upright/knuckle, steering rack and tie rods are OEM items from a VN-VS Commodore, and were modelled as rigid bodies. Stresses were not calculated for these components due to the fact that they come from a vehicle with similar suspension geometry, weighing more than those for which this front end is designed. These OEM components are therefore considered to be suitable for this application.

Maximum track was taken to be 1400mm and the tyre's maximum rolling radius of 340mm was used.
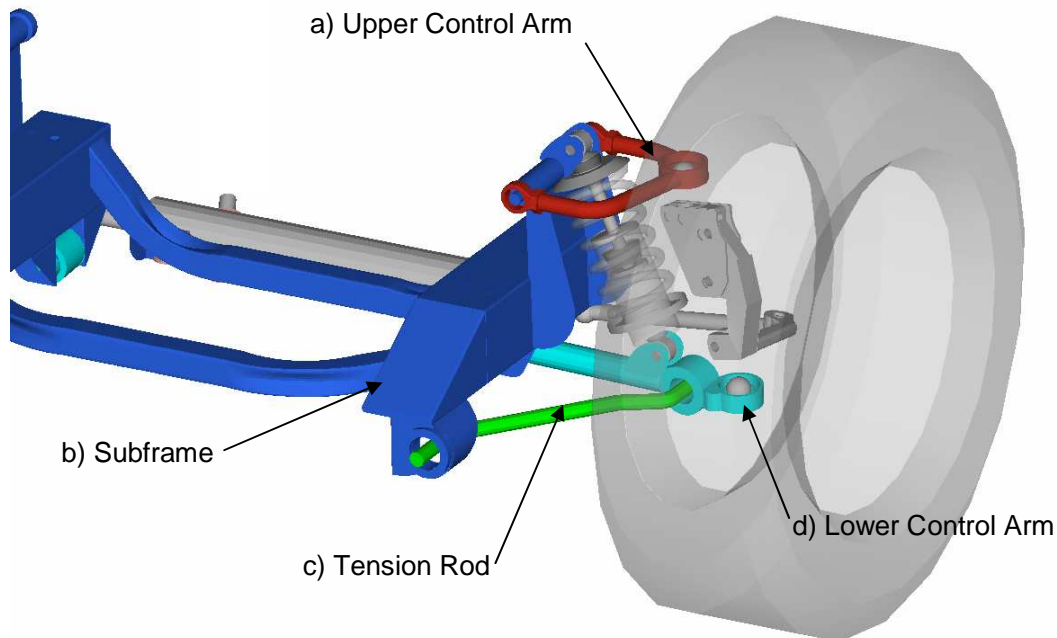

Figure 1: Model Geometry – Flexible Bodies

3.1.1. Upper Control Arm

The upper control arm is a fabricated steel component consisting of ⌀22.2mm x 4.45mm bent tube welded to a machined support for the upper ball joint, as shown in Figure 2. Rose joints are used at the inner mounts to the subframe.
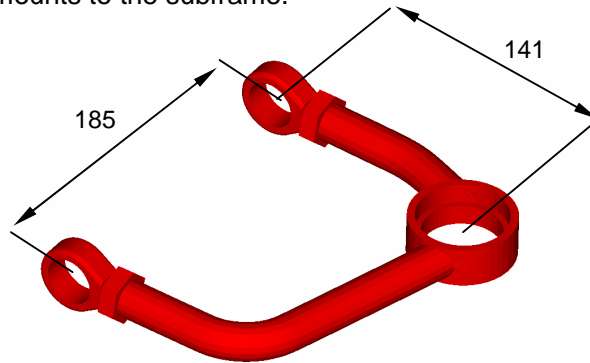

Figure 2: Upper Control Arm

3.1.2. Lower Control Arm

The lower control arm as shown in Figure 3, is a fabricated steel component consisting of ⌀35.0mm x 3.25mm tube, welded to steel machinings for bush and ball joint supports. 6.0mm thick tabs provide the lower mount for the coil over shock absorber.
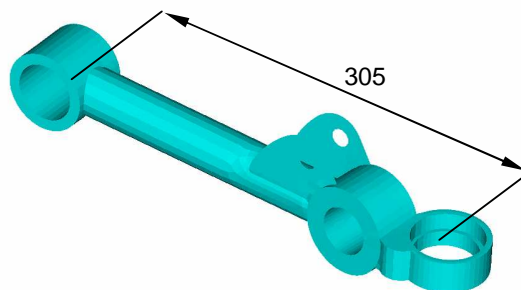

Figure 3: Lower Control Arm

3.1.3.   Tension Rod

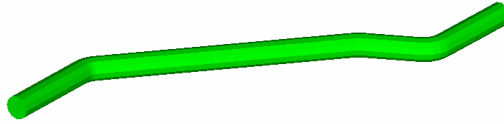The tension rod shown in Figure 4 is ø19mm steel.

Figure 4: Tension Rod

3.1.4.   Subframe

Figure 5 shows the subframe assembly and its various component section sizes. Each colour represents a single material thickness as noted.
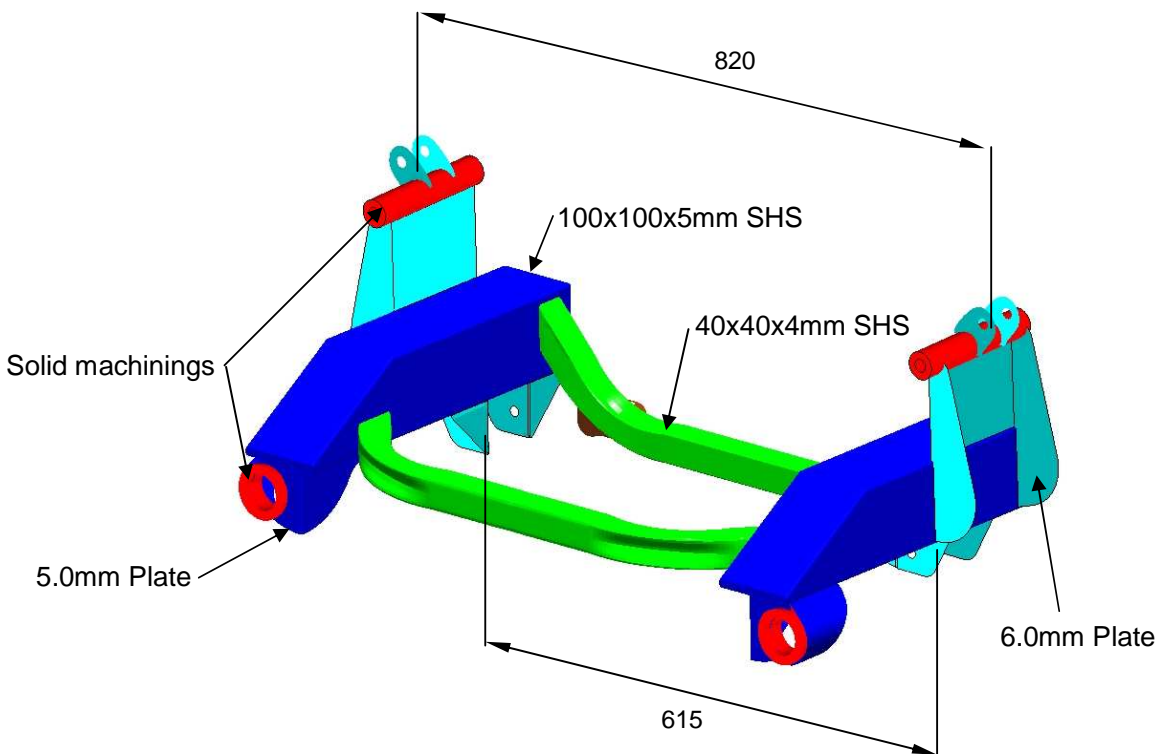

Figure 5: Subframe Assembly

3.2. Component Material & Section Sizes

Material for each flexible component, along with section size and thickness is summarised in Table 1 below:

	Description	Material	Section (mm)
a)	Upper Control Arm	DIN 2391 Tube	Ø22.2 x 4.45
b)	Subframe	Mild Steel	100 x 100 x 5.0 SHS
		Mild Steel	40 x 40 x 4.0 SHS
		Mild Steel	5.0 & 6.0 plate
c)	Tension Rod	Mild Steel	Ø19.0 solid
d)	Lower Control Arm	DIN 2391 Tube	Ø35.0 x 3.25

Table 1: Flexible Component Material & Section Sizes

3.3. Material Properties

Tube used for the control arms was seamless cold drawn tube, grade DIN 2391 45.2. Mechanical properties for each of the materials used is shown below in Table 2:

Material	E (GPa)	Poisson Ratio	Density (T/mm^3)	Yield Strength (MPa)	Ultimate Strength (MPa)
UCA DIN 2391 Tube	210	0.3	7.9e^{-9}	653MPa (act)	710MPa (act)
LCA DIN 2391 Tube	210	0.3	7.9e^{-9}	355MPa (min)	470MPa (min)
				472MPa (act)	607MPa (act)
40x40x4.0 SHS	210	0.3	7.9e^{-9}	510MPa	
100x100x5.0 SHS	210	0.3	7.9e^{-9}	490MPa	
Mild Steel Plate	210	0.3	7.9e^{-9}	290-480MPa	420-540MPa

Table 2: Material Properties

3.4. Model Joints

Ball joints and rose joints were modelled as idealised rigid joints with no compliance and no friction. Some joints such as the inner lower control arm and tension rod connection joints use Nolathane bushes, and these joints were modelled as compliant bushes with varying stiffness in each of the relevant translational and rotational directions.

Actual bushing stiffness values were not provided, and it was considered beyond the scope of this analysis to conduct physical testing to determine bush stiffness. Instead, assumed typical bush stiffness values were used, and a sensitivity analysis using bush stiffnesses 10x and 0.1x the original values, was performed during the geometry assessment to determine the effect of bush stiffness on geometry changes.

4. LOAD CASES

4.1. Loadcase Summary

The following load cases were considered for the stress analysis of the suspension system, all of which meet or exceed those specified in VSB14:

	Description	Load Applied
1	Bump Load	4g vertical
2	Braking Load	2g vertical combined with 1.2g longitudinal
3	Overturning Load	2g vertical combined with 2.5g lateral
4	Geometry Assessment	± 50mm travel from static ride height
5	Geometry Assessment	± 360 deg steering wheel rotation

Table 3: Load Cases

In all loadcases, "g" refers to static load at tyre contact patch. The maximum front axle weight was taken to be 800kg, which equates to a static 1g load of 4kN at each front tyre contact patch.

5. STRESS ANALYSIS RESULTS

5.1. Bump Loadcase

The VSB14 loading specified for the Bump loadcase applies a 4g vertical load to the wheel. This is considered to be an extreme load event, effectively placing the whole vehicle mass on one wheel. In such an extreme loading event, it would be expected that some suspension components may be damaged. For this reason, the material's ultimate strength (UTS) will be used as the stress limit rather than the yield strength. This assumption means that components may bend and remain deformed, but will not fracture and all components will remain attached to the vehicle.

The MBD analysis for the Bump loadcase showed stresses above the respective material yield strength in the subframe and lower control arm, so these components were subsequently analysed using non linear FEA techniques to determine stress and strain results.

Figure 6 below shows the stress contour for the subframe for the Bump loadcase, while Figure 7 shows the stress contour for the lower control arm.
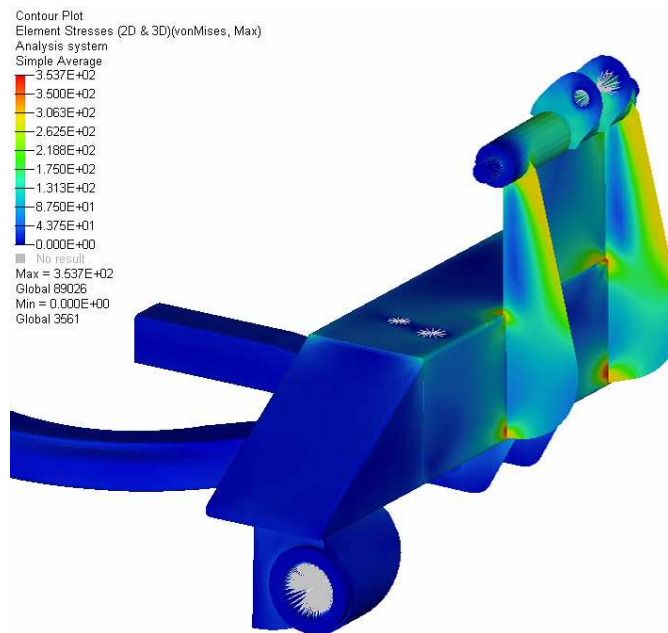

Figure 6: Von Mises Stress Contour – Subframe, Bump Loadcase

Figure 6 shows some small areas of localised stress where the shock tower meets the main 100x100 SHS subframe members. Maximum stress in these areas is 354MPa, which is below the specified yield strength for the SHS and within the range of yield strength for the 6mm plate. All stresses are below the relevant UTS, with maximum strain in these regions of around 1.7%. These stress and strain values are considered acceptable for this loadcase.
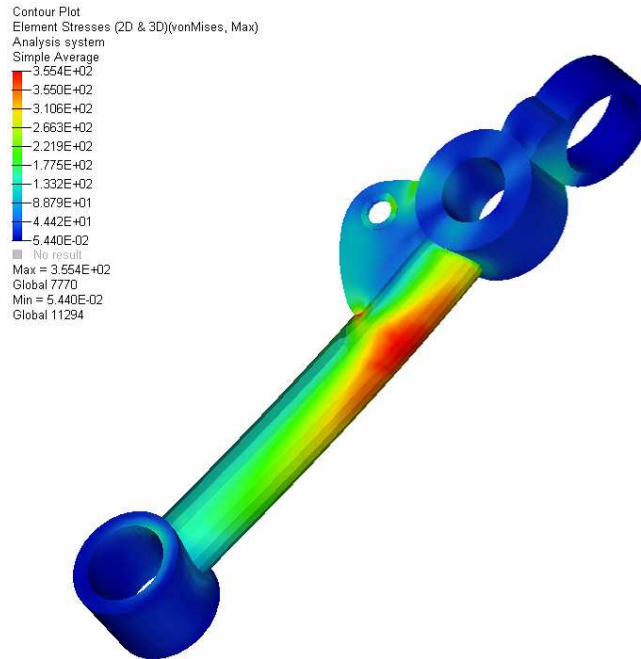
Figure 7: Von Mises Stress Contour – Lower Control Arm, Bump Loadcase

Figure 7 shows the deformed shape of the lower control arm for the bump loadcase. Deformation is scaled up 20x for clarity. The maximum contour value is set to 355MPa, which is the minimum yield strength for the tube material used in the lower control arm.

Maximum stress in the arm is 355MPa which is located on the underside of the main tube portion of the arm and is equal to the material yield strength of 355MPa but below the UTS of 470MPa. Plastic strain in this region is less than 0.1%, which is below the maximum elongation of 10%, suggesting the part is still well below the point of fracture.

Permanent plastic deformation at the ball joint is 0.004mm, which can be considered negligible and will not affect the functionality of the part.

These results indicate that under the prescribed loading, the part may bend slightly but should remain in a serviceable condition and will not fracture, meaning all components will remain attached to the vehicle.

All other components showed stresses below their material yield limits for this loadcase.

5.2. Braking Loadcase

The VSB14 loading specified for the Braking loadcase applies a 2g vertical load combined with a 1.2g longitudinal load to the wheel. Parts should not remain permanently deformed under this loading, so the material yield strength has been used as the upper limit for stress values.

The upper and lower control arms showed the highest stresses for the Braking loadcase, as shown below in Figure 8 & Figure 9.
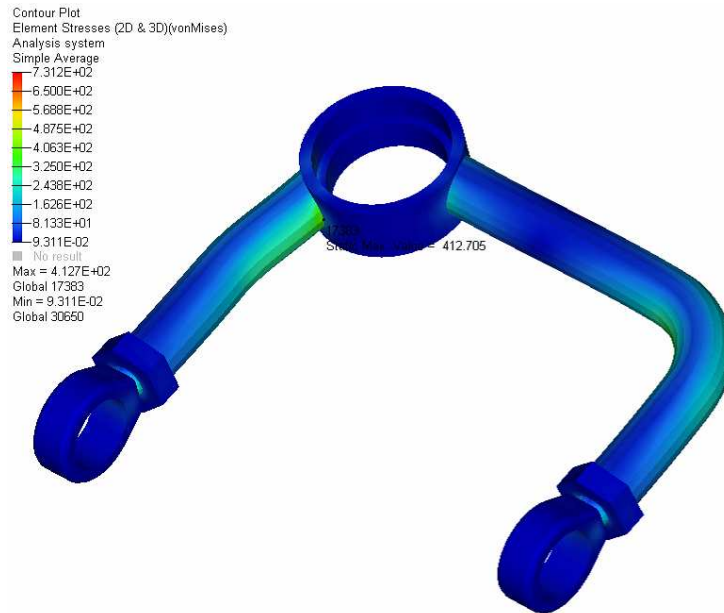

Figure 8: Von Mises Stress – Upper Control Arm, Braking Loadcase

Figure 8 shows maximum stress of 413MPa in the tube of the upper control arm, which is below the material yield strength of 653MPa and is considered acceptable for this loadcase. This stress occurs near where the rear tube meets the ball joint cup. The maximum contour value is set to 650MPa, which is the minimum yield strength for the tube material used in the upper control arm.
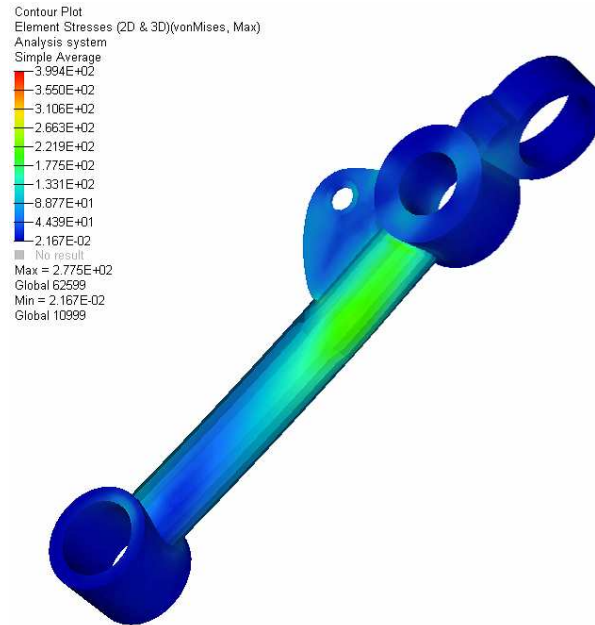
Figure 9: Von Mises Stress – Lower Control Arm, Braking Loadcase

Figure 9 shows the maximum stresses in the lower control arm of 278MPa, which is below the material's minimum yield strength of 355MPa and is therefore considered acceptable for this loadcase. This maximum stress occurs in the lower side of the tube below the shock absorber mount.

All other components showed stresses below their material yield limits for this loadcase.

5.3. Overturning Loadcase

The VSB14 loading specified for the Overturning loadcase applies a 2g vertical load combined with a 2.5g lateral load to the wheel. Parts should not remain permanently deformed under this loading, so the material yield strength has been used as the upper limit for stress values.

Maximum stress for the Overturning loadcase was in the lower control arm. Von Mises stress results for the lower control arm are shown in Figure 10 below.
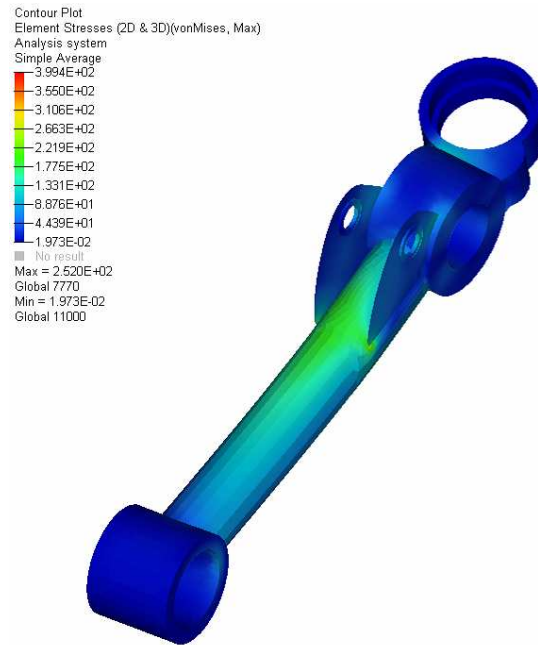

Figure 10: Von Mises Stress – Lower Control Arm, Overturning Loadcase

Figure 10 shows maximum stress in the lower control arm of 252MPa, located on the top side of the tube material. This is below the material's minimum yield strength of 355MPa and is therefore considered acceptable for this loadcase.

Stresses throughout the other components are also below the respective yield strengths and are considered to be at an acceptable level for the applied loads.

6. WHEEL TRAVEL GEOMETRY ASSESSMENT RESULTS

Geometry changes were assessed throughout the suspension system travel. The front shock absorber has a travel range of 64mm (2.5") and the suspension motion ratio is 1.5, so the wheel was moved through a vertical range of ±50mm and changes in toe and camber were assessed.

Overall, the front suspension system showed acceptable levels of geometry change throughout the range of wheel travel. The magnitudes of geometry change were within acceptable ranges, and the rates of change were smooth with no sudden changes in geometry which should result in predictable handling characteristics.

Since vehicle handling characteristics are determined by the interaction between front and rear suspension systems, it should be noted that any reference to handling characteristics are general in nature and only consider the contribution of the suspension system being analysed.

As mentioned previously in Section 3.4, actual bushing stiffness values were not provided, and it was considered beyond the scope of this analysis to conduct physical testing to determine bush stiffness. Instead, assumed typical bush stiffness values were used as a baseline, and the geometry assessment was also run with bush stiffnesses 10x and 0.1x the original values, to determine the effect of bush stiffness on geometry changes. Results are presented in this section for all three variations of bush stiffnesses.

All results presented here are for the left hand wheel.

6.1. Toe Change

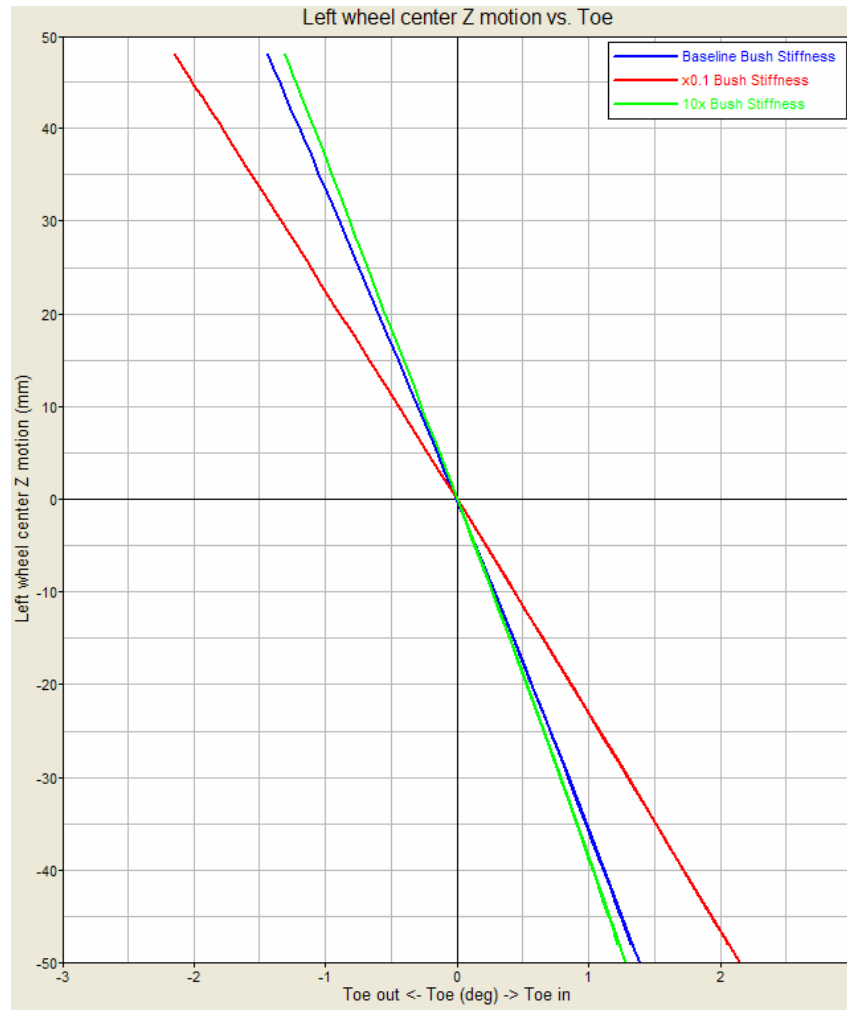A plot of toe change vs. vertical wheel travel is shown in Figure 11 below:

Figure 11: Toe Change v Vertical Wheel Travel

Figure 11 shows the system generates toe out on bump, and toe in on droop. In vehicle roll, this motion will induce a stable understeer characteristic. Maximum toe change for the Baseline configuration is less than 1.5deg, suggesting there is likely to be little roll steer effect. Similarly, bump steer effects should also be minimal.

Figure 11 also shows smooth, linear rate of change in toe throughout the travel with no sudden changes in geometry, which should result in a predictable handling characteristic.

It is also shown in Figure 11 that increasing or decreasing bush stiffness by an order of magnitude still results in acceptable toe change throughout the range of wheel travel, with maximum toe change for the 0.1x Bush Stiffness configuration less than 2.5deg.

6.2. Camber Change

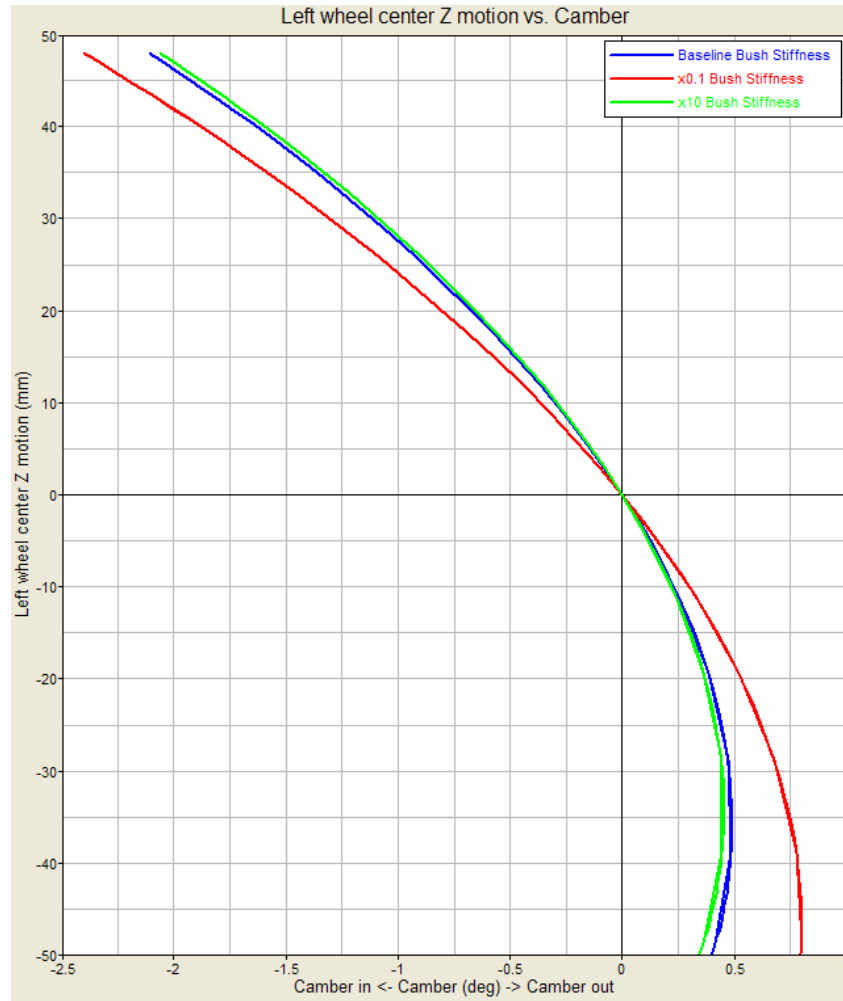A plot of camber change vs. vertical wheel travel is shown in Figure 12 below:

Figure 12: Camber Change v Vertical Wheel Travel

Figure 12 shows the system generates negative camber on bump, and positive camber on droop. In vehicle roll, this motion will keep the outside wheel at a slightly negative camber which will maximise cornering adhesion and minimise tyre wear. Maximum camber change at full bump is 2.1deg for the Baseline configuration and is considered acceptable for this application.

Figure 12 shows a smooth, near linear change in camber throughout the travel with no sudden changes in geometry, which should result in a predictable handling characteristic.

Again, increasing or decreasing bush stiffness by an order of magnitude still results in acceptable camber change throughout the range of wheel travel, with maximum camber change for the 0.1x Bush Stiffness configuration less than 2.5deg.

6.3. Track Change

A plot of track change vs. vertical wheel travel is shown in Figure 13 below:
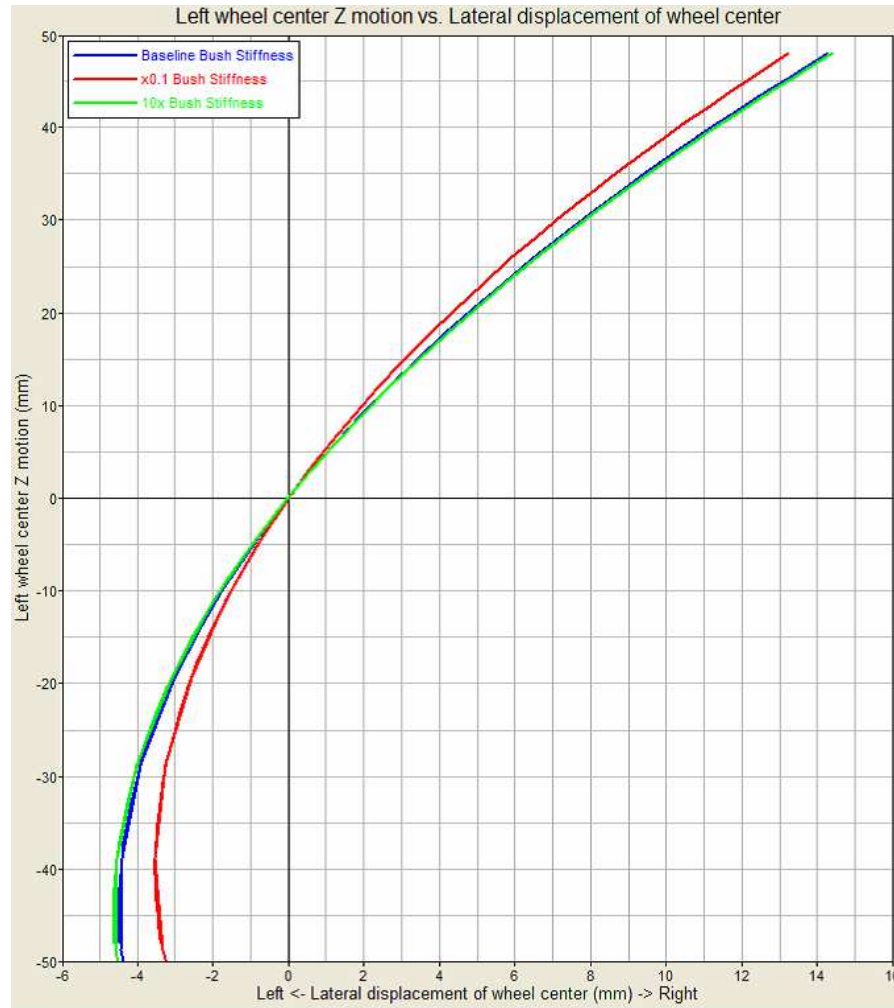

Figure 13: Track Change v Vertical Wheel Travel

Figure 13 shows the system generates a maximum total change in track of 18.5mm which, over a total track of 1400mm, represents around 1.3% change in track and is considered acceptable for this application.

Again, Figure 13 shows smooth changes in track throughout the travel with no sudden changes in geometry, which should result in a predictable handling characteristic.

6.4. Roll Centre Height

Static roll centre height is 44mm above ground level in the configuration that was analysed. The VSB14 requirements specify that the front roll centre height is lower than the rear roll centre height. This is obviously dependant on the rear suspension configuration of the vehicle and must be taken into consideration when installing and setting up both suspension systems.

6.5. Bump Views

Figure 14 shows orthographic views of the suspension system at full bump with the static ride height position is shown as grey wireframe.
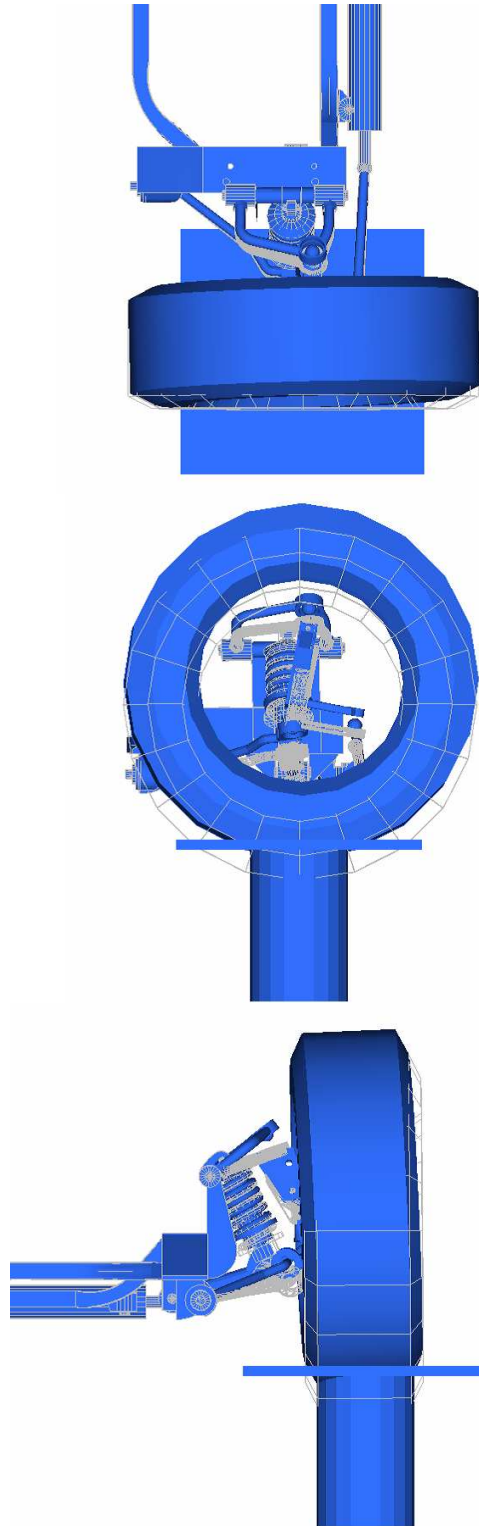

Figure 14: Orthographic Views at Full Bump

6.6. Droop Views

Figure 15 shows orthographic views of the suspension system at full droop with the static ride height position is shown as grey wireframe.
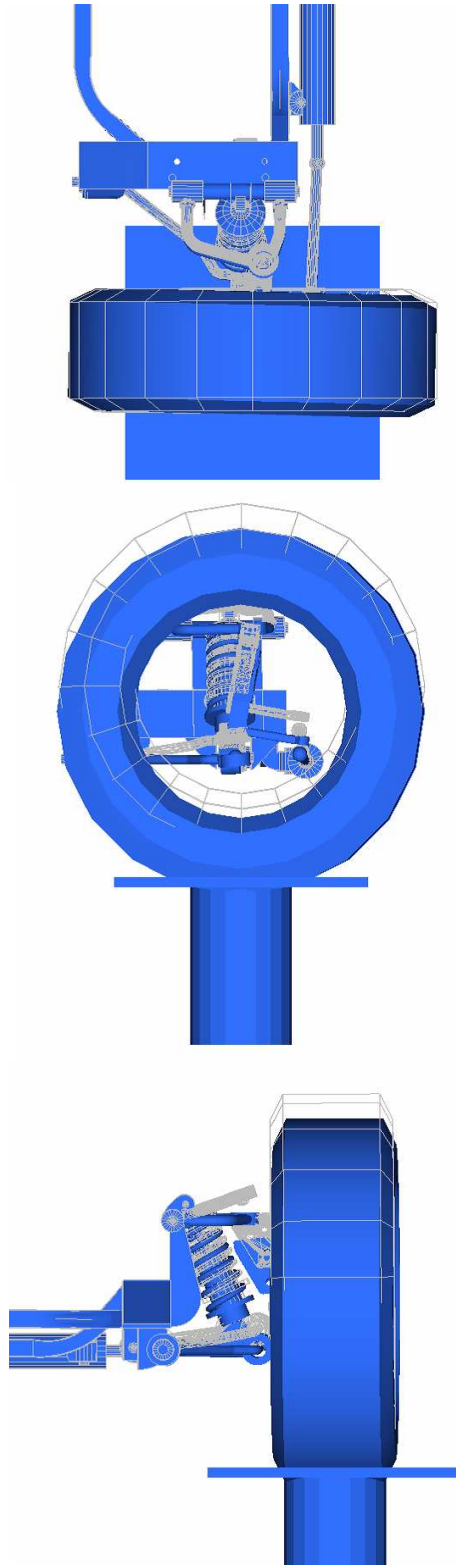
Figure 15: Orthographic Views at Full Droop

7. STEERING GEOMETRY ASSESSMENT RESULTS

7.1. Wheel Angle vs. Steering Wheel Angle

An assessment of geometry changes throughout the range of steering wheel motion was performed. The steering rack has 2½ turns lock to lock, with 130mm of total rack travel.

Overall, the steering system showed acceptable changes in geometry throughout the range of steering lock. The magnitudes of geometry change were within acceptable ranges, and the rates of change were smooth with no sudden changes in geometry which should result in predictable handling characteristics. All results presented here are for the left hand wheel.

Figure 16 shows the change in wheel angle (denoted as toe) with steering wheel angle.
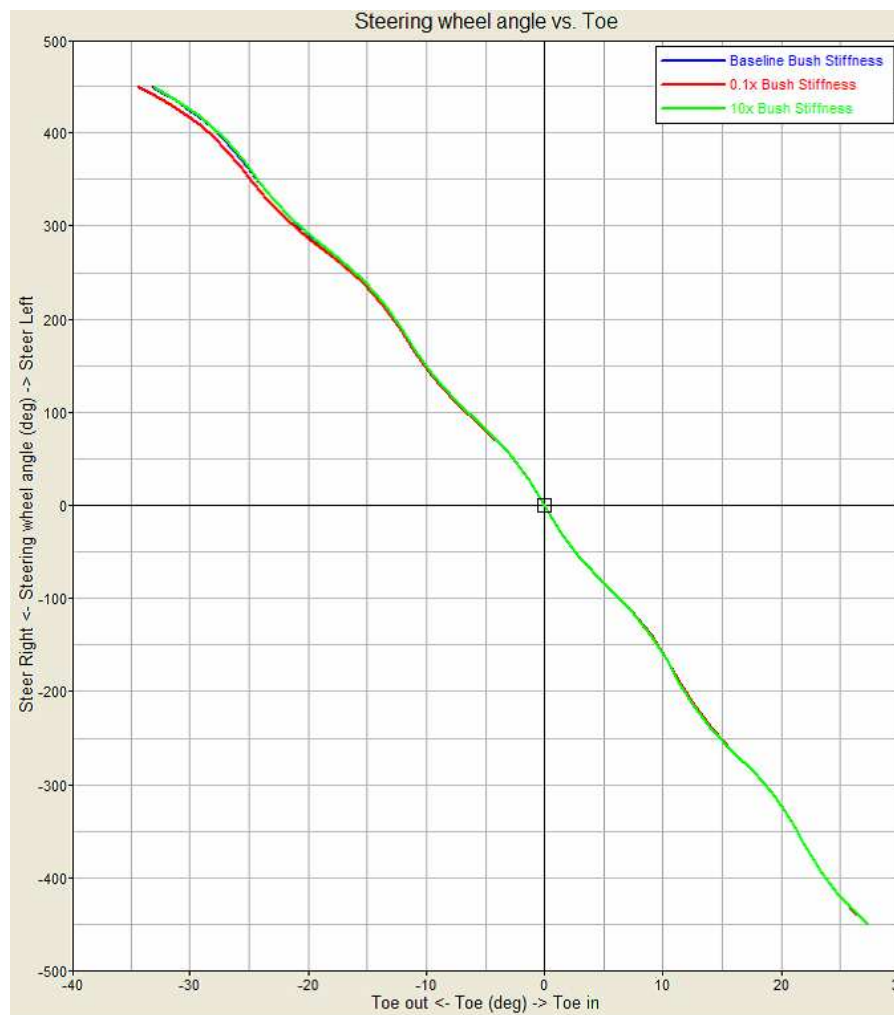

Figure 16: Wheel Angle (toe) vs. Steering Wheel Angle

Figure 16 shows a near linear relationship between steering wheel angle and wheel angle for the range of travel, which is a desired characteristic for the steering system. This will provide a linear steering feel and should result in predictable handling characteristics.

Figure 16 also shows little change between the various bush stiffnesses.

7.2. Camber vs. Steering Wheel Angle

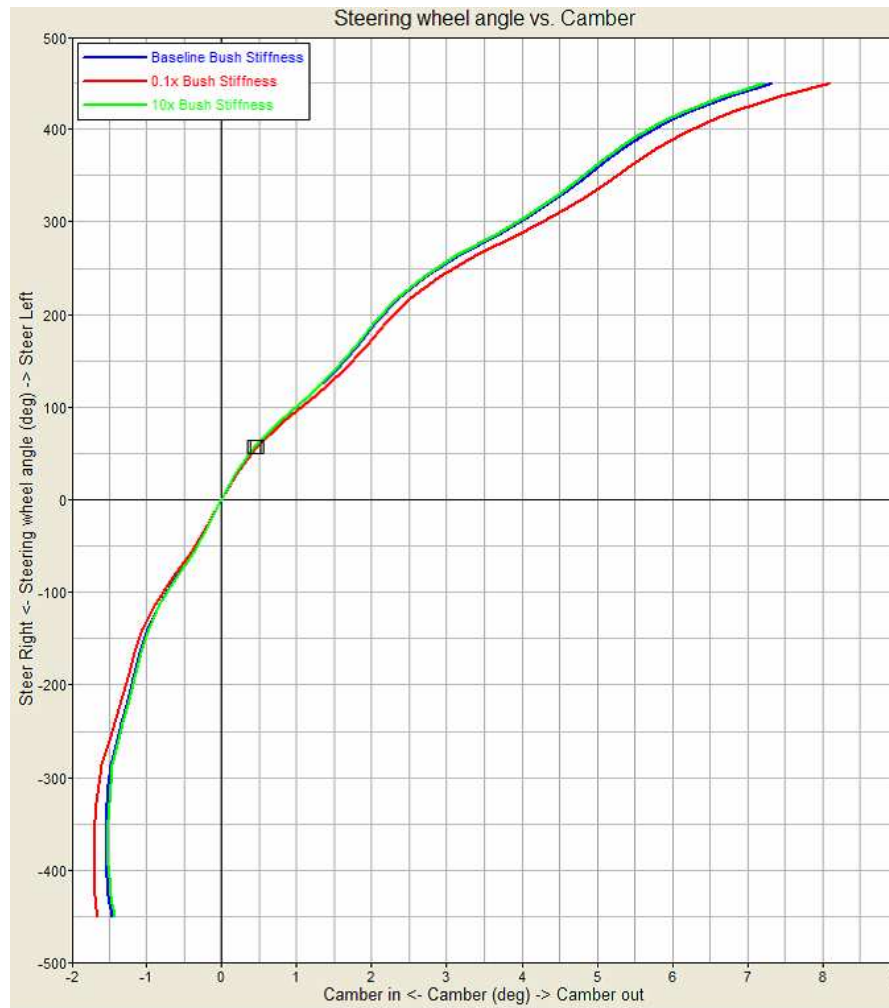Figure 17 shows the change in camber with steering wheel angle.

Figure 17: Camber vs. Steering Wheel Angle

Figure 17 shows that when turning to the right, the left hand (outside) wheel generates negative camber change, which is a desired characteristic. Similarly, when turning left, the left hand (inside) wheel generates positive camber change.

High speed driving conditions are unlikely to see steering wheel angles greater than ±180deg. Figure 17 shows that camber change within the ±180deg range remains less than 2deg which is considered acceptable.

Again, all curves are smooth with no sudden changes in geometry, which should result in a predictable handling characteristic.

7.3. Potential Lockup Conditions

Identification of potential lockup conditions was also performed as part of the steering geometry assessment. The main area of concern is the potential for the outer tie rod joint to overcentre on full steering lock.

The steering system shows no potential for such overcentering, as shown Figure 18. It can be seen that the angle between the steering arm and tie rod is 142deg, which means the joint would have to move through another 38deg of travel before reaching a potential lockup condition at 180deg. This is considered sufficient to prevent overcentering of the linkages.
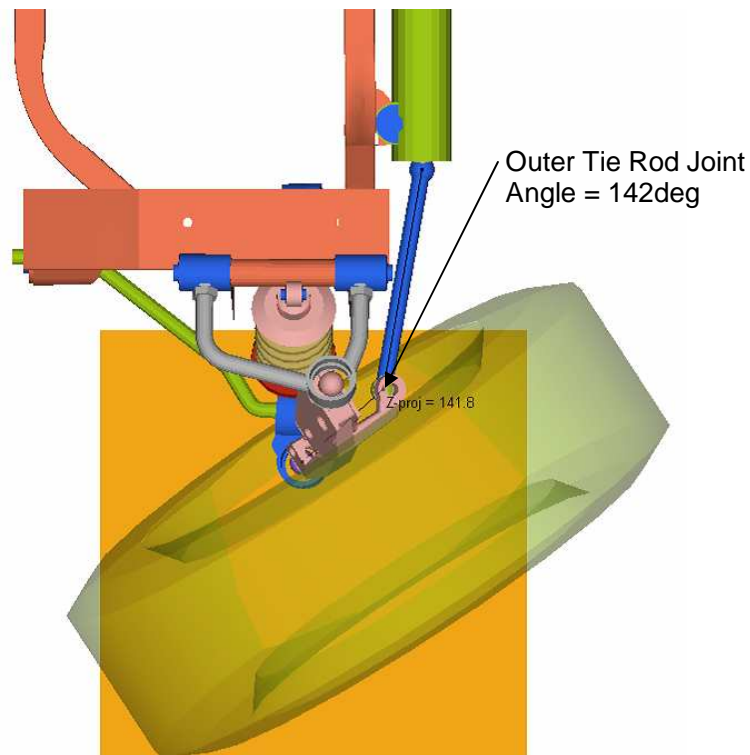

Figure 18: No Potential Lockup Condition at Full Steering Lock

7.4. Steering Lock Views

Figure 19 shows orthographic views of the steering system at full steering lock. The centred steering position is shown as grey wireframe.
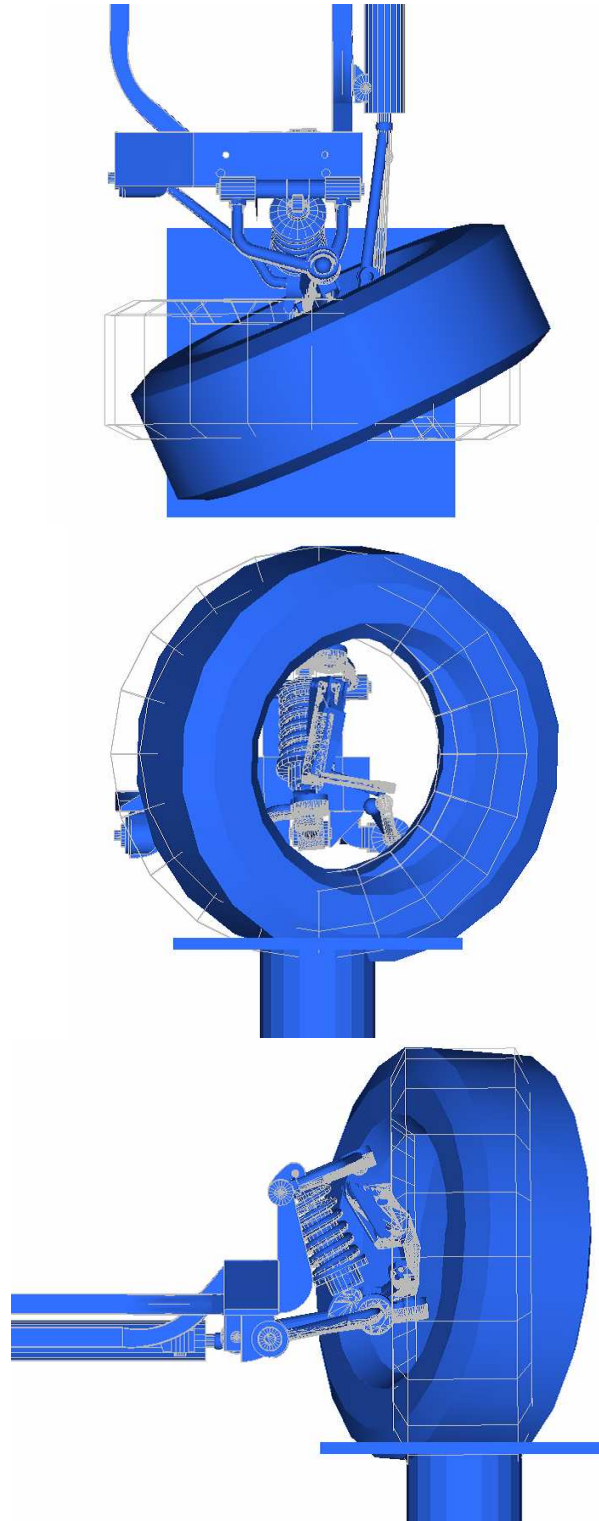

Figure 19: Orthographic Views At Full Steering Lock

8. CONCLUSION

This study has assessed and presented the strength and geometry changes of the front suspension and steering systems subject to the guidelines provided in VSB14.

Multi Body Dynamics (MBD) has been used to simulate the suspension system and calculate stresses in all suspension arms when subjected to bump, braking and overturning loads.

The analysis results indicate that the front end system meets the load requirements set out in VSB14 for a maximum front axle weight rating of 800kg.

The geometry changes throughout the full range of suspension travel were also assessed and were determined to be satisfactory for this application. These changes in geometry should provide the vehicle with a neutral handling characteristics and a tendency towards understeer. Changes in toe and camber are in the desired directions, within acceptable ranges and maintain a smooth rate of change throughout the range of travel.

Geometry changes throughout the range of steering travel were also assessed and were again determined to be satisfactory for the application.

No potential lockup conditions were identified.

9. ABOUT THE AUTHOR

9.1. Formal Qualifications

- Bachelor of Engineering – Mechanical (Hons1)
- Advanced Diploma – Automotive Design & Development

9.2. Experience

Brett Longhurst has worked in the automotive industry for over 10 years, specialising in the field of Computer Aided Engineering (CAE). After gaining his degree in mechanical engineering, he worked for GM Holden for over seven years. During this time he worked in the company's powertrain division performing a range of CAE analyses including stress, vibration and dynamic load analysis.

Since leaving Holden to pursue his own business interests in Bremar Automotion, Brett has performed a wide range of design and analysis projects across a number of industries. He has provided engineering development and application support for Altair Engineering, who are a leading global supplier of CAE software. He has also had experience with a variety of kit, ICV and race car projects and is passionate about making the benefits of CAE accessible to smaller clients without in-house engineering or simulation capabilities.