The High Performance, High Payload Driving School Car
Based on a 1995 BMW 318is

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Project Basis and Goal

The goal behind this particular project was to improve upon the vehicle dynamics of a 1995 BMW 318is. In terms of vehicle dynamics, the BMW 3 Series is known to be a relatively good handling vehicle, but the team intended to transform the car from a performance street car to a higher performance part-time track car capable of maintaining high performance when laden. This means the car should be able to perform well when laden with passengers (such as at a driving school), fellow enthusiasts (such as on the Nurburgring), or similar.

As the intent of the project was to base design decisions on a production vehicle, certain key parameters were left unchanged. The BMW 318is is a rear wheel drive two door coupe weighing 1412 kg when laden with the driver and half a tank of fuel. The vehicle was placed on scales which indicated a 48/52 front/rear weight distribution due to a small and light engine with a significant portion sitting behind the front axle. A dynamic index of unity was assumed, and the rule of thumb that the center of gravity height is 40 percent of the roof height was used, giving a nominal height of 52 cm. A stock wheelbase of 2.690 m and front and rear track widths of 1.430 m and 1.423, respectively were kept consistent through the various designs.

The stock suspension is a MacPherson Strut at the front with a one-piece L-shaped lower control arm plus an anti-roll bar, while the rear suspension is a three link suspension that behaves similarly to a double wishbone setup in the end view. The rear suspension also uses an anti-roll bar. The stock front and rear roll center heights were measured to be about 5 cm and 27 cm off the ground, respectively. The rear roll center seems rather high, probably to reduce the roll moment and to bias the weight transfer distribution to the rear slightly. The best estimates of spring rates and installation ratios from various sources and from measurements, as well as anti-roll bar geometry, are given in the parameters table.

The steering system is a front steer rack and pinion system with an approximate steering ratio of 16.8:1.

Although the team attempted to make an accurate model of the stock BMW 318is, two significant inaccuracies included the front MacPherson Strut that could not be modeled accurately using the double wishbone geometry simulator and unknown tire properties. Thus, the same tire parameters were used throughout the various cases (eta of 0.05 for the front and -0.20 for the rear). The rationale behind these values are discussed with the steady state handling analysis.

The team analyzed the stock suspension in the one-driver 1412 kg laden mass as well as adding 300 kg to simulate the vehicle loaded near its loading capacity. With 225 kg of this load over the rear axle, this changes the front/rear weight distribution to 44/56. Because the seat height and trunk floor height are about 3 cm below the estimated center of gravity height, the team assumed that loading the car did not change the center of gravity height significantly. This assumption was based on the fact that although the vehicle static height lowers slightly with added mass, some of the load may be higher up than the seat height (items stacked in the trunk and on the roof, for example).

For the team's redesign of the BMW 318is, one of the first steps was to increase the spring rates to raise the sprung natural frequencies from the stock values of 0.9 Hz front and 1.0 Hz rear to 1.3 Hz front and 1.5 Hz rear. This can reduce the roll rate (assuming similar suspension geometry), thereby decreasing the overall weight transfer. A reduced roll also decreases the time lost in fast transitions such as in slaloms. The spring rates are given in the parameters table and were chosen to satisfy the Olley criteria, with some exceptions detailed in the discussion on steady state handling and on ride quality. A mid-high damping ratio of 0.3 was chosen to provide relatively quick load transfer in the transients without compromising too much of the ride quality (more discussion later). The team specified the natural frequencies, determined the ride and roll rates and, using the position of the spring and shocks (assuming collinear mounting), determined the installation ratios to specify actual physical spring and shock rates. Anti-roll bar stiffness selection is discussed in the steady state handling section; values are provided in the parameters table.

In terms of suspension kinematics, a double wishbone design was used for both ends of the car. Discussion of the design is in the suspension geometry section. Some data on the geometry and steering system are also in the parameters table.

Suspension Geometry

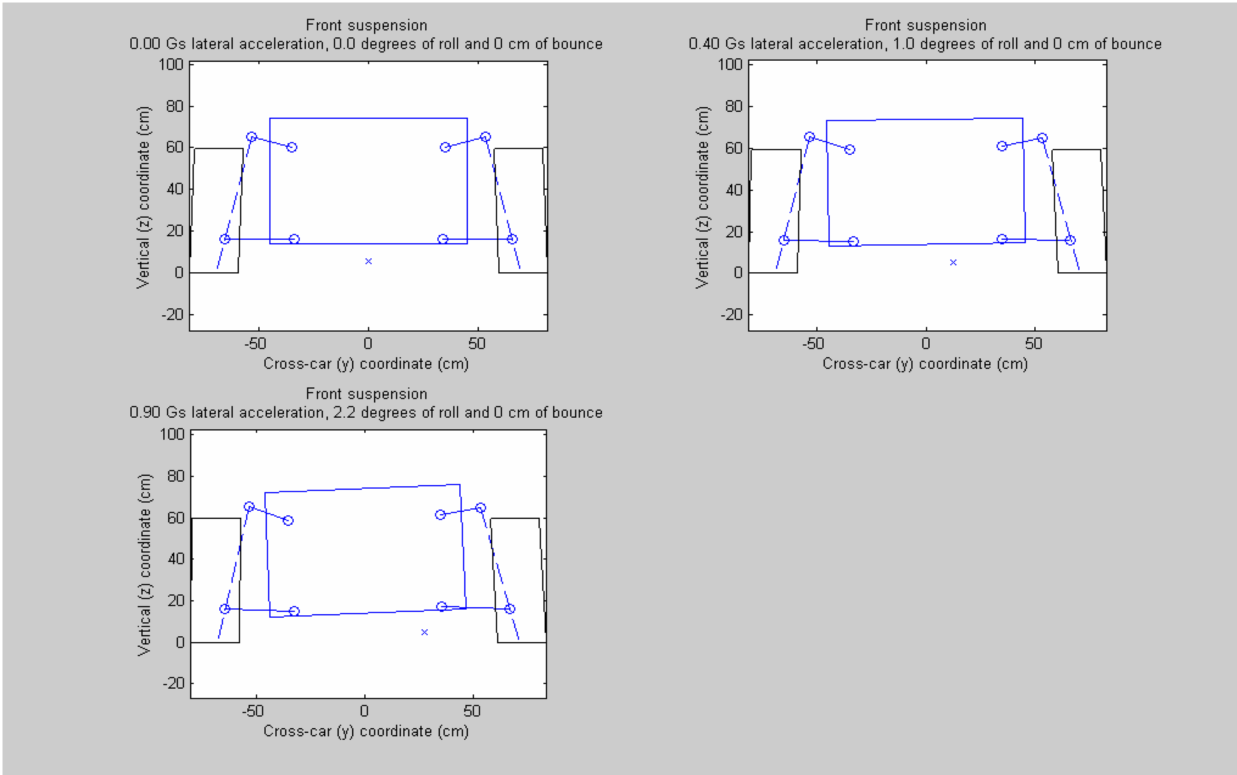
Front Suspension

The front suspension on the stock 318is is a MacPherson strut design. Because the suspension geometry program could only model double wishbone suspensions, the team attempted to simulate a MacPherson strut with the double wishbone program. The suspension links were modeled so that the lower control arm was in the correct position. The top wishbone was placed to try to match the roll center and instantaneous center of the stock 318is' MacPherson suspension in a static case.

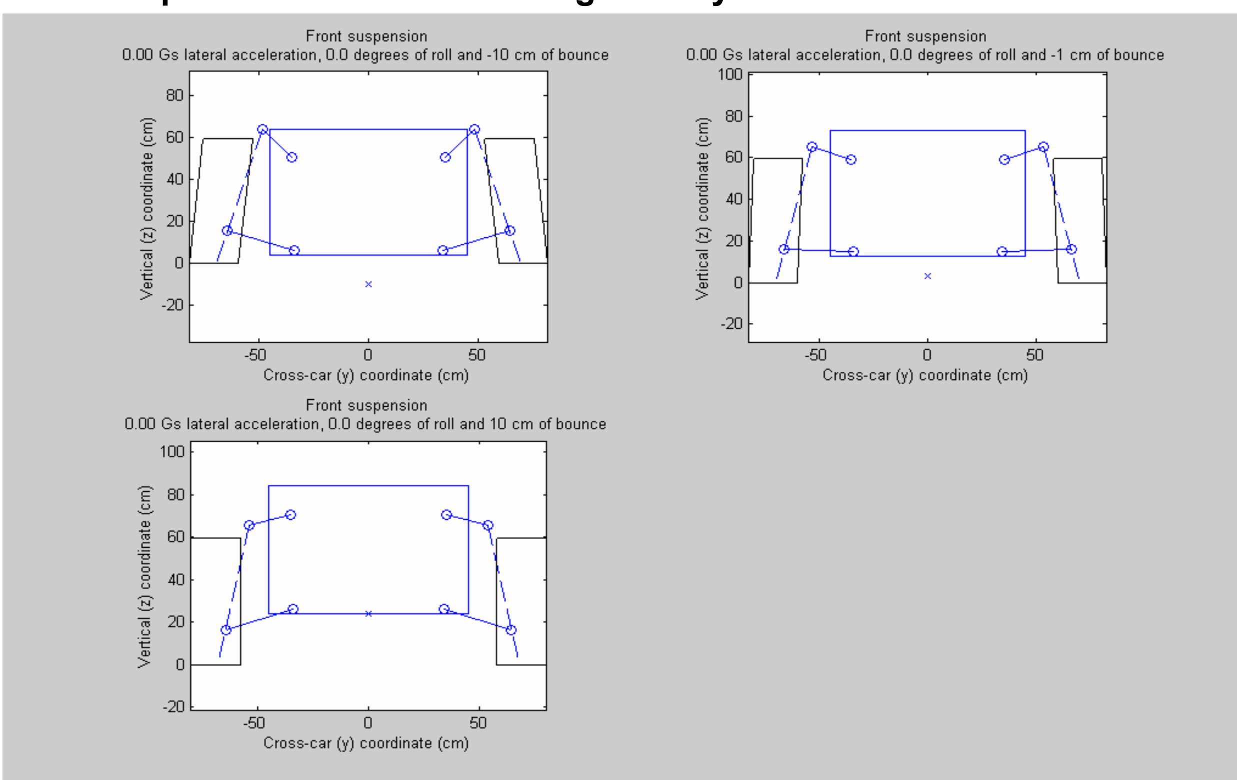
For the team's design, the team went about trying to improve the stock suspension using a true double wishbone setup as it allowed for more flexibility in selecting parameters. The link lengths and locations were changed, as the team set the front geometry to place the roll center near the ground. This was done in order to reduce scrub and to minimize jacking effects. The team also had the links travel so that the camber change during roll kept the wheel with negative camber with respect to the ground. The ability to provide more negative camber was also another reason why the team decided to design a double wishbone suspension rather than a MacPherson strut. Although more negative camber would allow higher lateral force capability and slightly increased cornering stiffness, tire wear would be more of a problem. This was decided to not be an issue, because the intent of the vehicle was to corner better, not to save make optimal use of tire life.

After modifying the suspension geometry several times, a geometry that met the desired properties was found. Below are the figures for the front suspension in roll and bounce. The static front roll center height is 5.4 cm. Both values are low to reduce scrub and the jacking effect. The change in roll center height between a vertical motion of +/- 10 cm from static height was found to be 34 cm. This is perhaps a bit large but as long as suspension travel is limited with stiff springs and dampers, this is not necessarily a problem. Furthermore, the bump and droop stops may disallow this much motion. For roll, where roll center movement is more important, the change in roll center height was only 0.6 cm over the range of about 0º to 2º of roll; the car does not roll much more than that as discussed later. The team felt that plots of the suspension geometry were more revealing than lines of roll center height change, so these are given below.

Front suspension geometry for modified car in roll

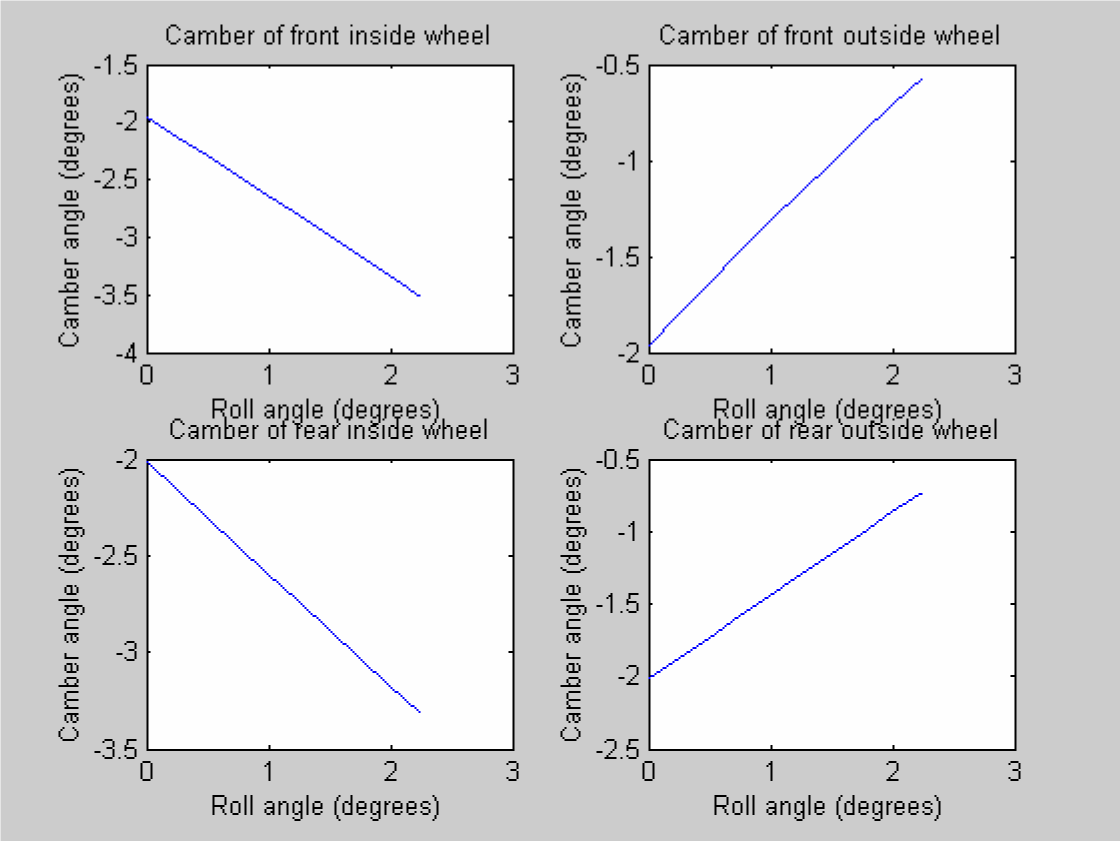


Front suspension for modified car geometry in bounce



The plot below shows that, for the suspension geometry selected, the tire will always have negative camber for the roll angles that it normally experiences. This is quite important for a performance car since tire deflection and compliance effects may worsen the camber.

Camber change over roll angles for steady state cornering for front and rear of modified car

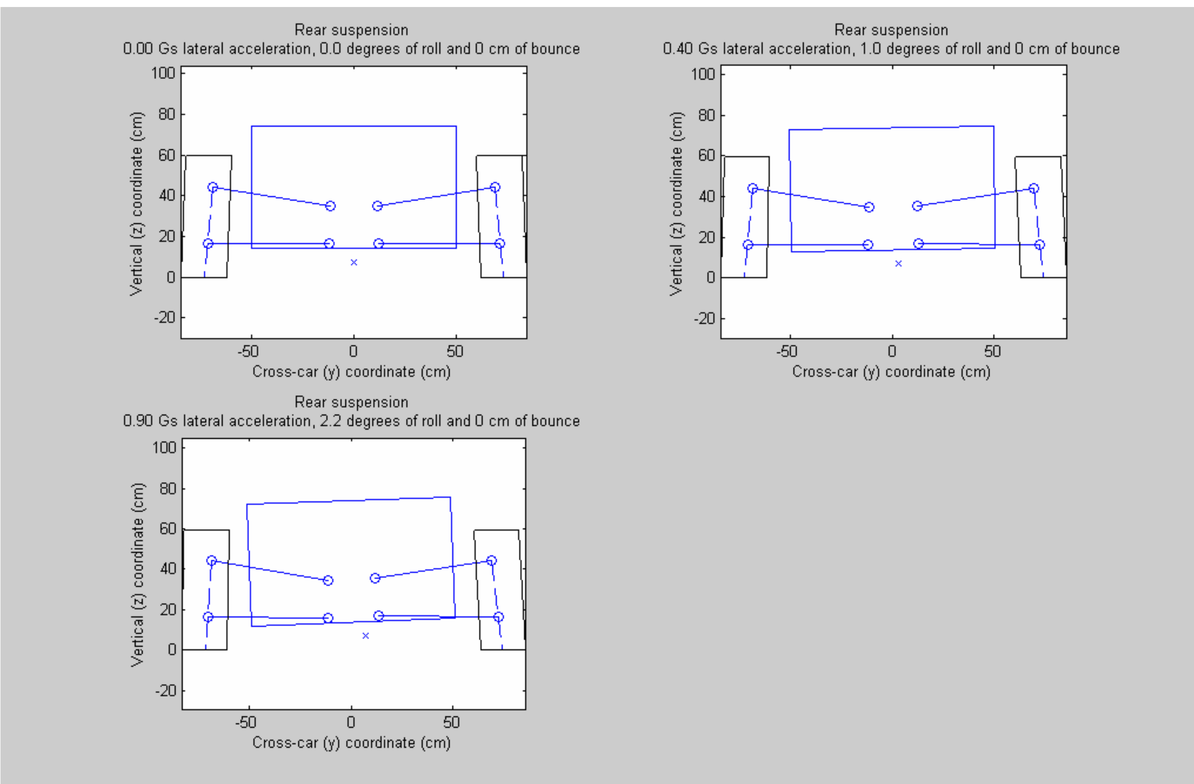


For the scale drawings of the front suspension geometry, see ensuing pages.

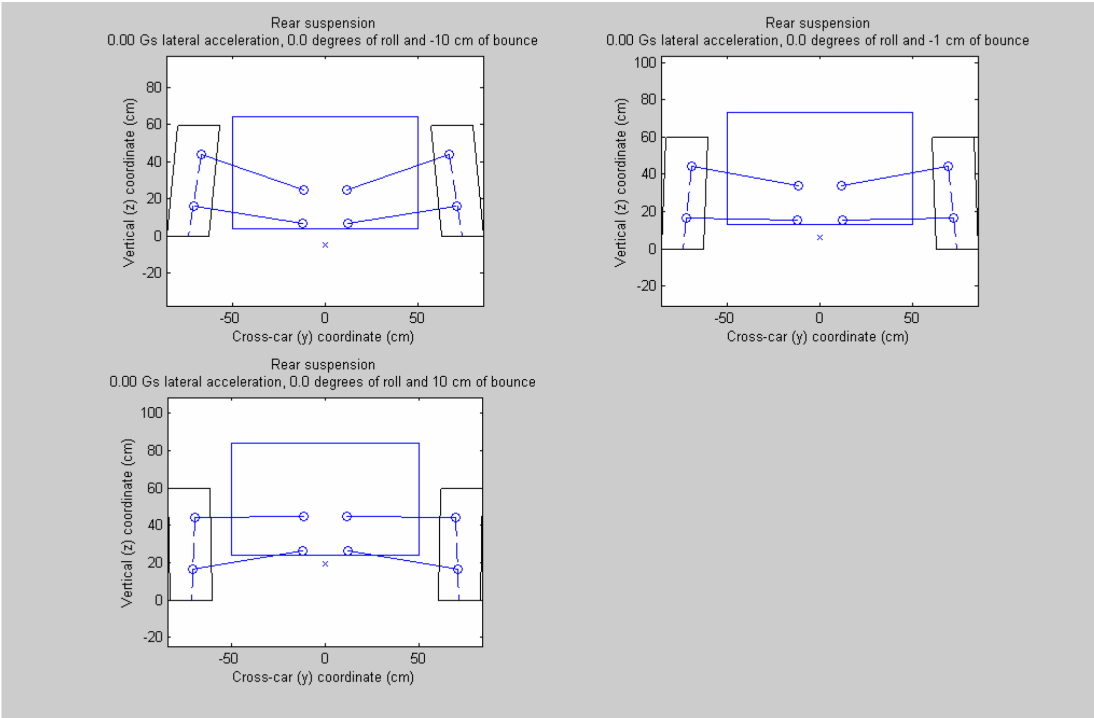
Rear Suspension

The stock 318 has a three link rear suspension with end view geometry virtually equivalent to a double wishbone rear suspension. After modeling it in the double wishbone program, it was noticed that the setup had a roll center that was relatively high (27 cm). In order to decrease the roll center height the team manipulated the links until the roll center height was decreased. As described above, the decreased roll center height allows for less scrub and minimizes the jacking effect. The suspension was then also modified to decreased the amount of camber during roll as the team found that the stock suspension offered too much negative camber in roll and bounce, causing significant negative camber under loading. In the theme of designing a car that can carry a relatively high load and maintain handling capacity, the team reduced the camber gain. The static rear roll center height is 7.4 cm, which is slightly higher than the front roll center—a rule of thumb that gives the driver a good feel of the vehicle. Pictured below are the plots of the rear suspension in roll and bounce.

Rear suspension of modified car in roll

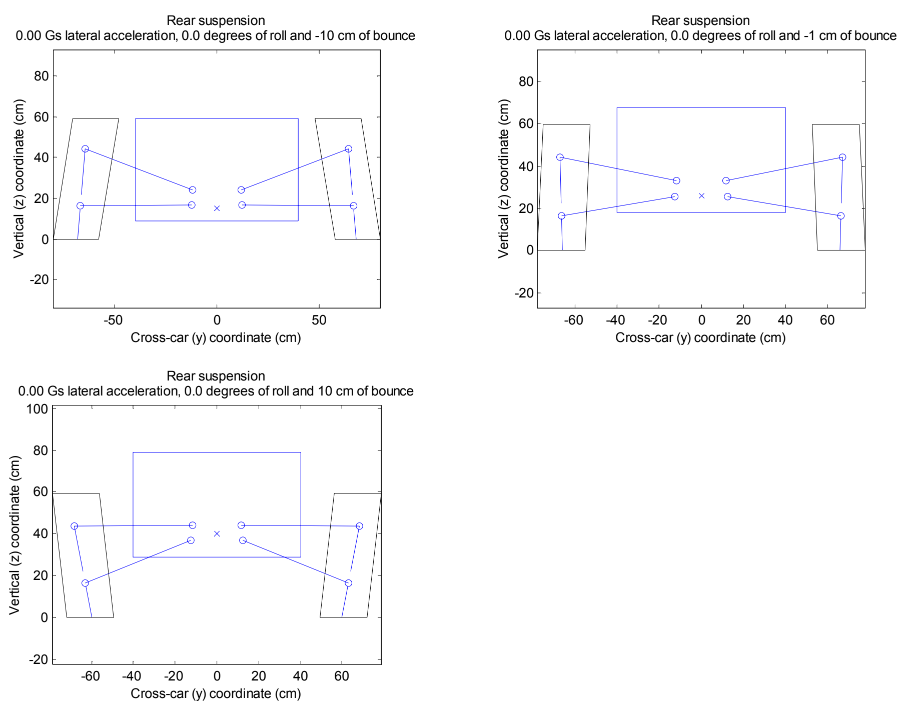


Rear suspension of modified car in bounce



For comparison, the stock rear suspension's large geometry changes in bounce are shown below.

Rear suspension of stock car in bounce



The change in roll center height between a vertical motion of +/- 10 cm was found to be 24 cm. Between 0º and 2º of roll, the change in roll center height was only 0.05 cm, so the rear suspension kinematics are quite stable with roll. The above camber change plot with roll angle shows the camber of the outer wheel will also always be negative, up to a roll of about 2º, allowing the wheel to generate more lateral force.

Steering Geometry

Please see attached figures of front suspension and steering geometry, rear suspension, and front side suspension geometry. These figures show the location of major steering system components, including the caster angle (5º), kingpin inclination angle (14.2º), scrub radius (1.1 cm), and mechanical trail (2.7 cm). The stock suspension's steering ratio of 16.8 was retained.

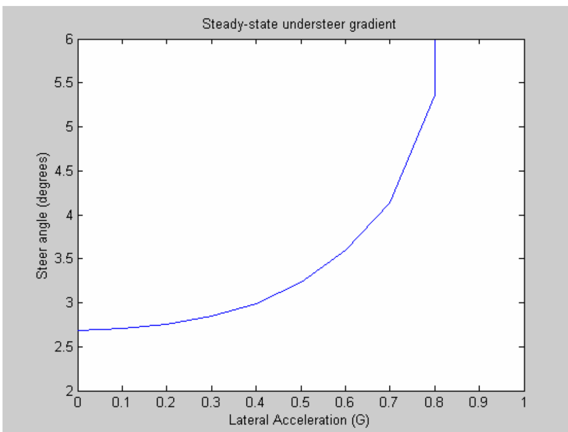
A caster angle of 5º was chosen to provide adequate self-centering of the steering wheel and some diagonal weight transfer under large steering angles. A moderate kingpin inclination of 14.2º was the result after setting the control arms to produce reasonable roll and instant centers as discussed previously. An attempt was made to prevent the kingpin inclination becoming too large which would produce undesirable positive camber on the front wheels. A small positive scrub radius was chosen to provide good feel without transmitting too much of the road imperfections. A small mechanical trail was chosen to provide sufficient sign of front tire breakaway (limit understeer).

Steady State Handling

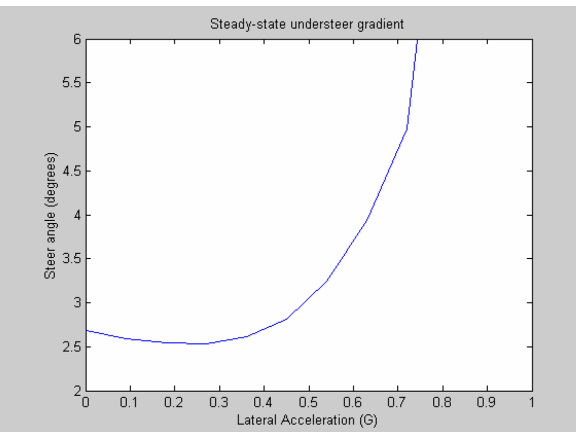
Understeer Gradient

The team's goal in designing the suspension was to create a very slightly understeering car that limit understeered. It was also important that at the limits, this understeer was substantial. This is because the car is rear wheel drive and thus throttle induced oversteer is a concern. Below are the understeer gradient plots for both the stock and modified versions and the load and unloaded cases of each:

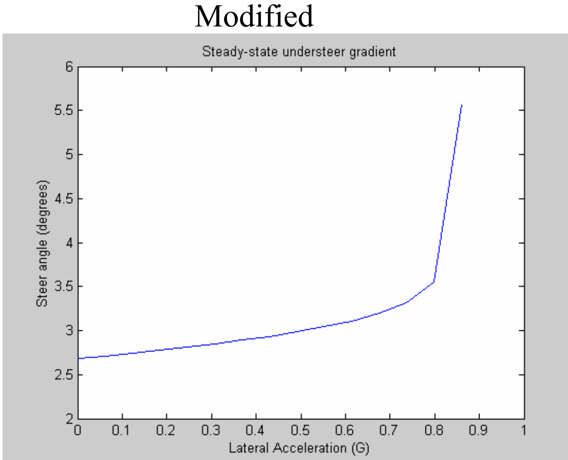
Stock



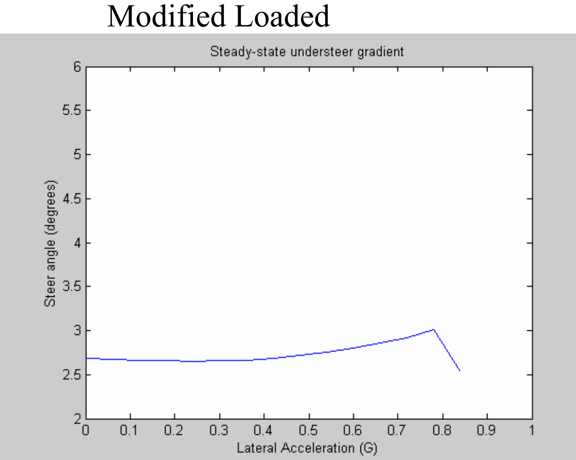
Stock Loaded



Modified



Modified Loaded



The above plots show a stock vehicle that is limit understeer regardless loading, although when fully loaded the 318is does initially oversteer due to the change in weight distribution. The modified car in both cases is very close to neutral steering at low and mid-lateral accelerations. The team was able to flatten the initial curve and extend the steeply rising part of the curve out to 0.8 G's from the 0.5 to 0.6 G range. While the unloaded case did have a sharp transition to limit understeer as the team had hoped, the loaded case made a change from mildly understeering to limit oversteer at around 0.78 g's. This effect is undesirable for a street car as it could lead to loss of control at the limit, but the designed vehicle is tailored for high performance use with a driver perceptive of the limits. The team found that this configuration was the best compromise as to get rid of the oversteer in the loaded case would result in more understeer for the unloaded case. and several configurations were attempted to resolve the problem but a solution could not be found without sacrificing the mild understeer gradient of the unloaded case.

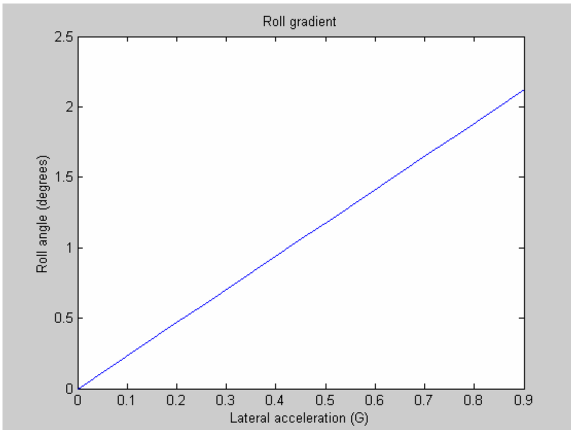
There were three main parameters that helped to shape the above curves. The first was the cornering stiffness of the tires. By using relatively stiff tires, the team was able to affect the initial part of the curve, and set the understeer gradient for low G's. The team found that, to offset the rear biased weight distribution of the car, especially in the laden case, stiffer rear tires were necessary. The high g range was influenced greatly by the remaining two factors, spring stiffness and anti-roll bar stiffness, which contribute through affecting the roll stiffness (in conjunction with the roll moment, which was fixed via the suspension geometry design). In order to minimize roll, relatively high natural frequency springs were chosen, but the relative magnitudes of the front and rear were constrained by Olley criteria. Spring selection is described in more detail in the ride quality section. With those values set, the front and the roll bars were used to adjust the weight transfer to bias the mid- and high G to tune the understeer gradient curve.

As the parameters table shows, the team's tuning allowed the unloaded car to go from a limit lateral acceleration of 0.83 G to 0.86 G and the loaded car from 0.81 G to 0.84 G, mostly via the later onset of limit understeer.

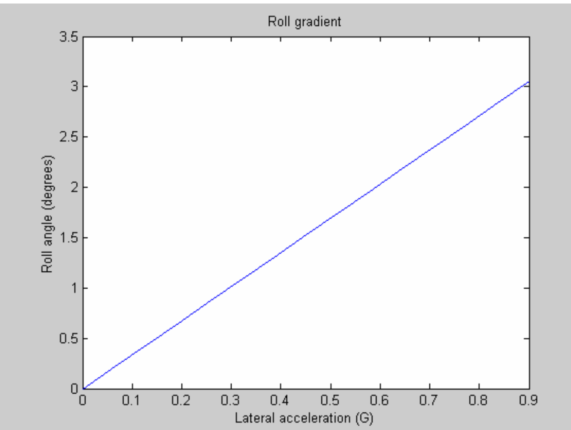
Roll and Rollover

In addition to the understeer gradient, steady state roll rates for the vehicles were examined. Plots comparing this for each of the vehicles can be seen here:

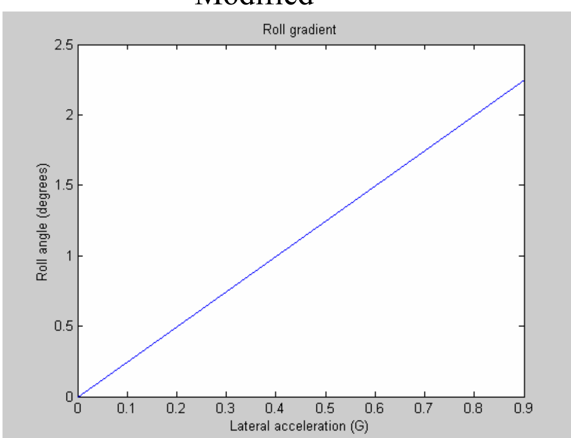
Stock



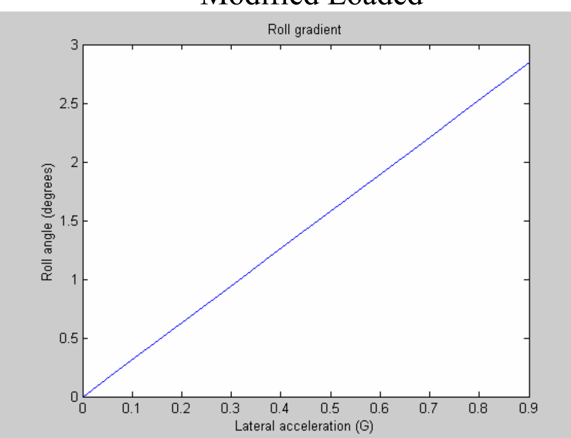
Stock Loaded



Modified



Modified Loaded



The above curves show the four slightly different roll gradients. It can be seen that the roll rate for the unloaded case actually increased slightly from around 2.3 deg/G to about 2.5 deg/G. While this increase is generally undesirable, the initial roll rate was low enough that the increase should have little effect on the car's feel. The loaded case did improve however, decreasing from about 3.3 deg/G to 3 deg/G. Despite having significantly stiffer springs on the modified vehicle, only small changes in roll gradient are seen, including an increase in the unloaded case. This result is due to the significant lowering of the roll centers in the modified vehicle creating larger moment arms between each of the roll centers and the center of gravity. Thus, the team decided that a lower roll center for reduced jacking and the proper camber change was beneficial and lost some of the effect of stiffer springs here.

In all four of the above configurations the car will lose traction first which is desirable for safety. The driver is much less likely to panic, and thus have a chance to recover, from the sensation of slip than the sensation of the tires lifting off the ground. The severity of an accident due to spin out will also likely be much less than that of a rollover. Below is a table of both the acceleration limits and rollover threshold for the four vehicles along with the roll and understeer gradients for each of the vehicles.

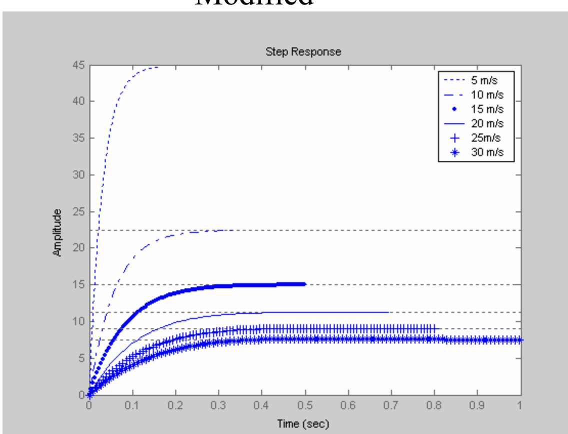
Steady state characteristics	Stock	Stock Loaded	Modified	Modified Loaded
Roll gradient (deg/g)	2.4	3.4	2.5	3.2
Initial Understeer gradient (deg/g)	0.35	-1.0	0.54	-0.1
Rollover threshold (g's)	1.06	0.90	1.27	1.30
Which wheel lifts off	Front	Front	Rear	Front
Limit characteristic	Understeer	Understeer	Understeer	Oversteer
Limit lateral acceleration (g's)	0.83	0.81	0.86	0.84

The above table shows all of the vehicles are generally comparable, with the largest improvements coming in the fully loaded case. The modified, fully loaded vehicle has less roll, is less oversteering, and has a higher lateral acceleration limit than the stock vehicle. Thus we have the high performance, high payload driving school car. The draw back of the loaded modified case is that it is limit oversteer; a problem the team hoped to fix by introducing an active roll bar to the vehicle as described later. Or, without the active control, the car would serve well as a trainer for drivers to control oversteer!

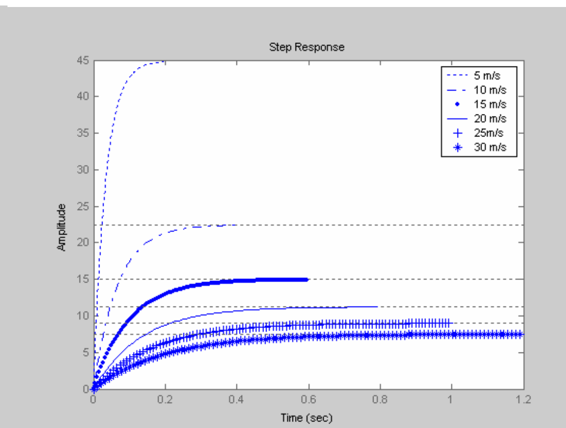
Transient Handling

Variation between stock and modified designs are not as significant when observing transient behavior. This is largely because the behavior in the implemented models is dependent primarily on the cornering stiffnesses and weight distribution. The weight distribution was kept the same between the stock and modified cases, and the cornering stiffnesses only changed marginally due to the different lateral load transfer distributions. The curves of yawrate vs. time all look similar, but the rise times do differ slightly at high speeds. Overshoot is not evident because the cars either have very low understeer gradients that make their response look like the 1st order response of a neutral steering car or actually have initial understeer. The rise times are very fast in general due to the high tire cornering stiffnesses.

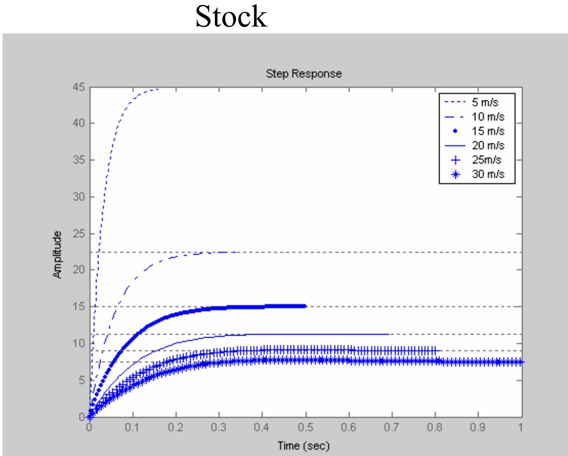
Modified



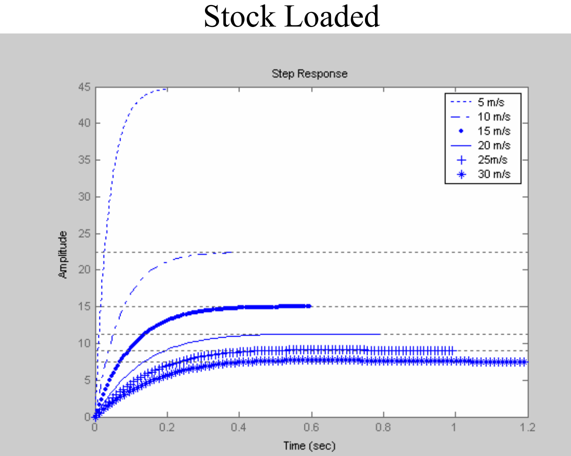
Modified Loaded



Stock

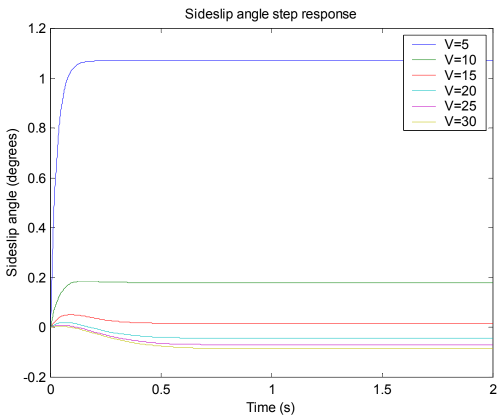


Stock Loaded

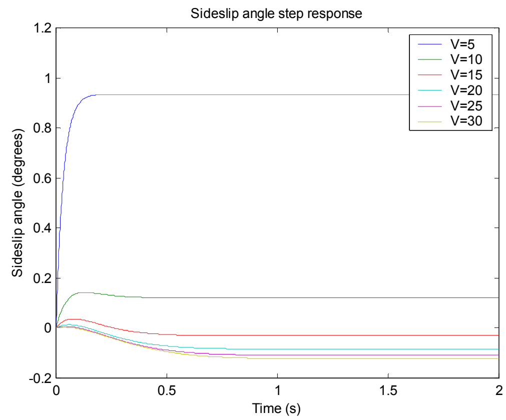


For both the stock car case and the modified car case, the steady state sideslip angle decreases as shown below when fully laden due to the rearward shift in weight; this shows up through the decrease in the "b" length and an increase in the "a" length.

STOCK



STOCK LOADED



Also, as speed increases, the steady state sideslip shifts toward the negative side as tire slip angles build and become close to or larger the Ackerman angle and the turning center moves forward.

Best estimates for the TB values are given in the parameters table. But since there is basically no overshoot, the values are not particularly significant. Times to the 90% steady state values were used instead of the time to the peak overshoot. In general, the TB values are very low due to the stiff tires.

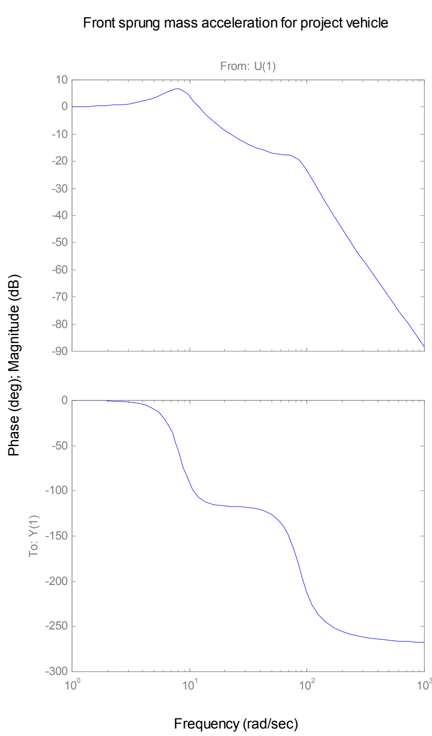
Ride Quality

In this ride quality section, the graphs of the stock car will not be presented as the stock car was not the team's design and because there would be too many plots. The important numbers such as the sprung and unsprung mass natural frequencies are provided in the parameters table.

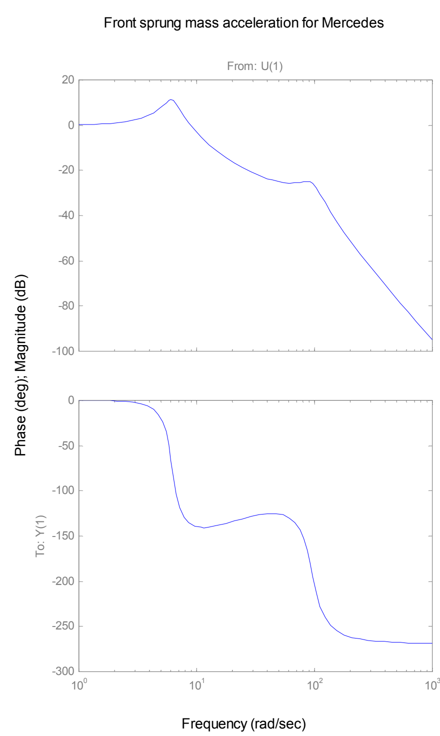
For the modified car, a natural frequency of 1.3 Hz was chosen for the front sprung mass. An Olley criterion is that this front rate is 30% less than the rear ride rate such that the rear suspension's movement does not trail the front's for too long. The team decided to satisfy this criterion as it could tune the handling with anti-roll bars. This resulted in a rear sprung natural frequency of about 1.5 Hz. Both of these frequencies are rather high according to Olley, but being a high performance car with a solid chassis, the team decided that these numbers were acceptable. Dynamic indices of unity were assumed, satisfying the third Olley criterion that the pitch and yaw frequencies were similar. A damping ratio of 0.30 was chosen. The team chose this relatively high damping ratio to quicken the load transfer in transients.

Bode plots of the road acceleration to the front sprung mass acceleration of the modified car compared with the Mercedes is shown below for the standard loading case, followed by a comparison of the rear sprung mass acceleration, both for the regularly loaded case.

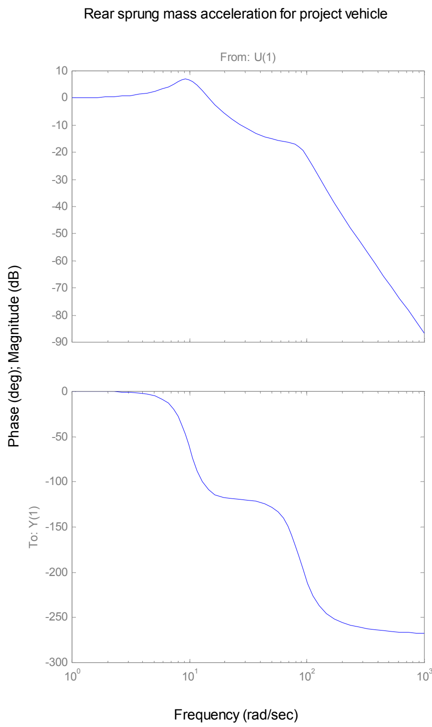
MODIFIED



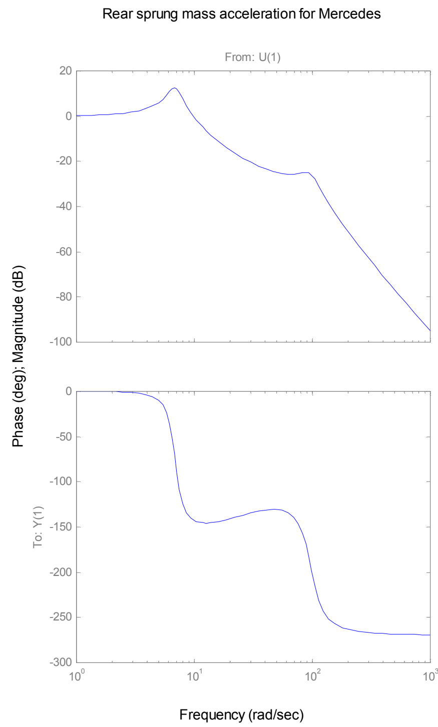
MERCEDES



MODIFIED



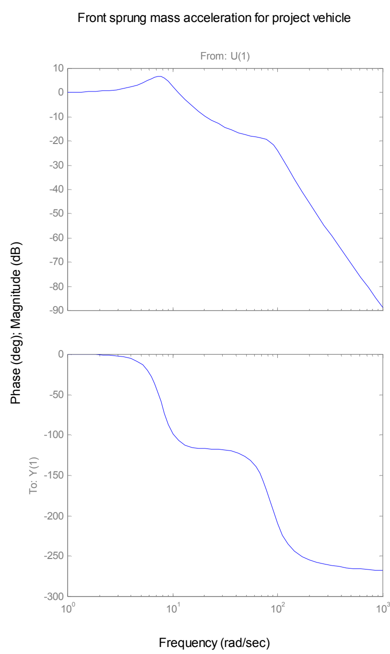
MERCEDES



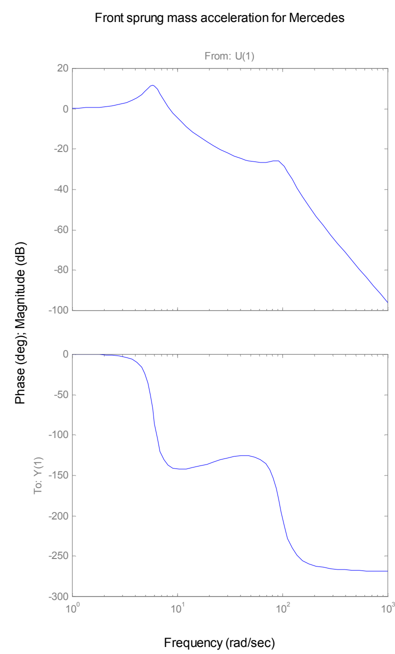
These graphs above highlight a few significant differences. First, the higher natural frequency for the modified car is evident by its peak closer to 1.3 Hz vs. 0.85 Hz of the Mercedes. However, the peak at the sprung natural frequencies, the Mercedes actually shows slightly higher acceleration gains. The team believes this is due to the lower damping of the Mercedes suspension. The parameters given for the Mercedes show a damping coefficient of 1500 N/(m/s), which is a damping ratio of about 0.15 for the Mercedes' effective rate. This is noticeably lower than the team's 0.30, so the peak is more abrupt. However, the higher natural frequency of the team's design means that the gain from the sprung mass frequency to the unsprung mass frequency to which passengers are most sensitive is a little higher. Therefore, although the modified car may show reduced float over low frequency surfaces, it may be perceived as slightly harsher. Finally, the unsprung frequency peaks of the Mercedes occur a little higher (about 15 Hz) due to the higher tire stiffness of 337500 N/m as opposed to the 237500 N/m for the front and the 300000 N/m for the rear of the modified car, which gives an unsprung frequencies of about 14 Hz. These tire spring stiffnesses are a by-product of the tire cornering stiffness choices.

Here, the same front and rear axle sprung mass acceleration Bode plots are given for the loaded case.

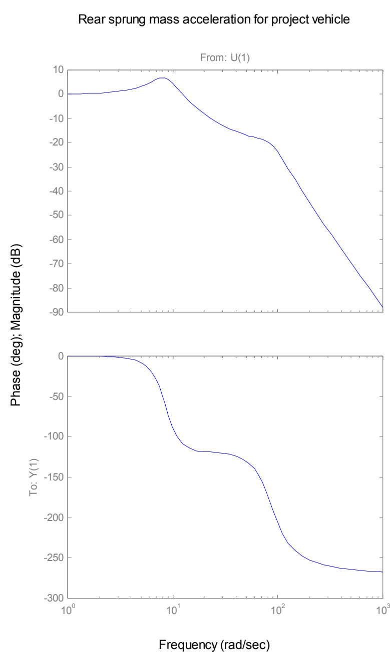
MODIFIED LOADED



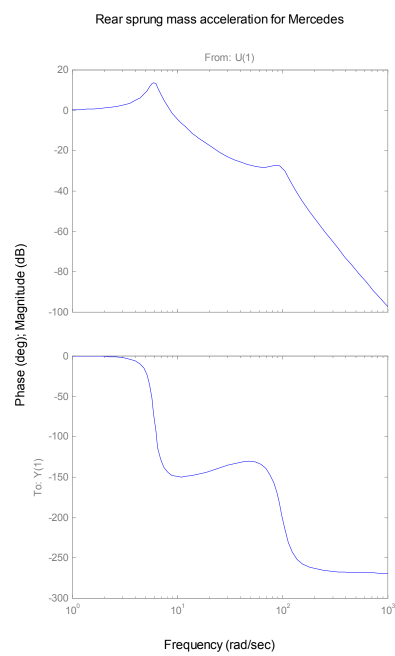
MERCEDES LOADED



MODIFIED LOADED



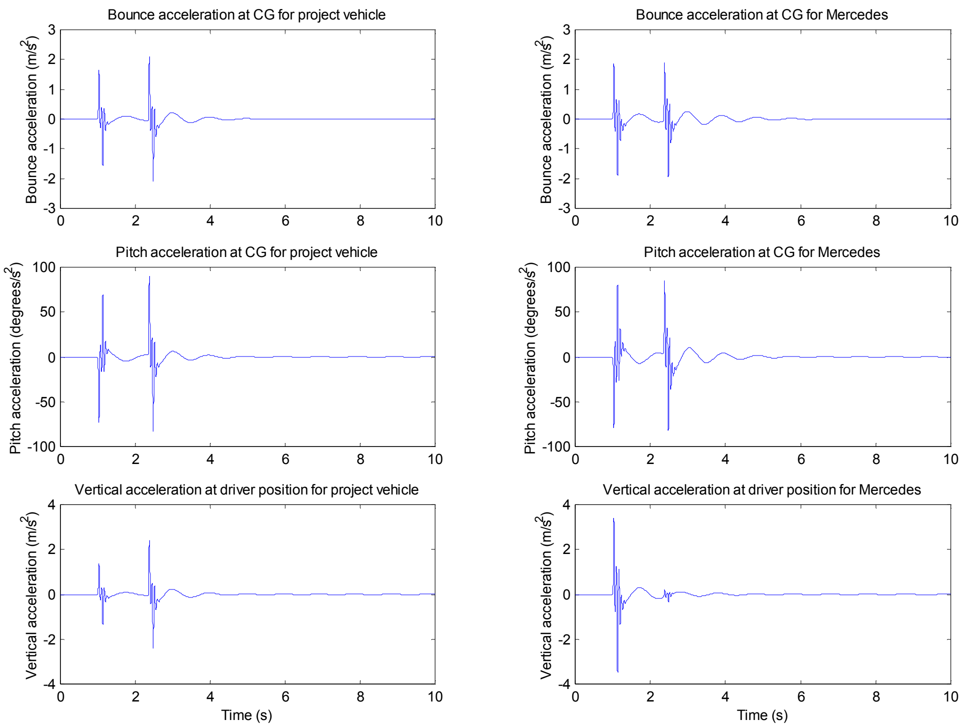
MERCEDES LOADED



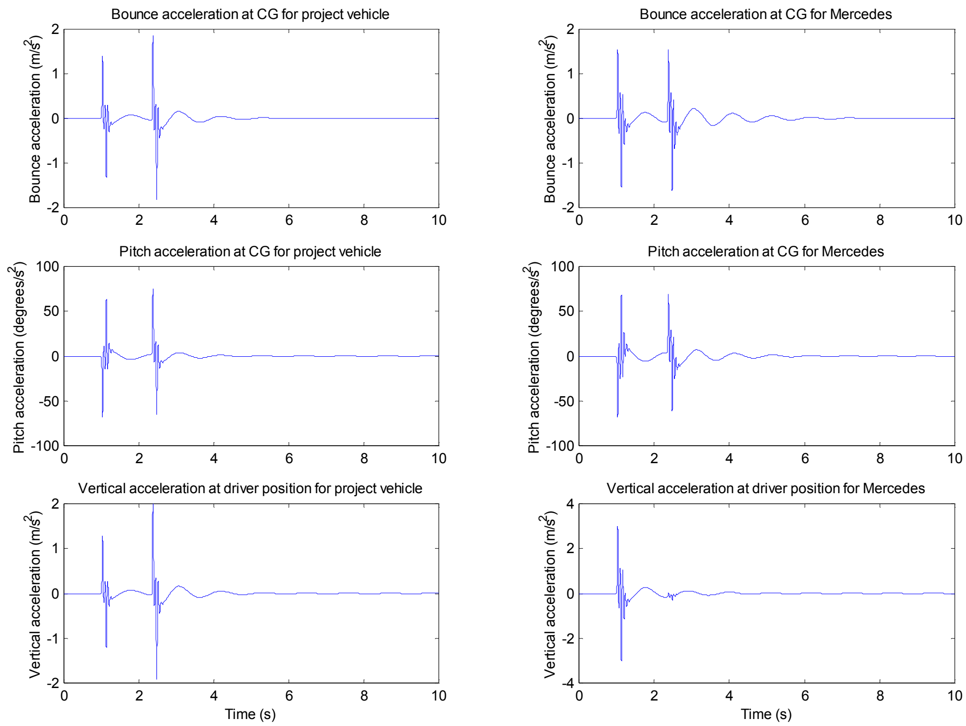
The aforementioned trends are still visible here. The most significant difference is the slight drop in the sprung mass natural frequencies due to the extra 300 kg load. Please see the parameters table for more info.

Here are the time domain plots of the various modes of sprung mass accelerations for the regularly loaded case. Overall amplitudes are similar, and the most visible difference is that the modified car's accelerations decay noticeably more quickly due to the higher damping coefficient. In the modified car, the driver sits slightly aft of the center of gravity (1.6 m behind the front wheel as measured), while the team estimates that he/she sits slightly in front of the center of gravity in the Mercedes. Therefore, in the modified car, the vertical acceleration that the driver feels when the front axle hits the bump is attenuated by the pitch of the vehicle, while the driver in the Mercedes has the vertical acceleration increased by the pitch. The converse is true for the rear axle hitting the bump.

MODIFIED MERCEDES

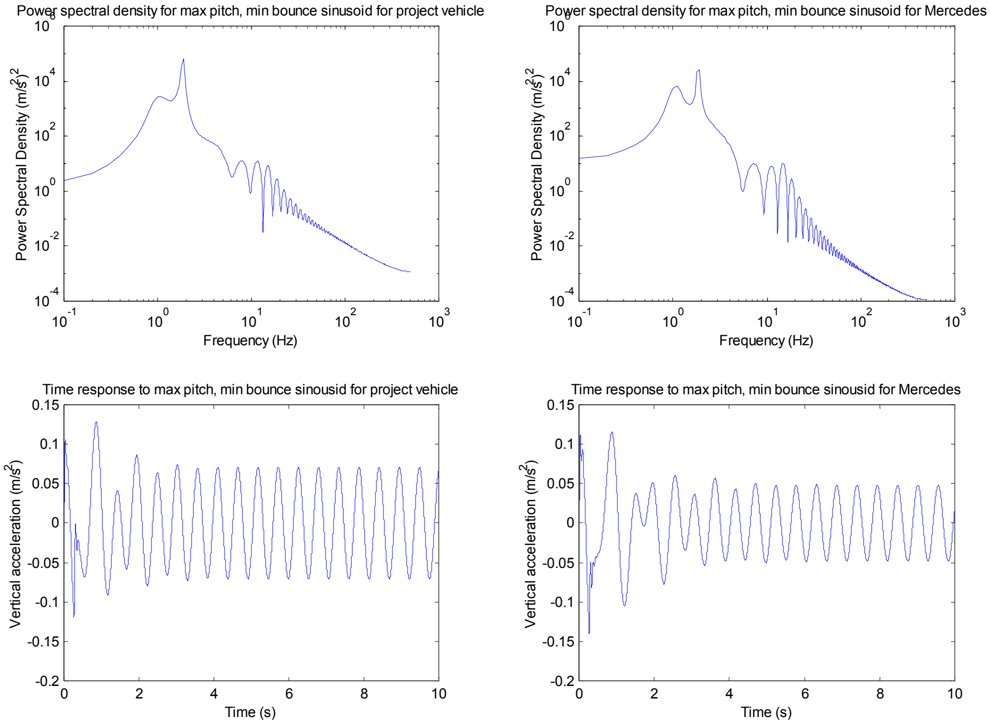


Here, the same plots are shown for the fully loaded vehicle. The same characteristics are present with reduced magnitudes due to the heavier mass and effectively lower rates.



Power spectral density graphs are given here for a sinusoidal road that gives maximum bounce and minimum pitch. There is still some minor vertical acceleration as shown in the accompanied time response plots. Clear peaks are shown at the frequency of the steady state sinusoidal motion as well as some higher frequencies that are coming from the transients.

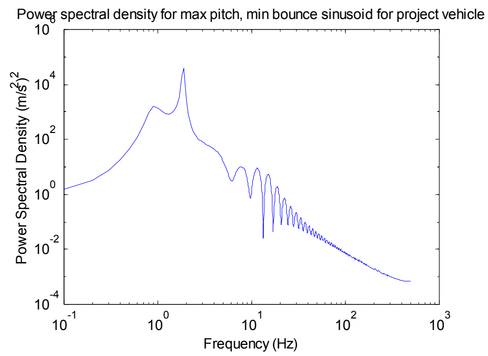
MODIFIED MERCEDES

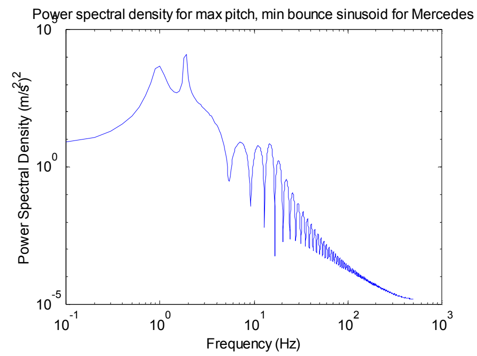


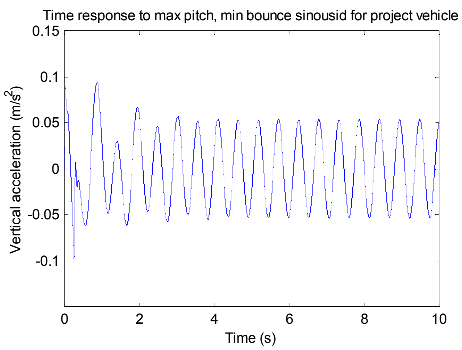
Here are the same plots for the fully loaded vehicle, showing lower magnitudes due to the higher mass.

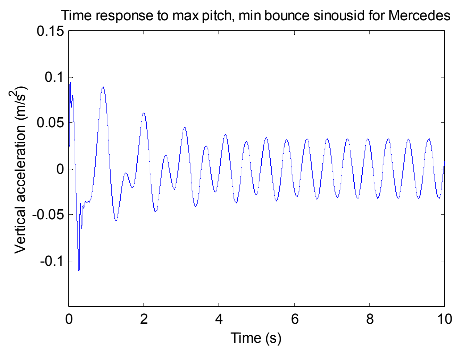
MODIFIED LOADED

MERCEDES LOADED



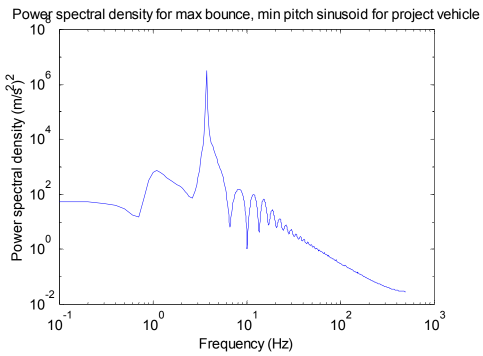


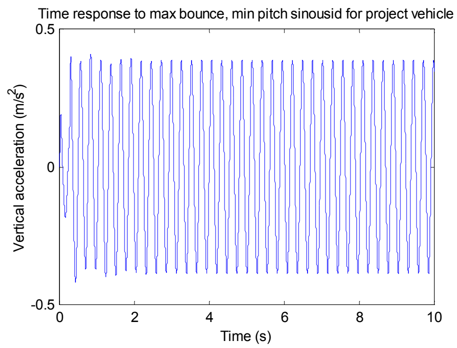




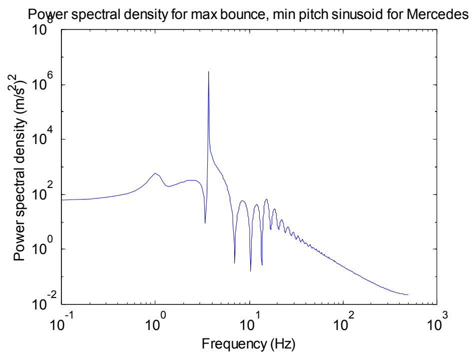
The vehicles are now traveling over a road that gives maximum bounce and minimum pitch, which is the reason for the higher vertical accelerations. Once again, the peak in the power spectral density plots reflect the fundamental frequency of the vertical acceleration.

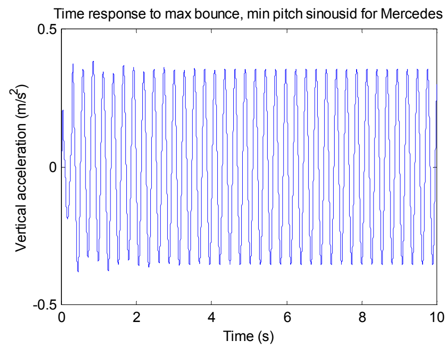
MODIFIED





MERCEDES





Here are the same plots for the loaded vehicles, showing similar characteristics but with lower magnitudes.

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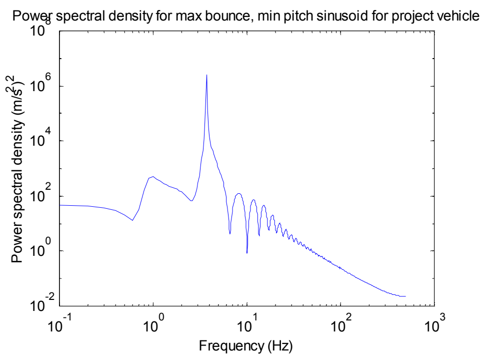
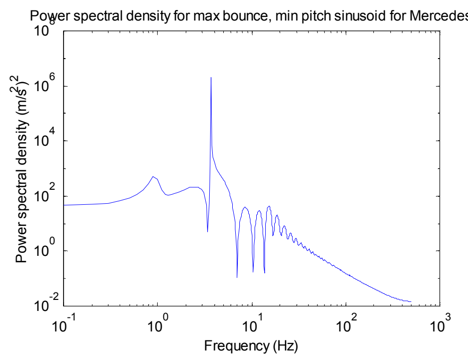
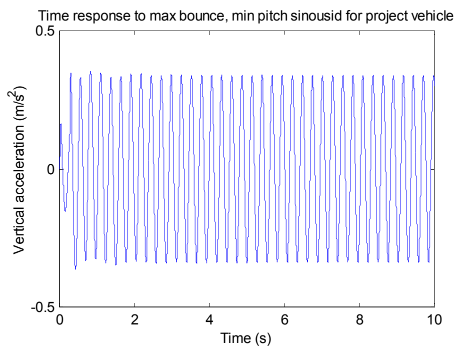
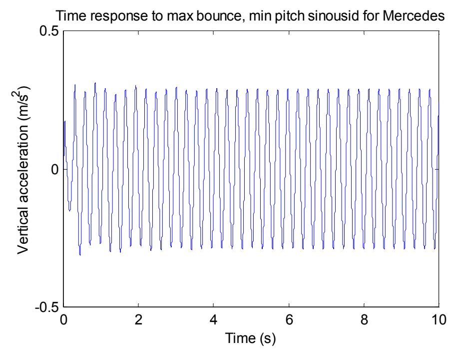
MERCEDES LOADED









Active Anti-Roll Bars

For the final part of the project the team decided to design an implement an active anti-roll bar. This device would allow the car to be very close to neutral steering while ensuring the vehicle would always be understeering at the limits, regardless of loading. Ideally the car would demonstrate almost no change in handling characteristics between loaded and unloaded cases.

The team's first attempt at implementing the active device was a closed loop system that compared an "ideal" understeer gradient to the actual. To do this a vector was created that gave the desired steer angle for a given lateral acceleration, thus defining a curve that could be used as a baseline for comparison. So, as the program moves through the range of lateral accelerations and calculates the steer angle, it finds the steer error based on the equation:

steer_error=steer_angle-steer_desired

The sign of this result indicates whether the car is too understeering (steer_error is positive) or too oversteering (steer_error is negative). This information is then used to decide if more load transfer is needed on the front or the rear to produce the desired curve and from this an appropriate action was taken. In addition to the desired curve, a desired roll rate was specified which dictated an overall required stiffness. How this stiffness was to be distributed between the front and rear was a function of the needed load transfer. What this appropriate action would be was where problems were encountered. Many different algorithms were pursued while trying to get this to work, the last being as follows:

K_act_f(i)=(US_bias(step_vector(j))*track_f/track_r*(Ksphir+K_needed(i)+Wr*hr*G(i)-Ksphif-Wf*hf*G(i)))/(1+US_bias(step_vector(j))*track_f/track_r)

Where K_act_f(i) is the needed active stiffness on the front of the vehicle and US_bias and step_vector are vectors that provide a scale for the gain. This is an example of when the car was found to be too understeering. Once K_act_f was founded it was subtracted from K_needed (defined by roll rate) to get K_act_r (the needed rear stiffness). In this code each iteration of the loop has an incremental change in gain in an attempt to move within a preset allowable deviation of the desired understeer gradient.
An earlier version of the code used the steer error to proportion the gain using the following equation:

USgain=steer_error(i)/(steer_error(i)+3*count-3)

This is again the case where the car was too understeering and here the gain is set by the steer error and iteration number (count). This value starts out as unity and gets smaller following each run. The required front and rear roll stiffnesses are determined during each run using the equation:

K_act_f(i)=(USgain*track_f/track_r*(Ksphir+K_needed(i)+Wr*hr*G(i)-Ksphif-Wf*hf*G(i)))/(1+USgain*track_f/track_r)

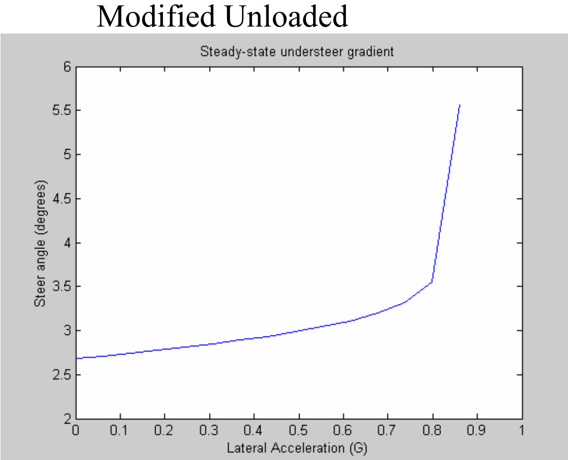
Once the gain for a run is entered into this equation and the spring stiffnesses are calculated, the steer error is re-evaluated and the sequence starts over.

Both of the above cases attempted to run iterations until the steer error was less then a preset level, but for any significantly small preset level the code would time out when the maximum number of iterations is reached. The above descriptions give the two general methods the team pursued. Multiple variations of each were tried, but none were able to converge to create an understeer gradient to match the steer_desired vector and the team decided to move on to an open loop system. This was about the time that I fell asleep, so Kevin and Harris will explain what they did in the open loop system.

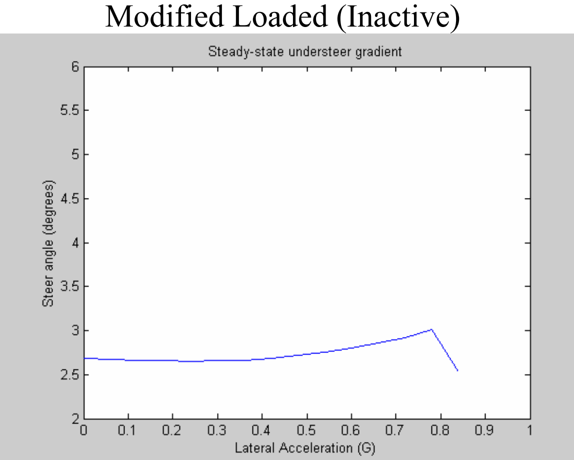
A new system was devised to control the anti-roll bar torque. The purpose of this system was to make a loaded vehicle's understeer gradient curve look like an unloaded vehicle's curve. Trial and error was used to determine how stiff to set the front and rear anti-roll bars for each lateral acceleration. The coefficients needed to produce these stiffness were then placed in a look-up table. This open-loop system was a simple last-ditch effort at controlling the anti-roll bars after all else had failed, and Bob had passed out.

Tuning this system was fairly simple and helped the team better understand the effectiveness of anti-roll bars. To begin, the modified (in-active) vehicle's setting for the anti-roll bars were used. The understeer gradient of the loaded modified vehicle was then plotted to determine if the vehicle was understeering or oversteering, relative to the unloaded modified vehicle's curve. If the vehicle was understeering, stiffness was added to the rear bars, and removed from the front bars. If oversteering, stiffness would be removed from the rear, and added to the front. The coefficients that determined how much stiffness would be added or subtracted from the bars were entered into the look-up table. The analysis was then re-run with the loaded modified active vehicle. This tuning process continued until the loaded vehicle's understeer gradient matched the unloaded vehicle's gradient as closely as possible.

Modified Unloaded

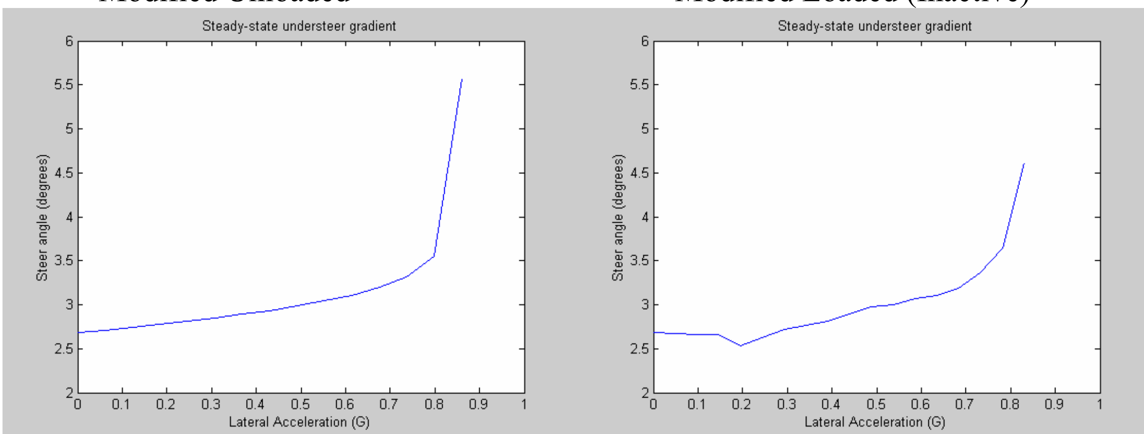


Modified Loaded (Inactive)



Above left is the unloaded understeer gradient curve. The curve on the right is the modified vehicle when loaded. After the tuning the anti-roll bars using trial-and-error, the new understeer gradient for the loaded vehicle with active bars looked like the following:

Modified Unloaded Modified Loaded (Inactive)



The plots above show that the look-up table method is a valid way to control the anti-roll bars. It can be seen that the active anti-roll bars can make an oversteering vehicle understeering. However, under about 0.4 g lateral acceleration, the anti-roll bar does not do much good to influence the understeer gradient curve as there is insufficient total load transfer to affect the tire cornering stiffnesses. Thus, the low G handling characteristics have to be tuned with the tire choice and weight distribution which are more dominant effects.

Lastly, the team output the actuator torque if the anti-roll bar were to be implemented with a motor mounted along the twist axis of the stock anti-roll bars. Values of about 2 kN were obtained. In fact, the rear bar had to twist in the direction to roll the car in order to get the loaded car to understeer. The physicality of such a system may not be realistic, but the team had fun with the concept (except for the coding).

Conclusion

In summary, the team has modified a stock BMW 318is for increased performance, especially with added payload as is the case for a driving school car that takes extra passengers to high lateral accelerations.

Although the team retained basic dimensions and weights of the BMW 318is, it was able to redesign the suspension geometry for improved roll center locations for reduced scrub and new camber gain curves for optimal cornering, even in the loaded condition. The rates were set to provide a stiffer setup to a more balanced handling until a higher lateral acceleration with increased rollover resistance despite higher load. Ride quality did suffer slightly, although the use of moderately high damping ratios smoothed out resonance peaks.

Finally, an active anti-roll bar system was implemented to retain desirable understeer gradient characteristics at mid- and high G loads.