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ABSTRACT

Race teams are interested in understanding the influence of the various structural members on the torsional stiffness of a NASCAR Winston Cup race car chassis. In this work we identify the sensitivity of individual structural members on the torsional stiffness of a baseline chassis. A high sensitivity value indicates a strong influence on the torsional stiffness of the overall chassis. Results from the sensitivity analysis are used as a guide to modify the baseline chassis with the goal of increased torsional stiffness with minimum increase in weight and low center-of-gravity placement. The torsional stiffness of the chassis with various combinations of added members in the front clip area, engine bay, roof area, front window and the area behind the roll cage was predicted using finite element analysis. Torsional stiffness increases and weight from several competing chassis designs are reported. Twist angle and the rate of change in twist angle under torsion is calculated at several locations along the frame. With strategic placement of structural members to a baseline chassis, the torsional stiffness can be more than tripled with only a 40 lb increase in weight.

INTRODUCTION

Increased torsional stiffness of a race car chassis improves vehicle handling by allowing the suspension components to control a larger percentage of a vehicle's kinematics, i.e., predictable handling can best be achieved if the chassis is stiff enough so that roll stiffness acting between the sprung mass and the unsprung mass is due almost entirely to the suspension [1]. In addition, a race car chassis must have adequate torsional stiffness so that chassis structural dynamic modes do not adversely couple with the suspension dynamic modes.

Winston cup racing teams typically purchase their basic chassis from two different manufacturers -- Hopkins or Laughlin. These baseline chassis are called `roller chassis', and are typically modified by adding structural members for increased strength or stiffness. While designing a new chassis or modifying a roller baseline chassis, structural members must be strategically located in order to reduce twist of the frame and minimize local deflections of suspension support points. In order to reduce twist and deflection, a minimum level of chassis stiffness must be achieved, while at the same time keeping the chassis weight and center-of-gravity low.

So far only very few attempts have been made to model a chassis of a Winston Cup race car in a way that torsional stiffness prediction can be made [2]. Structural design of these chassis have been done by trial and error method, possibly accompanied by some simple measurements of torsional stiffness.

The main objective of this study is to develop an improved design for a Winston Cup chassis structure with increased torsional stiffness based on simple modifications to a baseline Hopkins chassis. The goal is to design a new chassis design with at least three times the torsional stiffness of a Hopkins chassis with a minimal increase in weight and low center-of-gravity placement. In order to achieve this goal, a sensitivity analysis will be performed on a baseline Hopkins finite element model to help identify the structural members with the most influence on the torsional stiffness. Based on this information, the torsional stiffness for several different chassis configurations will be computed using finite element analysis. Results from the sensitivity analysis are used as a guide for strategic placement of members with the greatest impact on increased torsional stiffness. In addition, the twist angle and rate of change in twist angle along the side rails due to torsion will be calculated to help identify flexible areas of the competing chassis designs. A Physical 1/20 scale rapid prototype (RP) model was built to aid in visualization and to check for chassis/component packaging clearances. Finally, a sensitivity analysis of the final design is used to identify areas which may be further modified to increase chassis torsional stiffness.

CHASSIS DESCRIPTION AND DESIGN CONSTRAINTS

The main components of a Winston Cup race car are the chassis and the front and rear suspension. The chassis consists of the front clip, main cage, and the rear clip (see Figures 1 and 2). The front clip includes frame members and tubing forward of the firewall, the main cage houses the driver, while the rear clip consists of all members behind the main cage. The function of the chassis is to provide safety for the driver and a stable platform for mounting engine, transmission, and suspension components. The members of the chassis are constructed primarily of rectangular box beam members, and circular cross-sectioned tubular members, many of which are specified by the National Association of Stock Car Auto Racing (NASCAR)[3]. The dimensions of a Winston Cup race car with body shell is approximately 175 inches long, 56 inches wide with 44 inches height. The chassis material is primarily mild carbon steel. All frame joints use continuous welds.

The basic roll cage consists of tube members with a specified outside diameter of 1.75 inch and a minimum wall thickness of 0.090 inches. Figure 3 shows the basic frame rail of the chassis structure. The side rails consist of rectangular box beam members with a minimum of 3 inches width and 4 inches height, and wall thickness of 0.120 inches.

Figure 4 shows the sheet metal in the fire wall and the floor pan regions of the chassis. The sheet metal thickness varies over the chassis. The thickest part is the floor pan with a thickness of 0.065 in while the thickness of the firewall and the sheet metal in the rear is 0.04 in. All sheet metal is tack-welded onto the frame rails and the roll cage.

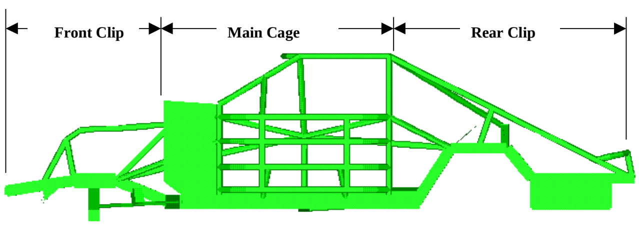

Figure 1. Side View of the Chassis: Front Clip, Main Roll Cage and Rear Clip.

NASCAR rules specify several constraints on the design of a Winston Cup race car. These rules are based on ensuring safety, limiting cost for race teams and fair competition. The most important rules restricting the chassis design are:

- A minimum of weight of 3,400 lbs (with a minimum right side weight of 1,600 lbs)
- Construction of the engine firewall of at least 22 gauge steel.
- Parallel side frame rails with minimum length 65 inches, minimum wall thickness 0.120 inches.
- Front and rear sub frames minimum wall thickness equals 0.083 inches.
- 110 inch wheel base, minimum roof height of 51 inches.
- Maximum allowable width between outer edges of the frame rails is 60 inches.

Even with these NASCAR design constraints, there is considerable flexibility in the structural configuration options of the chassis, especially in the front and rear clip.

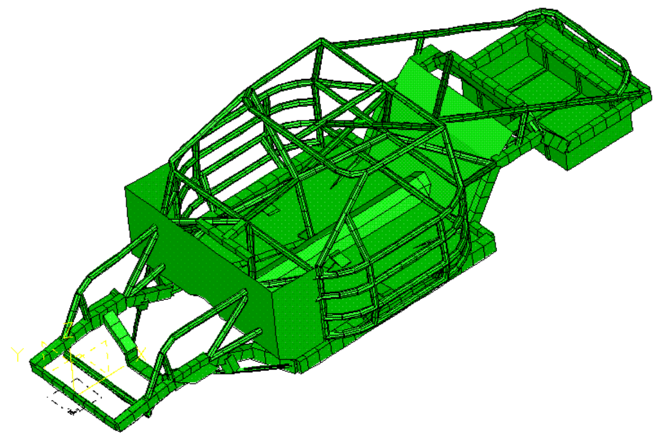

Figure 2. Isometric view of Finite Element Model (FEM) of a Baseline Hopkins Chassis

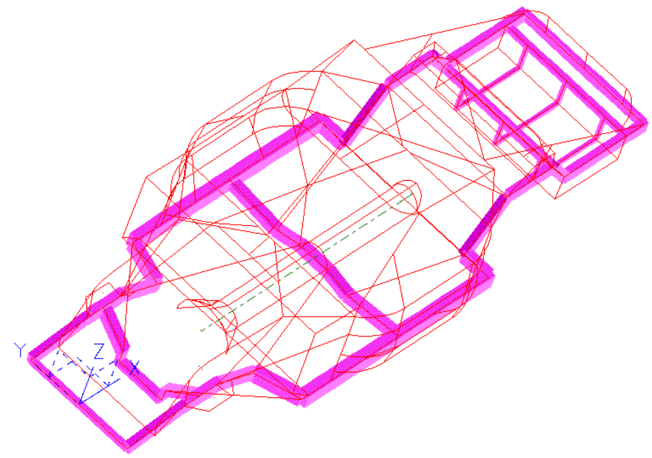

Figure 3. Frame Rail of the Chassis

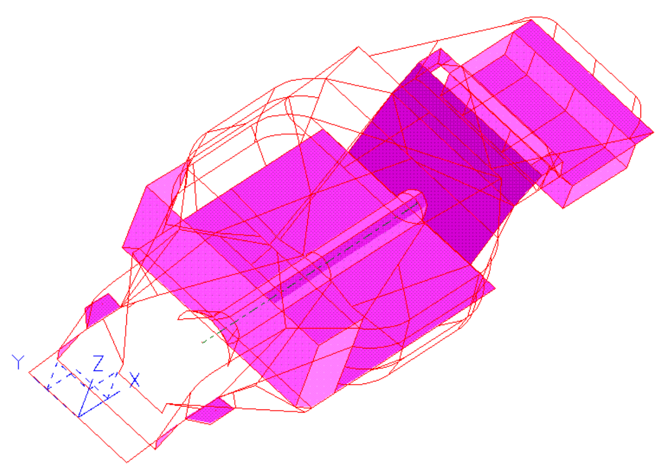

Figure 4. Sheet Metal in the Fire wall, Floor Pan Region and Rear Clip of the Chassis

FINITE ELEMENT MODEL OF BASELINE HOPKINS CHASSIS

The chassis model was constructed using I-DEAS (Integrated Design Engineering Analysis Software) from SDRC (Structural Dynamics Research Corporation) [4]. All computational work, including pre- and post-processing was performed on a Sun Sparc20 graphics workstation with dual 150 MHz HyperSparc processors and 196 Mb RAM. The geometry for the finite element model (FEM) of the unsuspended Hopkins chassis was measured at one of the race teams. The chassis was measured by projecting the centers of the welded joints onto a surface plate to determine the x-y components of the key-positions. The heights of the key-points above the surface plate were measured to determine the z-coordinates. The origin of the chassis coordinate system is placed at the center-line, at the front-most part of the chassis. This coordinate system is defined such that the positive x-axis is directed toward the rear, the positive y-axis is directed toward the right along the front lateral cross-member, and positive z is perpendicular to this x-y plane directed up. Linear beam elements are used to model the frame rail members with box-beam cross-section. The tube members are modeled using straight pipe elements, with circular cross-section. The sheet metal for the firewall and the floor pan sections are modeled using thin shell elements having an appropriate thickness. Local detail of the spring perches is not modeled. For simplicity, a rigid flat plate model is used with a large thickness dimension. Figure 2 shows the FEM of the baseline Hopkins chassis. The chassis model was constructed of steel with material properties given in the following table.

Modulus of elasticity	3.049×10^7 psi
Poisson's ratio	0.3
Weight Density	0.2847 lbf/in^3

The following assumptions were made for the chassis model:

- Key-point geometry is measured within 0.25 inches.
- Some sheet metal in the rear was difficult to measure and is neglected, under the assumption that it will not alter the stiffness results significantly.
- Contact of tube members and the sheet metal of the chassis, which may be created by deflection and would interfere with the normal deflection of those members is ignored in the FE model.
- Tube and beam connections were assumed under the usual structural frame assumptions of neutral axis intersection, with full coupling of shear and moments. This assumption should accurately represent the stiffness of the joints reinforced with gussets. However, where tubes are welded to one face of box beam frames, and are not fully integrated at the junctions, these assumptions will in general lead to a stiffer beam connection than the physical system.
- The material is assumed linear elastic and calculations are performed using linear static analysis with small deformations resulting in constant stiffness predictions.

CHASSIS TORSIONAL STIFFNESS ANALYSIS

In order to evaluate the torsional stiffness of the chassis structure boundary conditions are applied to the model as shown in Figure 5.

- A torque is applied to the front end of the chassis by applying equal and opposite vertical forces on the frame rails at a point in the vicinity of the front suspension pick-up points on the driver's and the passenger's side. A force $F = \pm 768$ lb is applied producing a torque, $T = F \cdot d = 2000$ ft-lb, where d = 31.25 inch, is the lateral distance between the driver and the passenger load application points.
- At the rear suspension spring mounts, the chassis is restrained in all x, y and z translations ($U_x = U_y = U_z = 0$) and in lateral and vertical rotations ($\theta_y = \theta_z = 0$), with the longitudinal rotations at these points free (θ_x = free). Other suspension-to-chassis connection points are not restrained.

These boundary conditions are representative of constraints applied by a twist fixture used by several race teams to measure torsional stiffness [5]. Recent studies given in [6] have shown that these restraints at the rear spring perches are "over-constrained" leading to stiffness predictions which are elevated by 9% over the minimum constraint condition. However, for the purposes of this study, use of the boundary conditions described above is sufficient to predict relative changes between competing chassis configurations.

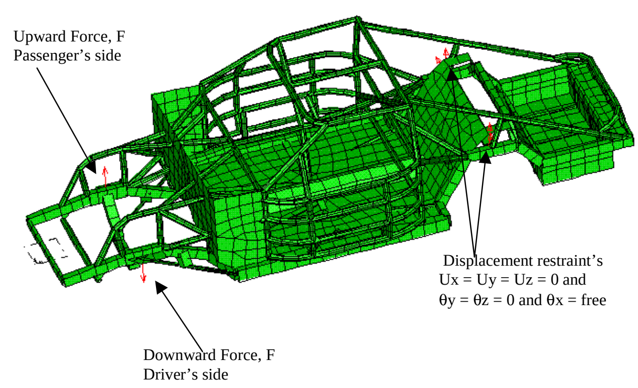

Figure 5. Applied Torque and the Restraints Used to Twist Chassis Models

The applied torque T = F d, produces a twisting effect on the chassis. Due to asymmetry within the chassis, the differential loading does not result in equal-and-opposite deflections at the front load points. The deflections v, of the front suspension pick-up points in the vertical z direction are measured in the post-processor. Torsional stiffness of the chassis is calculated from,

$$K_c = \frac{T}{\phi} = \frac{F \cdot d}{0.5 \cdot (\phi_d + \phi_p)} \qquad \text{(Eq. 1)}$$

ϕ is the average twist of the chassis due to the applied torque, ϕ_d, is the twist angle calculated from the vertical deflection v_d, of the frame rail on the driver's side,

$$\phi_d = \arctan\left[\frac{v_d}{d/2}\right] \qquad \text{(Eq. 2)}$$

ϕ_p, is the twist angle calculated from the vertical deflection v_p, of frame rail on the passenger side,

$$\phi_p = \arctan\left[\frac{v_p}{d/2}\right] \qquad \text{(Eq. 3)}$$

and d/2 = 15.625 inches is the lateral distance from the centerline to the passenger and driver side load points.

The rate of change of twist angle with respect to the longitudinal distance x, is also calculated from

$$\frac{d\phi}{dx} = \frac{\phi(x_{i+1}) - \phi(x_i)}{x_{i+1} - x_i} \qquad \text{(Eq. 4)}$$

where $\phi(x_i)$ is the twist angle at the point x_i measured on the frame rails.

With the differential loading and constraints discussed earlier, the torsional stiffness for the baseline Hopkins chassis is calculated to be K_c = 9934 ft-lb/deg which is approximately 9% higher than the stiffness predicted using ball joints at the rear spring perches in place of the rotational restraints. The weight of the chassis is 821 lb, while the center-of-gravity (CG) is at (x,y,z) = (78.04, 0.3933, 6.346) inches measured in the chassis coordinate system. The weight is calculated by multiplying the weight density of the material, with the volume of each element and summing for the total weight. The weight and CG calculation is performed automatically within I-DEAS.

SENSITIVITY ANALYSIS OF BASELINE HOPKINS CHASSIS

In order to determine the members with the greatest influence on the torsional stiffness of the baseline Hopkins chassis, a sensitivity analysis is performed using the optimization solver in I-DEAS [4]. Sensitivity analysis allows for parameter studies of beam cross-sections that provide information useful for redesign of the structure. For this analysis, sensitivity is the rate of change of displacement response of the chassis with respect to a change in the beam section design parameter. Sensitivity values gives insight into which parts of the chassis are most sensitive to change, and which parts are controlling its behavior. To monitor torsional stiffness, the vertical displacement at two points on the side frame rails near the front spring mounts are used as the structural response. A high sensitivity indicates a large change in flexibility of the structure when the design parameter changes. Areas with the highest sensitivity are targeted for redesign.The outer diameter for the tube members and the height for the box beam section are chosen for design parameters in this study. The height dimension of the box beam is used as the design parameter instead of the base since the beam section is stiffer (higher moment of inertia) in this direction. In order to facilitate the sensitivity analysis, each beam element in the chassis is assigned to a structural member or group of members. Each structural member or group is then assigned as a design parameter. In total, over 100 design parameters are defined.

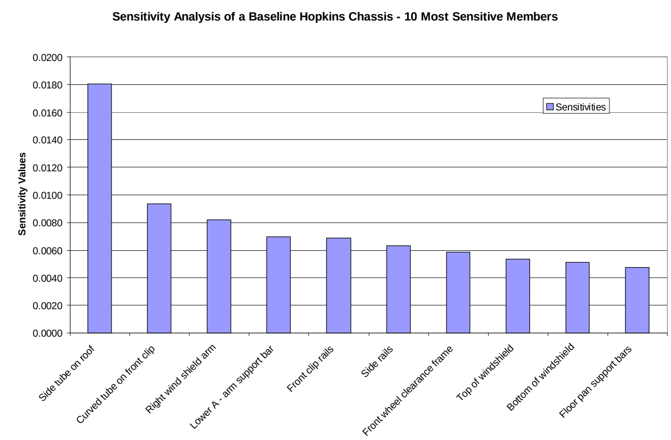

Figure 6. Sensitivities values for the ten most sensitive members to torsional stiffness.

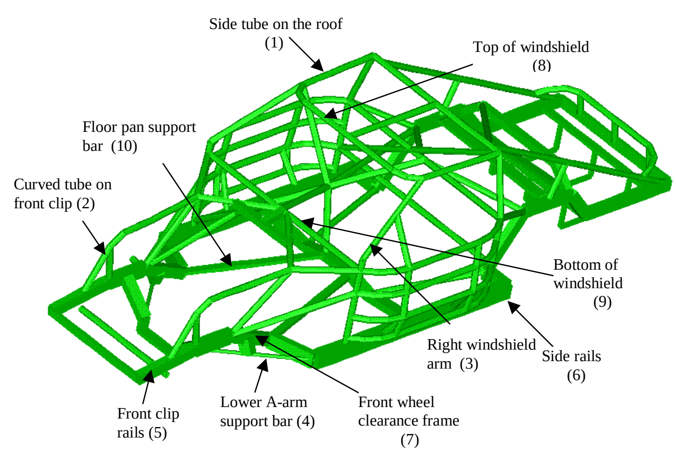

Figure 7. Locations of the ten most sensitive members for the baseline Hopkins chassis. (Sheet Metal Removed for Clarity)

The ten most sensitive members of the baseline Hopkins chassis to torsional stiffness are presented in Figure 6. Figure 7 shows the location of these critical members on the chassis. The areas with the most sensitive members occurred in the roof, windshield and front clip of the chassis. In summary,

- The side tubes on the roof are the most sensitive members indicating the stiffness could be increased by reinforcing these tubes with additional supporting members.
- The second most sensitive member is the curved tube in the front clip. This member has little lateral support between the left and right side rails of the front end of the chassis. Both the front clip, and front wheel clearance frames were listed in the top ten most sensitive areas, thus indicating that the front clip area is a good candidate for strategic placement of reinforcing members, or relocation of existing members.
- The side, top, and bottom members of the windshield area are listed in the top ten most sensitive members, indicating that torsional stiffness could be increased by reinforcing the windshield with supporting members.
- The inclined frame rail below the firewall in the transition region between the front clip and the roll cage area also has a relatively high sensitivity value.

The least sensitive members are located primarily in the rear clip and main cage area. This implies that structural reinforcement in the rear will not produce significant improvement in the overall torsional stiffness of the chassis.

TORSIONAL STIFFNESS OF ALTERNATIVE CHASSIS DESIGNS

Using the sensitivity analysis as a guide, various structural changes to the baseline Hopkins chassis are considered with the goal of increased torsional stiffness. While cross-sections of the structural members are used to identify sensitive members, the stiffness will be increased by adding supporting members or relocating members with standard tube diameters and thickness typically used by race teams. A total of 24 design combinations are considered in sequence, culminating in the selection of a final design with significantly reduced flexibility yet only a small increase in weight.

For convenience, the longitudinal axis of the car is divided into five sections as shown in Figure 8. Section A spans from $0 \le x \le 42.25$ inches and is the front section of the chassis. Section B is the transition section spanning from $42.25 \le x \le 47.5$ inches, between the front and the roll cage region. Section C is the main roll cage section that spans from $47.5 \le x \le 110.3$. Section D is the transition section between the main cage and the rear, spanning from $110.3 \le x \le 119.5$ inches. Section E is the rear chassis section spanning from $119.5 \le x \le 168.4$ inches. The load application points are at x = 23.5 inches.

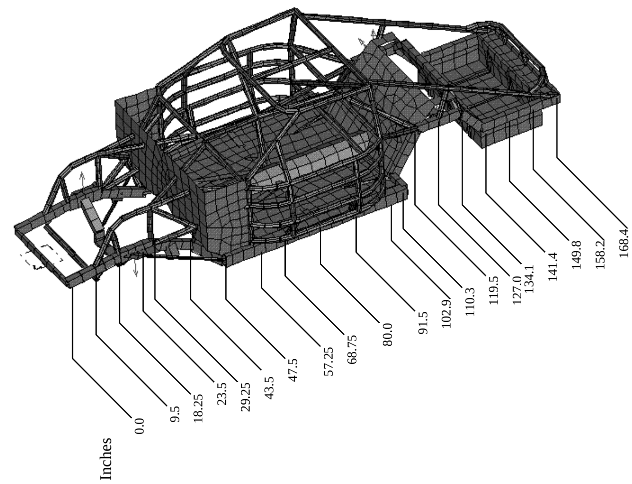

Figure 8. Longitudinal position in inches along the length of the chassis.

CASE 1: BASE HOPKINS CHASSIS – Figure 10 shows the twist angle for the base Hopkins chassis. As mentioned earlier, the chassis structure is asymmetrical with support bars in the passenger's side of the roll cage area and a rear diagonal bar right behind the roll cage (shown in Figure 9).

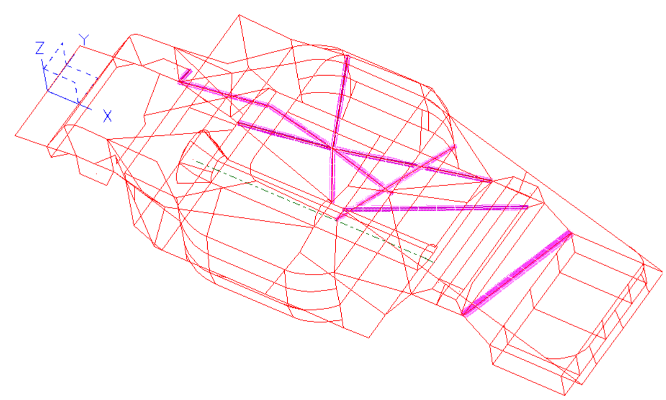

Figure 9. Asymmetry in the Baseline Hopkins Chassis (Isometric view from rear showing chassis coordinate system at front end)

As a result the change in twist $\Delta\phi = \phi_d - \phi_p > 0$, implying that the driver's side twists more than the passenger's side. At the front spring perches, the twist angle on the driver's side is 8 % higher than on the passenger's side. The deflections are largest in the front clip indicating this section would benefit from stiffening. The largest rate of change in twist occurs in the transition section between the front clip and the roll cage area, just forward of the fire wall, indicating that a large change in stiffness occurs in the transition region where the frame changes from narrow to wide.

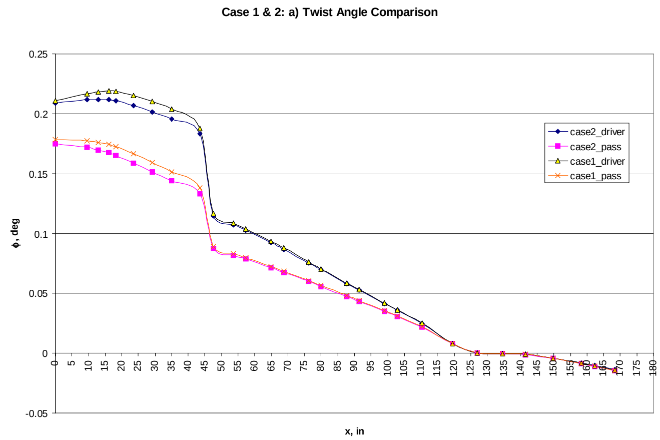

Figure 10. Comparison of Twist Angles of Case1 and Case 2

CASE 2: BASELINE MODIFICATIONS – In this case the side supports in the front clip are moved forward of the firewall and upper and lower bars are added on both the driver's and the passenger's side as shown in Figure 11. The upper bars and the vertical support bar are standard tube members with dimension 1.0" OD and 0.065" wall thickness. Straight tube, 1.5" OD and 0.065" wall thickness, is used for the lower bars. The diagonal bar in the rear was also removed. The torsional stiffness prediction (with y and z rotational constraints at the rear spring perches) increases to K = 10816 ft-lb/deg, which is 9% more than the baseline Hopkins chassis. The twist angle on both the driver's and the passenger's side decreases slightly in section A, as shown in Figure 10. The weight of this chassis configuration is W = 826 lb which is a 5 lb increase over the baseline.

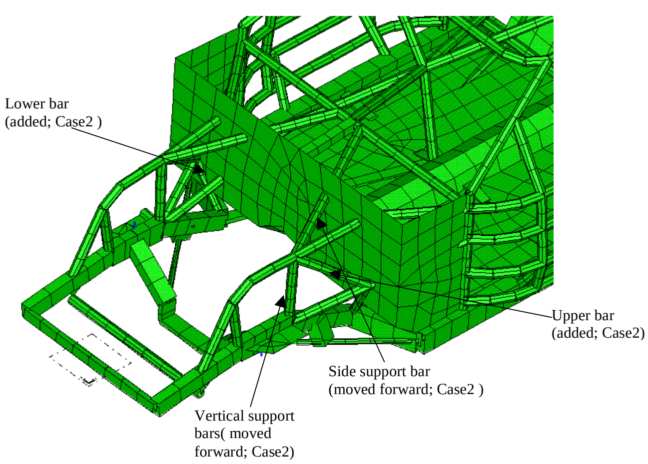

Figure 11. Structural changes made in Design Case 2.

CASE 3: ADDED A-BARS IN FRONT – In order to reduce the twist angle in the front of the chassis, several bars are added to the front clip as shown in Figure 12. These modifications include a horizontal support tube member with standard 1.0" OD and 0.049" wall thickness dimension, and two diagonal bars resembling an A-shape with standard dimensions 1.0" OD and 0.035" wall thickness. The cross-sections are smaller than the dimensions of the other tube members forming the roll-cage area to reduce weight and to provide more clearance to the engine, radiator, and other components in the front end of the chassis. For ease of serviceability and maintenance of the engine, these bars may be designed to be removable with bolted connections instead of permanent welds. For this case the torsional stiffness increases by 22% over Case 2. The increase in stiffness with respect to the standard baseline is approximately 33%. Figure 13 shows the twist angle for the driver and passenger side as a function of position along the length of the chassis. The twist angle in the front clip has been reduced significantly compared to Case 2, although there still is a large stiffness gradient in the transition region B. The increase in weight of these changes from Case 2 is only 2 lb. From these results, it is clear that the inclusion of these A-bars improves the torsional stiffness of the chassis significantly with only a very small increase in weight.

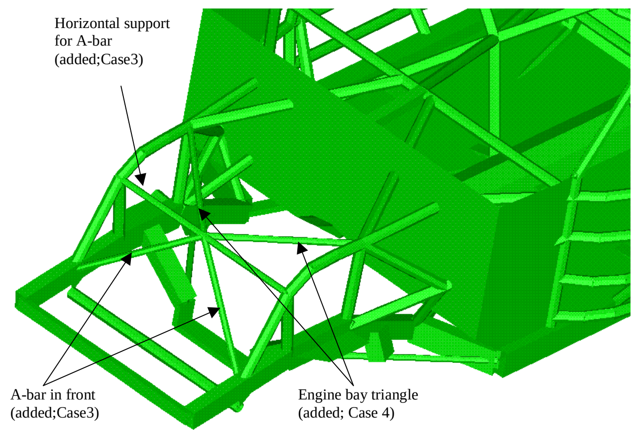

Figure 12. Structural changes made in Case 3 and Case 4.

CASE 4: ENGINE BAY TRIANGLE – To further reduce the twist in the front clip, a triangle is added to the engine bay area as shown in Figure 12. These members are positioned for adequate clearance to the engine and could also be designed to be removable members with bolted connections for easy access to the engine. The added members have the same standard dimensions as the front A-bar members. The torsional stiffness of this configuration is 49 % higher than the baseline and a 13 % increase over Case 3. Figure 13 compares the twist angle for Case 3 and Case 4. The twist angle in the front section A is significantly lower for Case 4 as compared to Case 3. The large rate of change in twist at the transition region B, between sections A and C is still present for this design case. The weight increased by only 2 lb over Case 3.

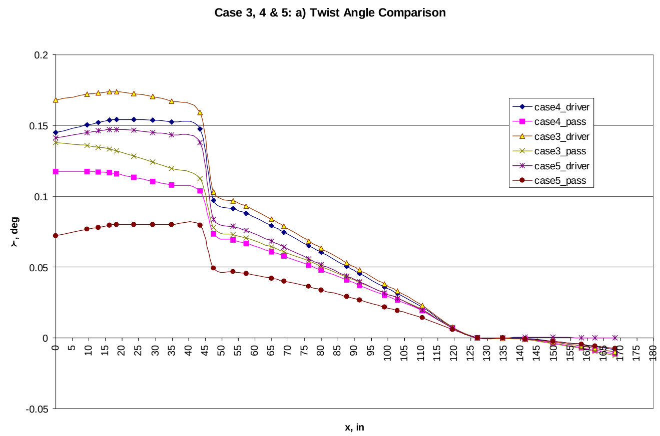

Figure 13. Twist Angle Comparison: Cases 3, 4 and 5.

CASE 5: CROSS BARS ON THE ROOF – To stiffen the roof area, several combinations of supporting members in different diagonal orientations were examined. The configuration with the most benefit to torsional stiffness is the cross-bars shown in Figure 14. Smaller sized bars with standard OD of 1.0” and thickness of 0.035” were used to minimize added weight to the top of the chassis in order to keep the center-of-gravity low. The torsional stiffness with the added roof bars is 77% higher than the baseline and an 18 % increase over Case 4. The weight only increased 2 lbs with this change. Figure 13 compares the twist angle between Case 3, 4 and Case 5. Interestingly, the twist angle on the driver side decreases only slightly while the twist angle on the passenger side decreases significantly. While the average twist angle is lowered for Case 5 with an increase in overall torsional stiffness, the difference in angles between the passenger and driver side has increased significantly, thus increasing the asymmetry in the chassis.

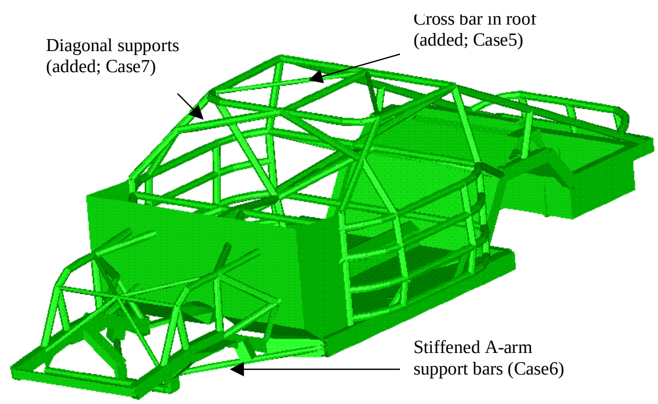

Figure 14. Structural Changes: Cases 5, 6 and 7

CASE 6: STIFFER LOWER A-ARM SUPPORT BARS – For this case, the straight pipe elements with 1.0” OD and 0.12” wall thickness used to model the lower A-arm support bars are replaced with a stiffer bar of 1.5” OD and 0.12” wall thickness, (see Figure 14). The torsional stiffness in this case is only 2 % higher than the previous Case 5. The twist angle is only reduced slightly by increasing the diameter of the lower A-arm support bars. The weight increase is 3 lb, however the added weight tends to lower the CG of the vehicle. The small change in deflection and stiffness was not expected based on the results from the sensitivity analysis.

CASE 7: A-BARS ACROSS THE WINDSHIELD – To increase the stiffness in the windshield region, diagonal bars are added across the windshield from the side tubes to the top center, see Figure 14. The added bars are straight tubes with standard 1.75” OD and 0.065” wall thickness. Figure 15 shows the twist angles along the length of the chassis comparing Case 6 and Case 7. The results indicate that the twist angle is reduced significantly on the driver side but only slightly on the passenger side, resulting in an average reduction in twist angle and an increase in torsional stiffness by 9 % over Case 6. The effect is to reduce the difference between the driver and passenger side twist angles, thus reducing the amount of asymmetry in the chassis caused by the addition of the diagonal roof bars.

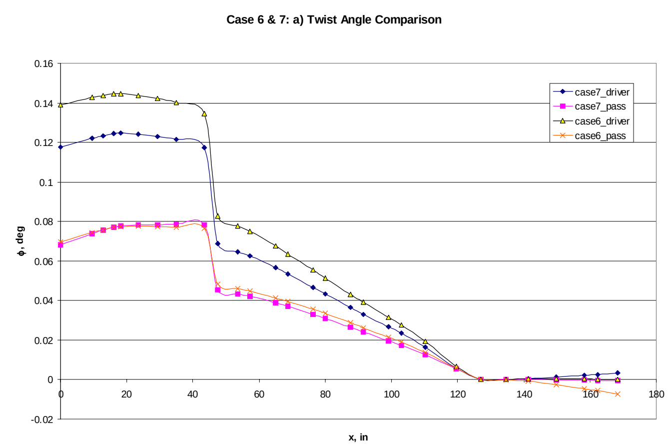

Figure 15. Twist Angle Comparison: Cases 6 and 7

CASE 8: REAR SUPPORT BARS – In this case, rear support bars are added to the chassis as shown in Figure 16. These members are intended to stiffen the frame rails around the rear wheels. However, the torsional stiffness increased by only 2 % over the previous case. The weight increased by 3 lb over Case 7.

CASE 9: REAR DIAGONAL MEMBER – In this case, the rear diagonal bars which were removed in Case 2 are re-added to the chassis as shown in Figure 16. The torsional stiffness increased by 5.5% over Case 8. The weight of the chassis increased by 5 lb with this change. The addition of the rear diagonal bar reduces the asymmetry substantially by reducing the twist on the drivers side and increasing slightly the twist on the passenger side, thus reducing the difference in angle between the two sides, see Figure 17. The twist on the driver's side is only 13 % higher than on the passenger's side at the load application points. Also the rate of change of twist angle in the transition section B is almost equal for both sides.

Thus it is clear that the rear diagonal bar plays a significant roll in controlling the asymmetry of the chassis stiffness.

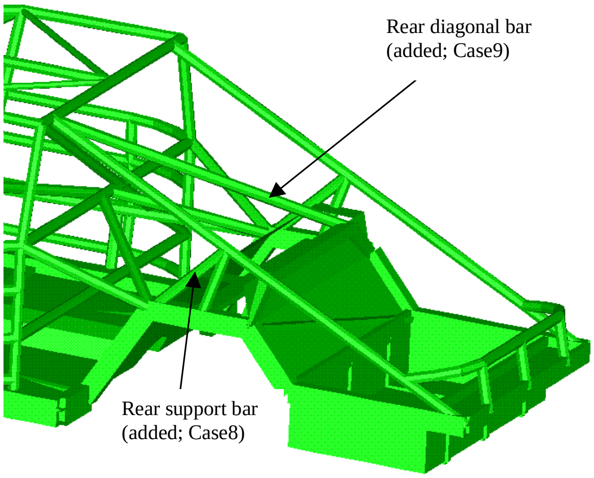

Figure 16. Structural Changes: Cases 8 and 9

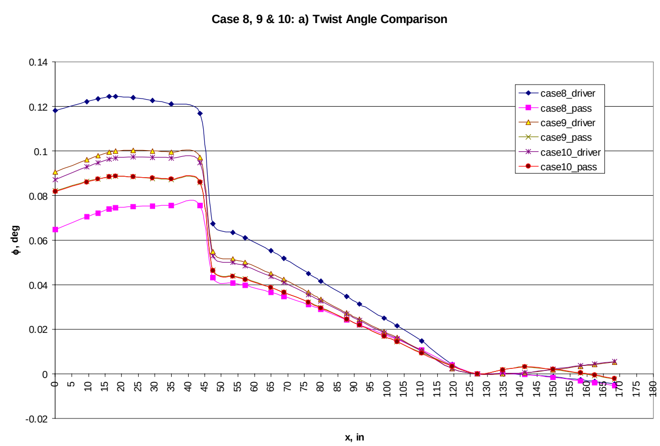

Figure 17. Twist Angle: Case 8 and 9 and 10

CASE 10: CENTER WINDSHIELD BAR – During the 1997/1998 season NASCAR changed the rules to require a bar down the center of the windshield as shown in Figure 19. This change was initiated after a driver was injured during a crash from an object entering the roll cage through the windshield. Thus the change was made for safety considerations. To study the effects of this new bar on torsional stiffness, the required center bar was added to the finite element model. As specified by NASCAR, the center windshield bar is standard 1.75" OD with 0.065" wall thickness and extends forward from the center of the roof bar, and down to a lateral support bar under the dash as shown in Figure 19. With this center bar added to the windshield the torsional stiffness increased by 2% over Case 9, while the weight increased by 4 lb. Figure 18 shows the twist angle comparing Case 8, 9 and Case 10. The twist angle for the driver's side in the front section A is slightly smaller than the previous case but the twist angle of the passenger side does not change.

CASE 11: STAR STRUCTURE BETWEEN ENGINE BAY AND ROLL CAGE – In order to further decrease the deflections in the front clip, the structural members between the fire wall and front dash were modified. The side and horizontal support bars in behind the fire wall area are removed and replaced with four bars forming a star structure shown in Figure 19. The bars are standard straight tube with 1.75" OD and 0.065" wall thickness and attach at a common point just behind the fire wall connecting the side bars in the engine bay and lower corners of the windshield. The arrangement of these bars gives a better triangularization between the engine region and the roll cage. The torsional stiffness increases significantly by 130 % over the baseline Hopkins chassis and an increase of 7 % over the previous case. As shown in Figure 61, both the driver and passenger side twist angles reduced nearly the same amount, leaving only a 10 % difference between the two sides. The weight increased by 3 lbs with this design modification.

CASE 12: MODIFIED SIDE BARS – To improve the load path, the side bars that run halfway through the engine bay and bend out to connect with the wide frame of the roll cage, are straightened as shown in Figure 19. This simple change increased the torsional stiffness by 4%, while at the same time reducing the weight by a large amount (9 lbs) over the previous case. As shown in Figure 18 the twist angles in the transition section between the engine bay and roll cage has been reduced considerably.

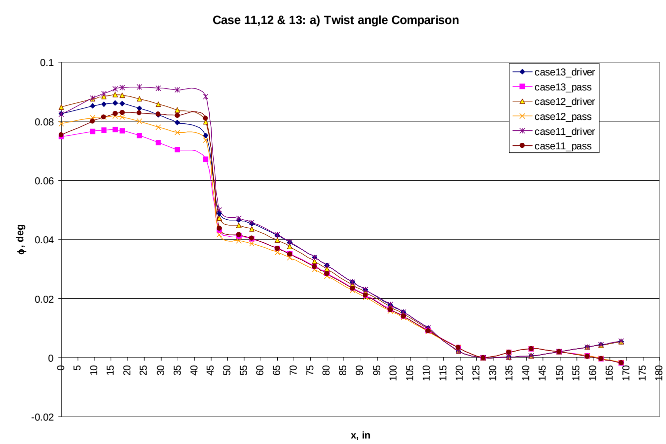

Figure 18. Twist Angle: Cases 11 and 12 and 13

CASE 13: LOWER SUPPORT BAR IN X-BAR STRUCTURE – To further decrease the twist in the transition Section B, a standard tube member is added from the lower end of the vertical post of the front roll cage, to the intersection of the upper and lower support bars. This additional bar completes an X-Bar structural configuration as shown in Figure 19. This change caused the torsional stiffness to increase by 5 % over Case 12. However, this increase in stiffness also came with an increase in weight of 12 lbs. The twist angle plot comparing Case 13 with Case 11 and 12 is given in Figure 18.

From these results, the twist is reduced for both the driver and passenger side uniformly throughout the length of the chassis.

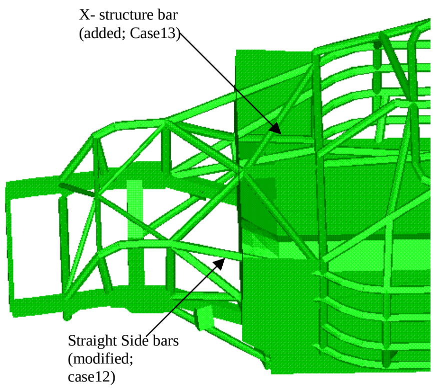

Figure 19. Structural Changes: Cases 10, 11, 12 and 13

CASE 14: V-BAR REPLACING THE REAR DIAGONAL BAR – In Case 9, it was observed that the addition of the rear diagonal reduces the asymmetry in the twist angle between the driver and passenger sides. In an attempt to reduce the deflections around the real wheel clearance frames and increase overall torsional stiffness, the rear diagonal bar is replaced with a V-bar structure consisting of standard pipe, 1.0” OD and 0.065” wall thickness. With this modification, the torsional stiffness decreased by 1 %. In addition, the difference between the twist angles and the rate of change of twist in the transition section increased considerably. The weight increased by 2 lb. From these results, it is clear that the diagonal bar used in Case 9 has more benefit to both torsional stiffness and reduction in asymmetry compared to the V-bar configuration considered in this case.

CASE 15: V-BARS REPLACING THE LOWER REAR DIAGONAL BAR – Similarly, replacing the lower diagonal bar in the area behind the rear wheel with V-bars does not improve stiffness significantly with a weight increase of 0.5 % compared to the previous case. Thus there is no significant benefit to changing the diagonal bar to a V-bar structure.

CASE 16: V-BAR STRUCTURES REPLACED WITH DIAGONAL BARS IN REAR – The V-bar structures of Case 14 and 15 did not improve torsional stiffness or reduce weight. Thus for this case, the V-bar structures added in Case 14 and 15 are changed back to the diagonal bar configurations of Case 13 except that the lower diagonal bar cross-section is changed from a box beam to a standard 1.75” OD and 0.065” wall thickness straight tube. This configuration results in a 4 lb reduction from Case 13 with a negligible change in torsional stiffness from Case 13. The twist angles are nearly identical to Case 13. Since the weight was reduced with this change, all further design cases considered will use straight tube for the lower rear diagonal bars.

CASE 17: LOWER SUPPORT MEMBER FOR THE X-STRUCTURE REMOVED – In this case the vertical lower support member connecting the frame rail at the transition between the front clip and roll cage, and the intersection point of the X-structure, which was added in Case 2 is removed. With this deletion, the torsional stiffness decreases by less than 1 % while the weight decreases by 2 lbs compared to the previous case. The twist angles do not change significantly from Case 16. Because there is no significant change in stiffness or weight, this bar is removed for all further cases considered.

CASE 18: MODIFIED X-BAR STRUCTURE – In the X-bar structure in the previous configuration, one of the cross members starts from the side rail and connects to the second top door bar in the roll cage area. This member is moved in this case, to be connected to the top most door bar in the roll cage, to form an X-structure as shown in Figure 20. There is no significant increase in the torsional stiffness due to this change. The torsional stiffness increased by 1% over Case 17.

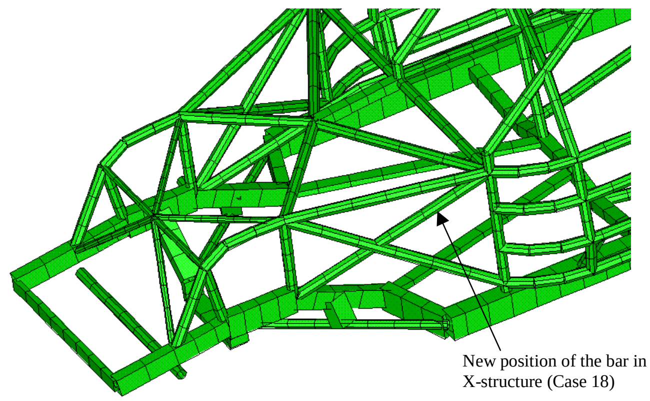

Figure 20. Structural Change: Case 18 (Sheet Metal Removed for Clarity)

CASE 19: MODIFIED WINDSHIELD A-BAR – In this case the position of the diagonal bars across the windshield added in Case 7 are modified to provide better visibility to the driver. In this case the bars are shorter and connect to points nearer to the edges of the windshield instead of at the center, see Figure 21 for the new positioning of these tubes. With this change the torsional stiffness reduced by 2 %, with a decrease in weight of 2 lbs compared to Case 18. Figure 22 shows the twist angle comparing cases 18 and 19. With the change in position of the windshield bars the twist angle on the drivers side increases slightly while the passenger side does not change. While this modification increases the flexibility and asymmetry of the chassis twist behavior, the change is necessary for improved visibility and safety of the driver and will be included in the remaining cases considered in this study.

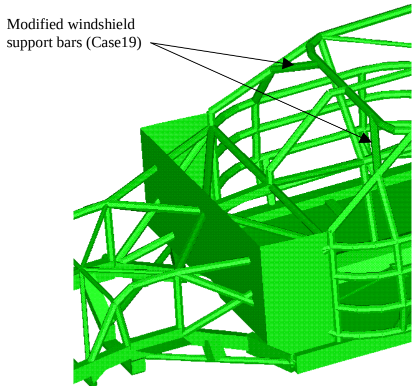

Figure 21. Structural changes: Case19

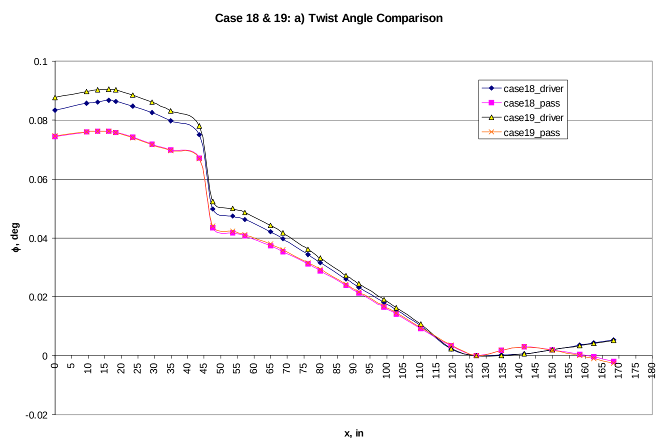

Figure 22. Twist Angle: Case 18 and 19

CASE 20: FLOOR PAN SUPPORT BARS – Additional floor pan support bars are added in the front and the rear section of the main cage area as shown in Figure 23. The front floor pan support bars are rectangular beam of dimension 1.0” base x 2.0” height x 0.12” wall thickness and the rear floor pan support bars are straight pipe of dimension 1.75” OD and 0.065” wall thickness. This results in a substantial increase in weight without much increase in torsional stiffness. The torsional stiffness for this configuration increased by only 1 % over Case 19 while the weight of the chassis increased significantly by 20 lb over the previous case.

CASE 21: VERTICAL V-BAR BEHIND FIREWALL – In this case a V-bar structure connecting the frame rails is added immediately behind the firewall as shown in Figure 23. The V-bar structure consists of rectangular box beam members with 1.0” base x 2.0” height x 0.12” thickness. This addition increased torsional stiffness by 29 % over the previous case. The weight of the chassis increased by 14 lb. As shown in Figure 24 the twist in the front clip and transition area are reduced considerably from the previous case, especially for the driver's side. The large reduction in twist results from the connection of the driver side and passenger side rails near the transition section B. The large increase in stiffness from the addition of this V-bar is significant.

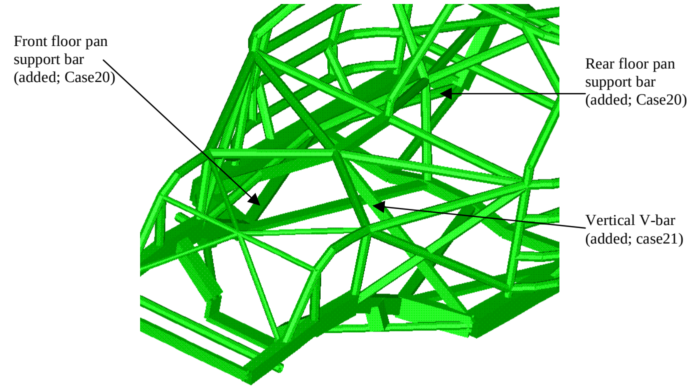

Figure 23. Structural Changes: Cases 20 and 21. (Sheet metal removed for clarity)

CASE 22: REPOSITIONED V-BAR – In this case the V-bar added in Case 21 is repositioned with the legs attached to the ends of the narrow side rails in the front clip instead of the wide side rails in the roll cage section, see Figure 31. This configuration increases the torsional stiffness by 6 % compared to Case 21 and also reduces weight by 4 lb. The torsional stiffness increases by 242 % compared to the baseline chassis. Figure 24 shows twist plots comparing Case 21 and Case 22. With the new positioning, the rate of change in twist angle on the driver's side is smoothed out considerably through the transition Section B. The rate of change in twist is also reduced on the passenger side and in contrast to all previous cases the twist angle increases slightly in the transition section before decreasing towards the rear.

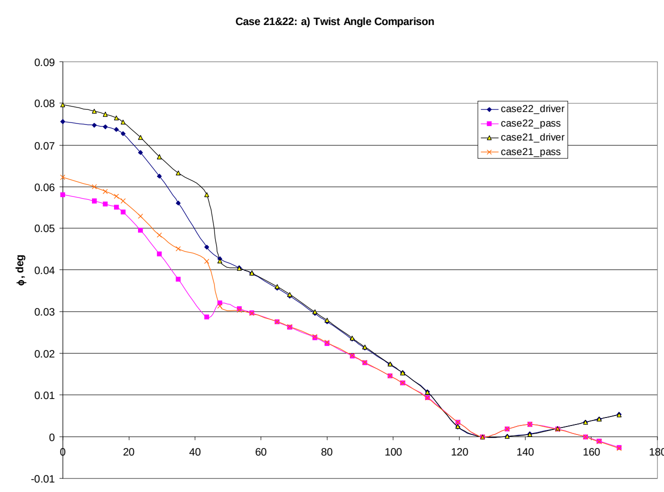

Figure 24. Twist Angle Comparison: Case 21 and 22

CASE 23: REMOVAL OF FLOOR PAN SUPPORT BARS – In this configuration, the floor pan support bars added in Case 20 are removed, since they do not account for a significant increase in the torsional stiffness of the structure. Removing the floor pan support members reduces the weight by 20 lbs, with only a 3 % decrease in stiffness. This stiffness value is 232 % higher than the baseline Hopkins chassis considered in Case 1, with only a 40 lb increase in weight. The series of changes made to the Hopkins chassis to arrive at the configuration of Case 23 is summarized below.

- Modifications of Case 2 with removal of lower support member (Case 17) and addition of the rear diagonal bar.
- Front and Engine bay A-bars (Case 3 and 4).
- Diagonal bars in the roof area (Case 5).
- A-bars and center bar across the windshield (Case 19 and 10).
- Stiffened lower A-arm support bars (Case 6).
- Rear support bars (Case 8).
- Modified X-bar structure in the front clip (Case 18).
- Star structure behind fire wall (Case 11 and 12).
- Vertical V-bar behind fire wall (Case 22).

The final structural configuration for Case 23 is shown in Figures 25 and 26.

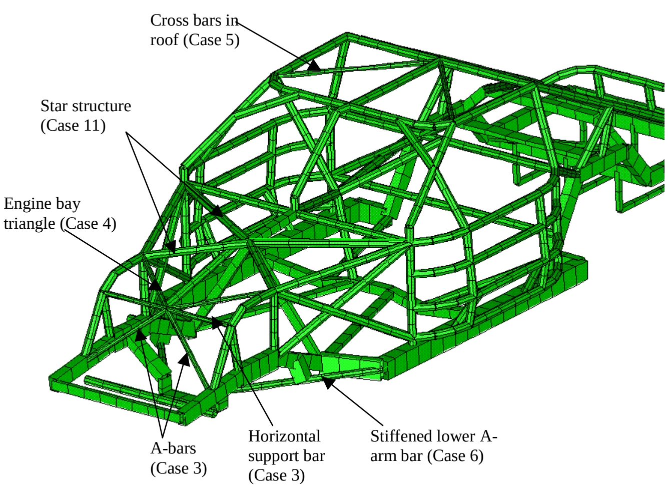

Figure 25. Isometric View of Case 23 Configuration (Sheet Metal Removed for Clarity)

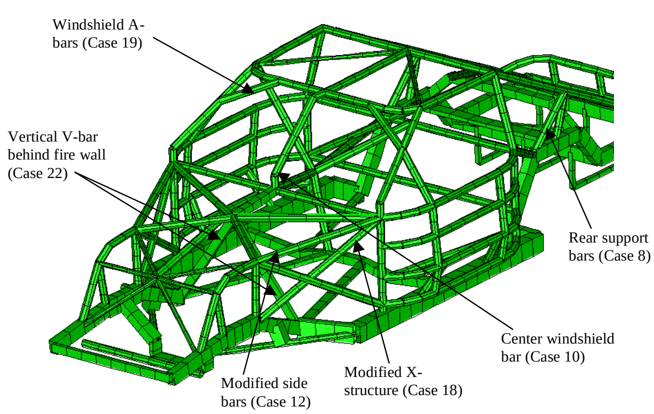

Figure 26. Isometric View of Case 23 Design (Sheet Metal Removed for Clarity)

CASE 24: MODIFIED SIDE FRAME RAILS – None of the previous configurations discussed have effected the frame rail design of the baseline Hopkins chassis. In this case, a modification to the frame rails is considered. In order to eliminate the sharp change between the wide spaced side frame rails of the roll cage and the narrow frame rails of the front clip, the layout of the frame in the transition section B is modified as shown in Figure 27. The side frame rails towards the base of the front clip and the inclined side frame rail under the firewall are replaced by a box-beam with dimensions 1.75” base x 1.75” height x 0.077” wall thickness, that runs from side rail in the front of the main roll cage to a position on the frame just aft of the suspension pickup flange. With this repositioning of the frame rails, the V-bar behind the firewall and floor pan support bars are also repositioned. With these changes, the torsional stiffness decreased by 7 % compared to Case 23, while the weight is lowered by 1 lb. The twist angle plots comparing Case 23 and Case 24 are shown in Figure 28. The twist angle changes smoothly throughout the front clip and transition section with small gradients in twist angle throughout on both the driver and passenger sides. This smoothing of the twist curves is attributed to the gradual change in the frame rails from narrow to wide through Section B. While this configuration has decreased stiffness compared to Case 23, it does have the benefit of a smoothly changing twist angle along the length of the chassis.

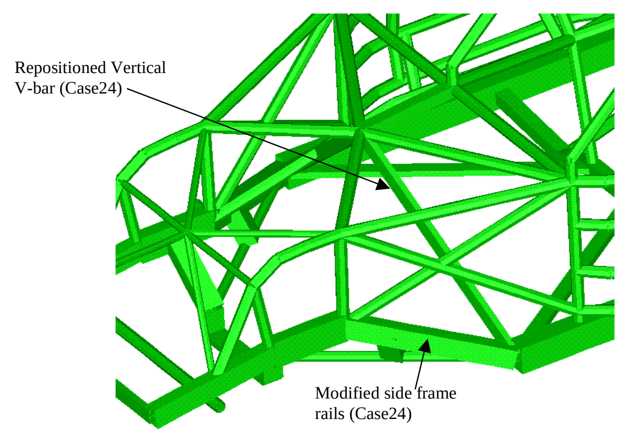

Figure 27. Structural Changes: Case 24 (Sheet metal removed for clarity)

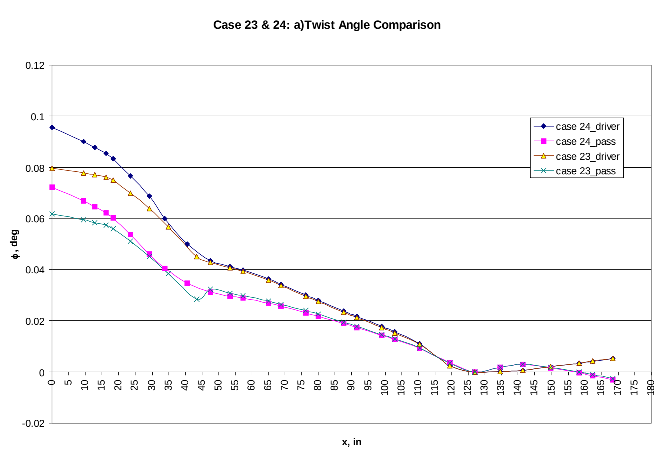

Figure 28. Comparison of Twist Angle: Case23 and 24

SUMMARY OF RESULTS

A summary of the torsional stiffness and weight of the 24 design cases that are considered in the study is given in Figure 29 and 30. Based on the torsional stiffness calculations, the most significant structural changes to the baseline Hopkins chassis in order of importance were:

1. V-bars behind fire wall (Case 22).
2. Diagonal bars on roof (Case 5).
3. Star structure (Case 11 and 12).
4. Front A-bars (Case 3).
5. Engine bay triangle (Case 4).
6. Windshield diagonal bars (Case 19).

The most significant increase in torsional stiffness occurred with the addition of the V-bar structure behind the fire wall described in Case 22. Another important member is the rear diagonal bar present in the original Hopkins chassis. This bar increases torsional stiffness by 5%, but more importantly the asymmetry in the twist behavior is reduced considerably with the addition of this bar.

The changes with the least influence on torsional stiffness in order are:

1. Rear support bars (Case 8).
2. Center windshield bar (Case 10).
3. Stiffened lower A-arm support bars (Case 6).

The added members with the greatest increase in weight were:

1. X-bar structure (12 lb).
2. V-bars behind fire wall (10 lb).

The other modifications considered increased weight by less than 5 lbs, for each case.

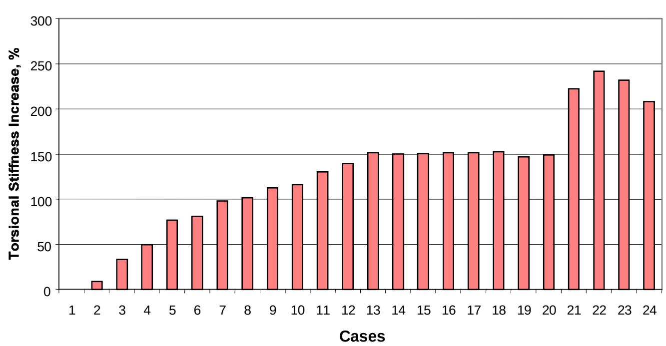

Figure 29. Percentage increase in torsional stiffness over baseline for the different chassis configurations.

Figure 31 gives the change in center-of-gravity (CG) for the different cases considered. The CG is based on the mass of the structural members represented in the chassis model, and does not include the car shell, suspension, engine, ballast and other weight of the car. All values of CG are reported with reference to the origin of the global coordinate system, which is at the intersection between the front-most member and the centerline of the chassis.

The added diagonal roof bars, diagonal windshield bars, center windshield bar and rear diagonal bar increased the vertical CG height by approximately 0.1 inch for each case.

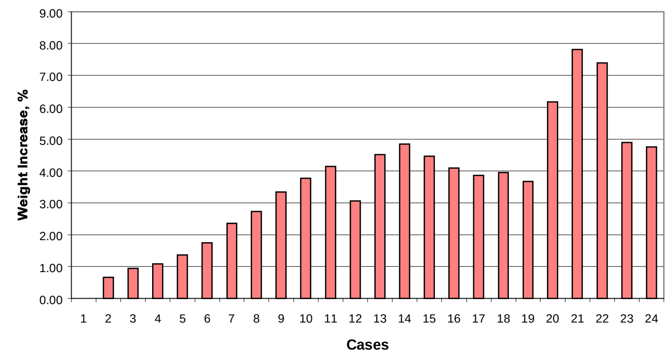

Figure 30. Percentage increase in weight over baseline for the different chassis configurations.

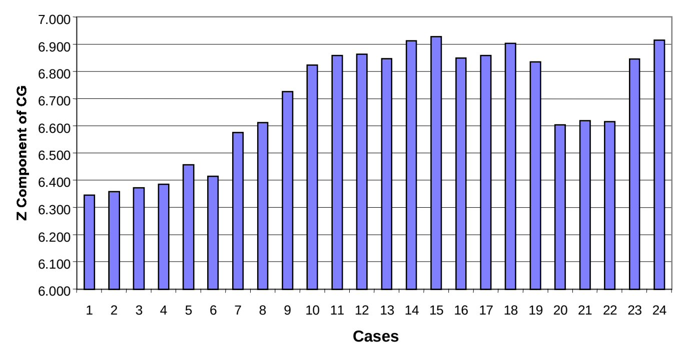

Figure 31. Vertical component of CG for the different chassis configurations.

SELECTION OF FINAL DESIGN

Most of the configurations studied contributed steadily towards increasing the torsional stiffness over the baseline Hopkins chassis. The over 35 % increase in torsional stiffness and negligible change in CG height with the addition of the V-bar more than offsets the 10 lb increase in weight. Other important members for increased stiffness include the removable front A-bar and engine triangle structures added in Case 3 and 4, the diagonal bars on the roof (Case 5), and the star structure behind the fire wall (Case 11). The floor pan support bars in Case 20 added 20 lbs to the chassis with only a 3% increase in torsional stiffness. While these bars lowered CG height by 0.2 inches, they did not justify the added weight with such a small increase in torsional stiffness. The change in the frame design in Case 24 reduces the weight by 1 lb, and smoothes the twist gradient along the length of the chassis. However, these benefits are offset by the over 7% reduction in stiffness compared to Case 23. The structural configuration of Case 23 includes the modifications with the most significant increase in torsional stiffness with minimal increase in weight and CG height and is therefore selected as the final design.

The torsional stiffness prediction of the final design is 232% higher than the baseline configuration. The weight of the final design is W = 861 lb (40 lb or 5% increase over baseline). The CG height of the final chassis design is 0.5 inches above the original baseline configuration, which is insignificant relative to the overall CG height of the car with a total weight of 3400 lb. The net increase in the total CG height of the car with the final design as compared to the baseline chassis is only 0.25 inches [7]. This improved chassis design requires only simple modifications and additions of standard tubing to the Hopkins chassis, yet does not require changing the configuration of the box beams of the frame rails.

Figure 32 shows a comparison of the twist angle on the driver and passenger side along the length of the chassis for the baseline Hopkins chassis (Case 1) and the final chassis design (Case 23). The twist angles of the new design are significantly lower than the baseline Hopkins chassis. Figure 33 shows that the high rate of change in twist angle in the transition section between the front and roll cage has reduced significantly in the final design with relatively uniform change in stiffness throughout the chassis length.

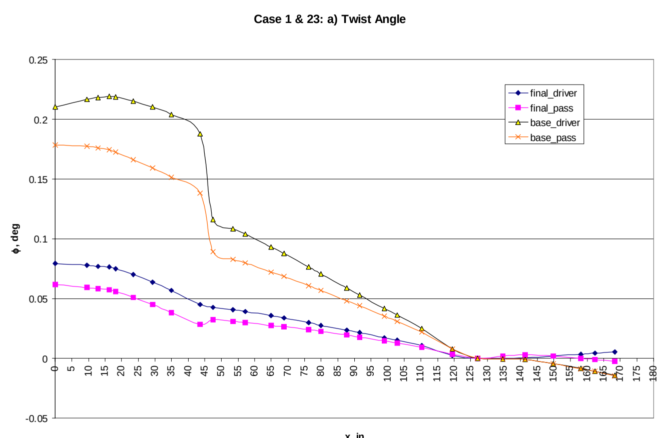

Figure 32. Twist Angle Comparison of Baseline and Final Design

SENSITIVITY ANALYSIS OF THE FINAL DESIGN

A sensitivity analysis of the final design Case 23 was performed to identify areas that may be modified to increase chassis torsional stiffness. The five most sensitive members in the final design are given in Figure 34. Figure 35 shows the locations of these members on the chassis. The largest sensitivity values for the final design are an order of magnitude lower than the Hopkins chassis indicating that there is little room for further increase in torsional stiffness as compared to the Hopkins chassis. The areas with greatest potential for further increase in torsional stiffness are the horizontal V-bar behind the firewall and the diagonal roof bars. Further reinforcement around the V-bar to stiffen the front transition section would have to be made such that sufficient clearance is allowed for engine and other components to be serviced. The diagonal bar on the roof could be increased in size, however this would also raise the height of the CG location.

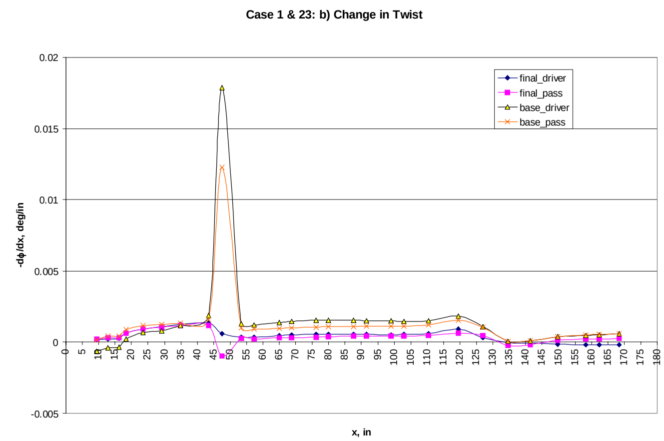

Figure 33. Change in Twist: Comparison of Baseline and Final Design

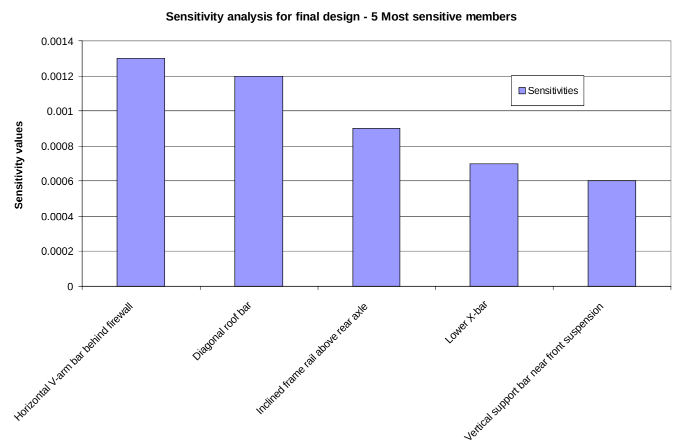

Figure 34. The five most sensitive members for the final design.

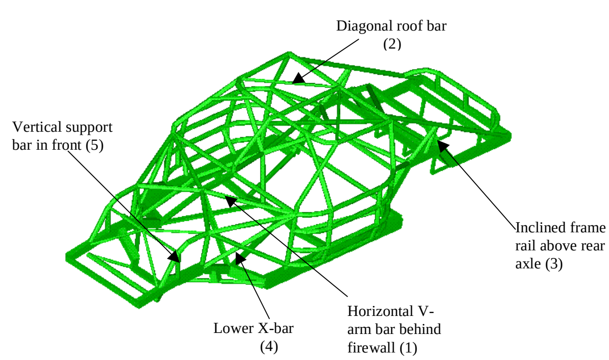

Figure 35. The 5 most sensitive members for the final design.

A rapid prototype (RP) at 1/20th scale of the final design was built in the Clemson Rapid Prototyping Lab using a Stereolithography (SLA) process, see Figure 36. This prototype was used to check for chassis/component packaging clearances and shared with race team engineers and managers to quickly communicate design changes.

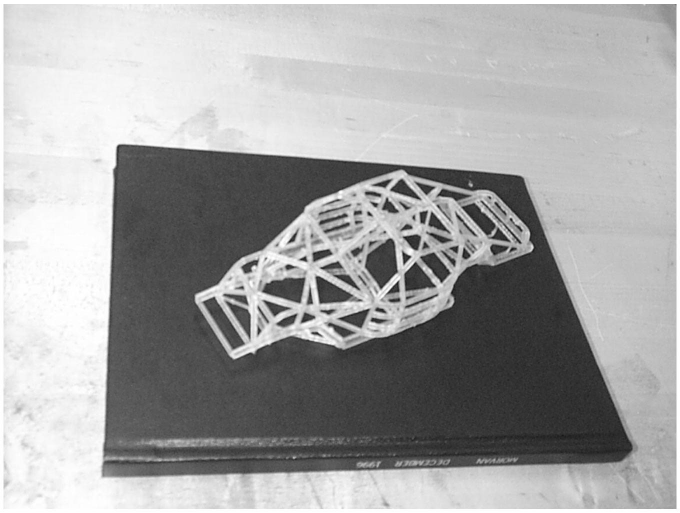

Figure 36. Rapid Prototype of Final Design

CONCLUSION

The sensitivity values for individual structural members on the torsional stiffness of a baseline Hopkins chassis were determined. A high sensitivity value indicated a strong influence on the torsional stiffness behavior of the overall chassis. Results from the sensitivity analysis identified the roof, windshield, and front clip of the chassis as areas with the greatest potential for redesign to improve torsional stiffness. These sensitivity results were used as a guide for modifying the baseline chassis with the goal of increased torsional stiffness with minimum increase in weight and low center-of-gravity (CG) placement.

The torsional stiffness of the Hopkins chassis with various combinations of added members in the front clip area, engine bay, roof area, front window, and the area behind the roll-cage was predicted using finite element analysis. A total of 24 different design cases were considered, culminating in a final design with a significantly increased torsional stiffness yet only a small increase in weight. Addition and relocation of structural members were positioned with adequate clearance for servicing engine and other vehicle and suspension components. Only standard size tube members were used with no modifications made to the frame rails. Twist angles of the driver and passenger side of the chassis and the rate of change in twist angle under torsion were compared for the different designs. The twist angle information showed that the transition section between the front clip and roll cage had a large gradient in deflections, indicating a flexible area of the chassis. Based on the torsional stiffness calculations, the most significant structural changes to the baseline Hopkins chassis were the addition of a V-bar structure behind the firewall, front A-bars in the front clip, and diagonal bars on the roof. Other modifications with significant impact on torsional stiffness were the addition of engine-bay bars and diagonal bars across the corners of the windshield. Another important member is the rear diagonal bar, which reduces the asymmetric twist behavior of the chassis considerably.

With strategic placement of structural members the torsional stiffness of the final design by 232% over the baseline Hopkins chassis design. The weight of the final design was W = 821 lb, an increase of only 40 lbs. The CG height of the final chassis design was only 0.5 inches above the original baseline configuration, which is insignificant relative to the overall CG height of the car with a required minimum weight of 3400 lb. The improved chassis design requires only simple modifications to the tubing of the original Hopkins chassis, and does not require changing the configuration of the frame rails. A physical 1/20 scale rapid prototype (RP) model of the final design was built to help communicate structural changes to other engineers and race team members.

The y and z rotational restraints at the rear spring perches used in this study model the constraints applied by a twist fixture used by several race teams to measure torsional stiffness [5]. Recent studies given in [6] have shown that these restraints are “over-constrained” leading to stiffness predictions which are elevated over the minimum constraint condition. For the purposes of this study, the boundary conditions were sufficient to predict relative changes between competing chassis configurations. We are currently evaluating absolute stiffness values of several competing chassis designs using “minimum-constraint” conditions determined in [6]. Results from this analysis with further optimization of the chassis for torsional stiffness and center-of-gravity will be reported in a future manuscript.

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