

Design, Analysis, and Simulation of an Automotive Carbon Fiber Monocoque Chassis

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ABSTRACT

While many composite monocoque and semi-monocoque chassis have been built there is very little open literature on how to design one. This paper considers a variety of issues related to composite monocoque design of an automotive chassis with particular emphasis on designing a Formula SAE or other race car monocoque chassis. The main deformation modes and loads considered are longitudinal torsion, local bending around mounting points, and vertical bending. The paper first considers the design of elements of an isotropic material monocoque that has satisfactory torsional, hardpoint, and vertical bending stiffness. The isotropic analysis is used to gain insight and acquire knowledge about the behavior of shells and monocoque structures when subjected to a vehicle's applied loads. The isotropic modeling is then used to set initial design targets for a full anisotropic composite analysis. The flexibility in composite layout and core design coupled with the superior material properties of carbon fiber composites is used to design and move toward an optimized monocoque composite design and layup to obtain satisfactory torsional, hardpoint and bending stiffnesses with minimal weight. Finally, some fatigue analysis considerations are outlined with emphasis on the endurance limit of the monocoque for a specific life span. The methods presented in this paper should be helpful in designing a monocoque structure chassis for FSAE race car or other applications.

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INTRODUCTION

This paper presents some rationale and techniques for the design of a composite monocoque chassis for a formula-type car. The techniques are presented based on the design of an FSAE monocoque. While the particular design is not representative of all composite chassis, the techniques and results should be of interest in most composite chassis situations.

Although carbon composite structures have great advantages in stiffness to weight they are not easy to design and build. Thus, before beginning design of a composite chassis, some key questions need to be raised. Why do we need the composite monocoque chassis? What does the composite monocoque do? How does the composite monocoque help us design a low weight, sturdy chassis? In addition, for an FSAE competition, how well will this help us win the competition?

The main purposes of the monocoque are, first, to provide rigid, safe, and sufficiently strong supports and protection to the driver, engine, and all components on a car at minimum weight. Second, the monocoque should enable the suspension to exhibit excellent handling. There are two approaches to designing a chassis to help handling, one is to make the monocoque sufficiently strong but allow it to be somewhat soft,

and design the whole chassis based on the predicted chassis deflections under various driving loads. This approach generally only works for cars with minimal or even no suspension systems, such as Go-Karts. However, the softer the chassis is, the bigger the deviations of the structural deflections get, and thus, the suspension design turns to be very difficult due to various combinations of tire camber, loading, and steer angle changes induced by varying local and overall chassis deflections. It is also very difficult to damp motions of a flexible chassis. The other approach is to design the chassis as rigid as needed to be able to design the suspension based on an essentially infinitely stiff monocoque. This is very effective for cars that rely on their suspension system for handling, and have a wide range of suspension setups. Since the suspension system has a crucial effect on the car's handling, it is usual to attempt to make the monocoque adequately stiff enough to be considered effectively infinitely stiff. [1]

VEHICLE LOADING

The monocoque being stiff for suspension performance means it is capable of absorbing all static and dynamic loads with deflections that are small compared to those occurring in the suspension. The main deformation/structural criteria for an automotive chassis are: [1]

1. Longitudinal Torsion
2. Local Bending around suspension points
3. Vertical Bending
4. Lateral Bending
5. Horizontal Lozengeing
6. Fatigue

In this paper we primarily discuss numbers 1, 2, 3 and 6 of these issues. (4 and 5 are usually not important for any reasonably rigid race car.) Also, we note that certain parts of the chassis associated with driver crash safety are often designed with strength as well as stiffness considerations. Thus for those sections the questions are how to minimize weight while maintaining sufficient strength and stiffness and how to use the stiffness of these areas to enhance overall chassis stiffness.

Longitudinal Torsion

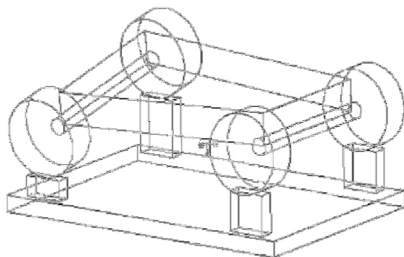


Figure 1. Longitudinal Torsion Deformation Mode

Torsion loads result from applied loads acting on opposed corners of the car. The monocoque can be thought of as a torsion spring connecting the two ends where the suspension loads act. Torsional loading and the accompanying deformation of the monocoque and suspension parts can affect the handling and performance of the car. Torsional stiffness is generally agreed to be a primary determinant of tunable chassis performance for a FSAE and most race cars as the total cornering traction and the cars dynamic stability are functions of relative front versus rear lateral weight transfer. [1]. The resistance to torsional deformation is often quoted as stiffness in Newton-meter per degree (foot-pounds per degree in US Customary units).

Historically, chassis design began with considerations of strength and safety and then progressed to stiffness and in particular, longitudinal torsional stiffness. Very early mentions of the need for torsional stiffness can be found in the 1962 book of Costin and Phipps [2] for example. Later more

analytical discussions of criteria for chassis stiffness can be found in papers by Riley and George [1] and in Deakin, et al. [3]

Local Bending Around Suspension Points

Localized bending around suspension mounting points is a result of suspension system loads. The local stiffness around the suspension mounting points has a large influence on camber change. If the suspension mounting points deflect, the attributes of the suspension change. This results in changes in camber angle, etc. and loss of overall chassis torsional stiffness. Other causes of suspension behavior changes can be from suspension component deflections and suspension joint slop. These are quite small on most purpose-built race cars [4] but should be accounted for when significant.

Vertical Bending

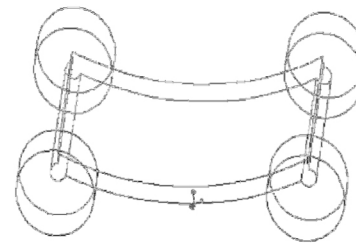


Figure 2. Vertical Bending Deformation Mode

The weight of the driver and components mounted to the monocoque, such as the engine and other parts, are carried in bending through the car chassis. Vertical accelerations can raise the magnitude of these forces. [1]

Fatigue

Components are often mounted on the monocoque by the use of inserts or "hardpoints" molded or bonded into the cellular sandwich structure. These joints are potential locations of fatigue deformation due to cyclic loading from components such as the engine and suspension mounted on the monocoque. [5]

It is generally thought that if the torsional, hardpoint and vertical bending stiffnesses are satisfactory then the structure will generally be satisfactory if its fatigue properties are also satisfactory. Fatigue is treated briefly at the end of this paper.

CHASSIS STIFFNESS TARGETS

With the loading conditions determined as discussed above, it should then be possible to design the structure to be strong enough not to fail under the global and local loads acting on it for the different load cases. Just as important, however for a racing car, is the stiffness of the entire chassis structure as it affects the proper vehicle dynamics and handling. Exactly how stiff to make the structure is difficult to determine theoretically, and historically was determined empirically based on experience gathered mostly from driver feedback. Analytically,

the problem is to examine how much of the overall vehicle compliance occurs in the structure compared to the deflections in the spring and tire. Obviously, for an infinitely rigid chassis the car will respond only to the spring, damper and anti-roll bar changes. Some stiffness approaching the infinite case, compared to suspension torsional rates, should provide a stable platform for the suspension to do its job as explained in Reference [1]. For the present paper we use an example target of about 5000 Nm/deg, based on the expected suspension rates of the vehicle.

After identifying the critical loads and setting strength and stiffness targets, the monocoque can now be designed. The approach we use is to first consider the design of a monocoque constructed of an isotropic material to guide the design of the anisotropic case. The observations and conclusions drawn from the isotropic material case provide initial insights that help in the design of the carbon fiber monocoque. A fundamental goal of chassis design is adequate strength which can be relatively easily checked for any given design. However, usually design for stiffness is the more difficult issue. The focus of this paper is primarily on stiffness and secondarily on strength.

ISOTROPIC ANALYSIS

The principal criteria in the design of a monocoque chassis are:

1. Torsional Stiffness
2. Hardpoint Stiffness
3. Bending Stiffness

Simplified Torsional Stiffness Analysis

There are very few open literature papers found on monocoque chassis design. A good overview of the problems in designing and assembling a full monocoque from individual shells was published by Weidner, et al. in 2002.[6] In the present paper we similarly start with some very simplified analyses to guide the later, more complex design of a monocoque including the best use of core and number and kind of plies and orientations on various parts of the monocoque.

A monocoque chassis can be modeled the most simply as a cylinder with a cutout as shown in [figure 3](#). The cutout roughly mimics the cockpit of a racing car. (For a road car, analogous geometric simplifications could be made.)

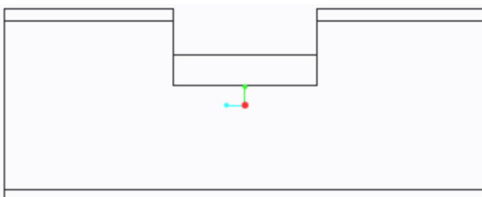


Figure 3. Cylinder with a cutout

Intuition and quick analysis in ANSYS Workbench reveals that the cut-out portion is the weakest part of the cylinder. If the cylinder is cantilevered with a torque applied at the other end as shown in [Figure 4](#), the front end of the cylinder deflects very little compared to the cut-out section. This is because the front end is a closed section tube, which performs very well in torsion. The cut-out section, however, is open and therefore deflects greatly. The cut-out section does not twist, but rather it skews. From the top view, the section deflects from a rectangle into a parallelogram ([Figure 5](#)). This means that when a torque is applied to the monocoque, the primary mode of deflection around the cockpit is not twisting but rather for the nose to deflect to one side or the other. The cockpit therefore undergoes a horizontal lozenging deformation mode.

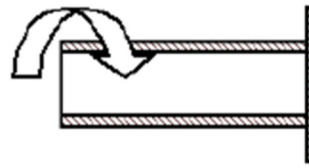


Figure 4. Cantilevered cylinder with applied torque



Figure 5. Top view of cylinder with skewed cut-out section after applied torque

Since the overall stiffness of the monocoque chassis is largely dependent on the stiffness of its weakest section, it is important to understand the mechanics of the cut-out section of the cylinder. A mathematical model to examine the torsional stiffness problem is developed as shown in [Figure 6](#):

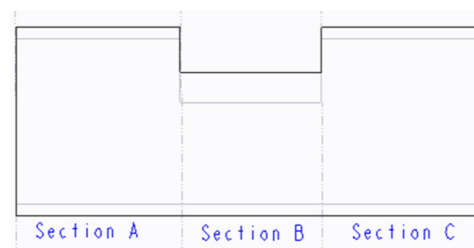


Figure 6. Cylinder divided into 3 sections of interest

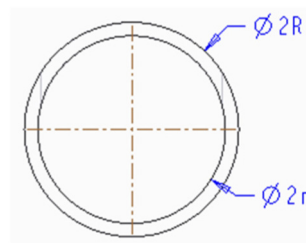


Figure 7. Cross-section of section A

The torsional stiffness of section A can be derived simply as:

$$T_A = \frac{GJ_A}{L_A} \phi \quad (1)$$

where $J_A = \frac{\pi(R^4 - r^4)}{2}$

G - Shear modulus

J_A - Polar moment of inertia of cylinder

section A

L_A - Length of section A

The torsional stiffness of section A can be written as:

$$K_T^A = \frac{GJ_A}{L_A} \quad (2)$$

The torsional stiffness of section B requires more subtle analysis. A cross-section of the cut-out section shows in a U-shape. Since this is not a circular cross-section, the twist induced by the externally applied torque is transformed into a horizontal lozenge deformation mode. [7] A quick analysis done in ANSYS Workbench reveals this is true.

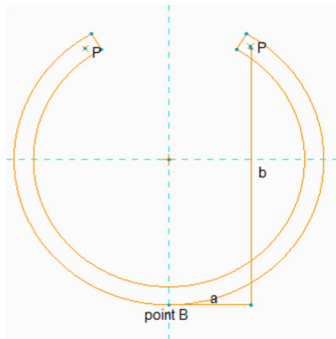


Figure 8. Cross-section of section B

From Figure 8, it is apparent that, we are twisting¹ about point B. We can see that there are 2 moment arms from the force (P) to the point B that cause the lozengeing.

$$T_B + M_B = \frac{GJ_B}{L_B} \phi - EIK \quad (3)$$

$T_B = P \cdot b$ will result in torsion

$M_B = P \cdot a$ will result in bending

To simplify the problem, we assume unconstrained lozengeing and assume that the effect of bending due to "P·a" is so small, it can be neglected. [7] Another way to think about this is that bending is along the strong axis. Therefore, it does not control the design and as such, we are designing to satisfy torsion.

$$T_B = \frac{GJ_B}{L_B} \phi \quad \text{where } J_B = \frac{L_B t^3}{3} \quad (4)$$

G - Shear modulus

J_B - Polar moment of inertia of cylinder section B

L_B - Length of section B

t - Thickness of cylinder

The torsional stiffness of section B will be given as:

$$K_T^B = \frac{GJ_B}{L_B} \quad (5)$$

Now that we understand the behavior of a cylinder (with a cutout) under torsion, we can deduce the combined torsional stiffness. The stiffness of the entire system can be found from

$$\frac{1}{K_T^{Total}} = \frac{1}{K_T^A} + \frac{1}{K_T^B} + \frac{1}{K_T^C} \quad (6)$$

Note that sections A and C have the same polar moment of inertia values. The relation between T and ϕ for both sections can be written as

$$T = \frac{\pi(R^4 - r^4)G}{2L_A} \phi$$

$$K_T^A = K_T^C = \frac{\pi(R^4 - r^4)G}{2L_A} \quad (7)$$

To find the stiffness of section B, we need to write P in terms of T. If we analyze the internal forces that result from the externally applied torque T, we find that, P is T/2R.

1. Depending on whether horizontal lozengeing of section B is constrained or not, we can have a significant amount of longitudinal bending in addition to twist.

$$P \cdot b = \frac{L(R-r)^3 G}{3L_B} \phi$$
$$\frac{T}{2R} \cdot b = \frac{L_B(R-r)^3 G}{3L_B} \phi$$
$$T = \frac{2L_B(R-r)^3 GR}{3L_B b} \phi$$
$$K_T^B = \frac{2R(R-r)^3 G}{3b}$$

(8)

Next, we validate the approximate analytical model with a simulation in ANSYS Workbench. The results are shown in Table 1.

Table 1. Analytical versus ANSYS 56 6789: values in SI units

Input Parameters	Output Parameter	
T = 50Nm G = 80GPa R = 0.10m r = 0.090m b = 0.19m L _A = L _B = 0.13m	Analytical Approximation (MNm/deg)	ANSYS Workbench (MNm/deg)
	K _T ^{Total} = 0.028	K _T ^{Total} = 0.031

The ANSYS result is within 11% of the analytical result. The reason for the difference between the two results is that, in our analytical solution, we assume unconstrained lozenging. Unconstrained lozenging means that, the cut-out section (section B) warps in isolation (In other words, sections A and C are assumed to be absent).

The main task in the design of the monocoque chassis for torsional stiffness is then to find a weight-efficient way to stiffen the cockpit region. The expression for K_T^B gives us an idea of how to vary the various parameters to increase the stiffness of the cockpit region.

Cockpit Stiffening Techniques

One common method of stiffening the opening is to use a lip or beam around the opening. We tried various other stiffening techniques in ANSYS Workbench and compared the specific stiffness (stiffness to weight ratio) of the resulting designs. We included some unconventional stiffening techniques in these simulations.

Table 2. Legend for the stiffening techniques

Representation	Stiffening Technique
1	Monocoque with cut-out
2	Monocoque with cut-out and 0.013m stiffener plate on the bottom of cut-out region
3	Monocoque with cut-out and 0.019m stiffener plate on the bottom of cut-out region
4	Monocoque with cut-out and 0.025m stiffener plate on the bottom of cut-out region
5	Monocoque with elliptical cut-out (Minor axis is along the length of the monocoque)
6	Monocoque with very small cut-out
7	Monocoque with no cut-out

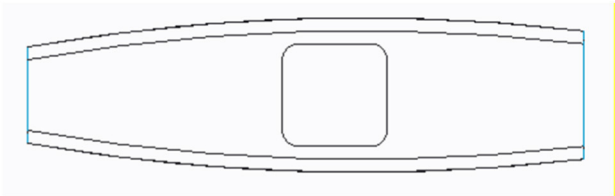


Figure 9. Monocoque with cut-out



Figure 10. Monocoque with cut-out and 0.013 m stiffener plate on the bottom of cut-out region



Figure 11. Monocoque with cut-out and 0.019 m stiffener plate on the bottom of cut-out region



Figure 12. Monocoque with cut-out and 0.025 m stiffener plate on the bottom of cut-out region

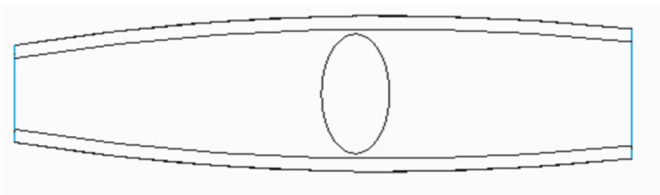


Figure 13. Monocoque with elliptical cut-out (Minor axis is along the length of the monocoque)

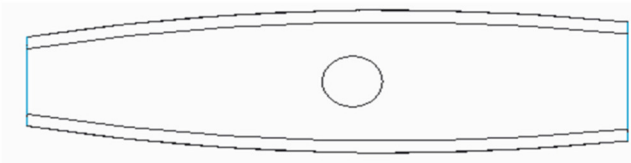


Figure 14. Monocoque with very small cut-out



Figure 15. Monocoque with no cut-out

The basis of the stiffening techniques is to vary the parameters in the K_T^B expression ($K_T^B = \frac{2R(R-r)^3G}{3b}$) and subsequently determine the percentage increase in the torsional stiffness of the overall monocoque. The technique denoted as 1 is the monocoque in its unstiffened form. Techniques 2, 3 and 4 are based on increasing "R-r" or the thickness of the bottom of the cut-out section. In techniques 5 and 6, the effect of constrained lozengeing on the stiffness of the cut-out section is investigated.

Our K_T^B expression, as mentioned earlier, is based on the assumption that lozengeing is unconstrained. In actuality, lozengeing is constrained and techniques 5 and 6 introduce different forms of constraining lozengeing and determining the increase in overall monocoque torsional stiffness.

Technique 7 is a closed section tube with no cut-out and therefore, performs very well in torsion. Technique 7 gives the target torsional stiffness value. Based on our K_T^B expression, of course there are other stiffening techniques such as increasing the outer diameter of the monocoque or changing the monocoque material from steel to another material of higher shear modulus that were not considered in our analysis.

The results from the stiffening analysis are presented in Figures 18 and 19.

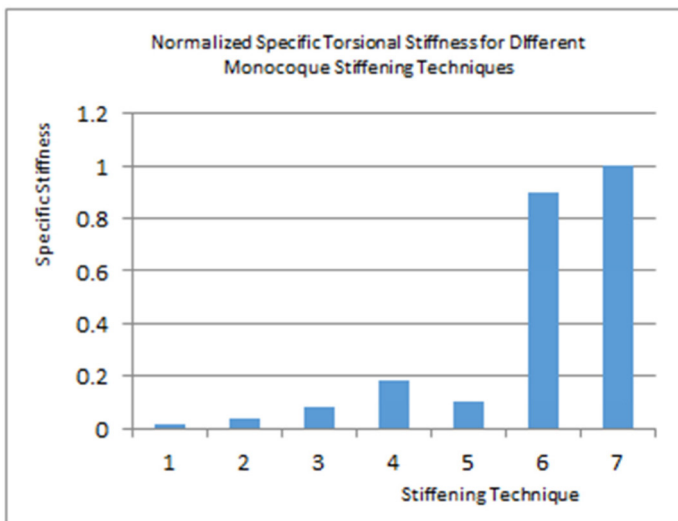


Figure 16. Normalized specific torsional stiffness of different stiffening techniques

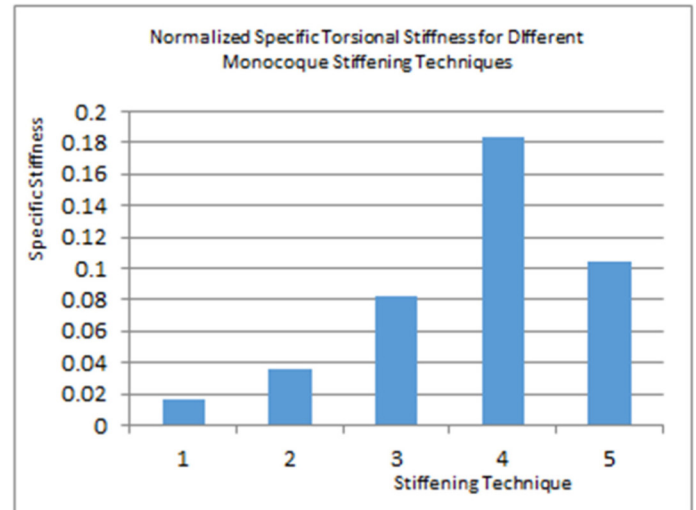


Figure 17. Normalized specific torsional stiffness of different stiffening techniques

To efficiently show the results and to make them general to any vehicle we normalized the values by the length of the monocoque. The final point on the graph represents the specific stiffness of a unit length monocoque without a cut-out (target specific stiffness). The second graph captures the first five stiffening techniques in detail. From the graphs, we can deduce that, increasing the thickness of the cut-out region increases the specific torsional stiffness of the monocoque. This means, the increase in stiffness due to the increase in thickness outweighs the added weight due to the increase in thickness. A monocoque with a small cut-out (1/5 times the size of the cut-out of technique 1) has a specific stiffness value equal to 90% of the target specific stiffness value.

These graphs should provide a general idea of what a monocoque designer should expect, in terms of stiffness versus weight savings, when implementing any of the example stiffening techniques.

Stiffness Analysis of Hardpoint Locations

Points where the suspension system, engine, etc. are attached to the frame or monocoque (hardpoints) need to have optimum local stiffness and strength. In some chassis designs such as tube frames or semi-monocoques the suspension and other loads can be taken at bulkheads, longerons, internal beams, or where other local internal structure can stiffen the attachments. This is a very desirable situation where practical. Even for a full monocoque chassis some loads can be taken at front or rear bulkheads or at engine mounting or roll bar locations.

However, in most Formula cars the desire for a narrow chassis plus the need for room for the driver means that some suspension attachments will likely be made to the monocoque itself. These may be either local inserts added into or built into the core or local geometry modifications such as tapering the core to zero thickness and locally adding extra composite

layers. As mentioned earlier, local deflection of suspension mounting points may result in changes in camber, steer angle, etc. and loss of overall torsional stiffness.

As an example we analyze loads from lower suspension links (which carry the heaviest loads for most suspension systems) that are mounted on a transition from the sides of the monocoque to the bottom of the monocoque. This transition can either be a bevel/chamfer or a radius (See [figure 18](#)). The attachment points experience extremely high loads on the order of 20 kN for a 2G bump for a Cornell's 2011 FSAE car for example. Ignoring the angles that the suspension loads make with the monocoque at the points of attachment, we simplify the above problem to a point load normal to a plate (chamfer or bevel transition) or shell (radius transition).

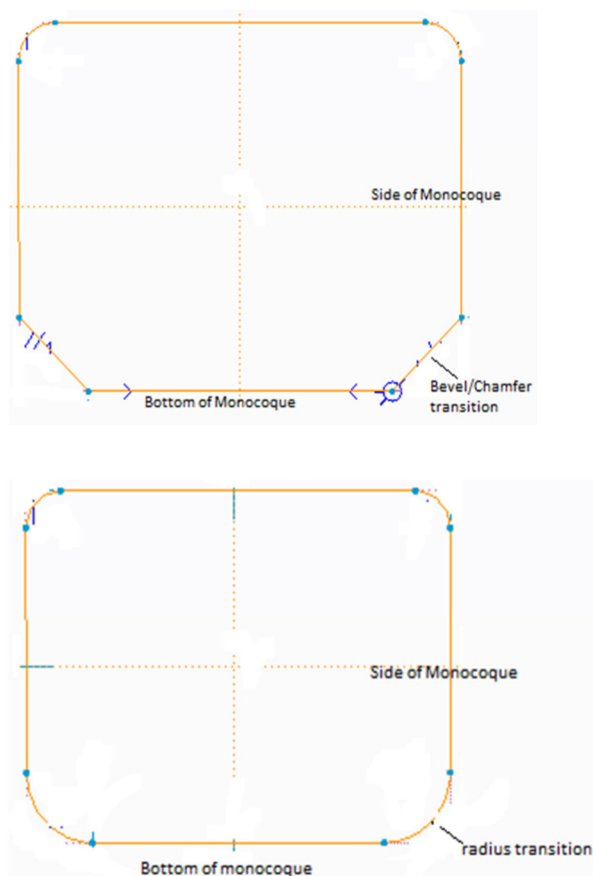


Figure 18. Front view of monocoque showing chamfer and radius transitions.

Plate Analysis

First we analyze an attachment on a local flat section (chamfer). Since the thickness of a plate is small in comparison with other dimensions, certain simplifying assumptions can be introduced which reduce plate problems to two-dimensional rather than three-dimensional analysis. The well-known Kirchhoff hypothesis (Classical Plate Theory) is an example. It assumes that lines normal to a plate's middle surface before deformation remain normal after deformation. [8]

Another less restrictive assumption asserts that displacements vary linearly over the plate thickness. All of these assumptions are kinematic in nature; they impose no restrictions on the order of magnitude of the strains in the plane of the plate or on the order of magnitude of the displacements. [9]

When a thin flexible plate is subjected to transverse loads, it displaces normal to its middle plane and forms a curved surface. If the transverse displacements are small in comparison with the thickness of the plate, the strains in the middle surface are usually small and negligible in comparison with those developed in the extreme fibers. [9] If the plate undergoes large transverse displacements, however, significant strains may be developed in the deformed middle plane. Mathematical descriptions of these phenomena involve highly nonlinear partial differential equations for the transverse displacements. Few exact solutions to these equations are available in the literature, [9] and, in the case of plates with irregular shapes and boundary conditions, exact solutions are practically difficult. Because of this difficulty in obtaining exact solutions, numerical methods such as the finite element method are often employed to obtain quantitative solutions to these problems.

We therefore ran numerous simulations in ANSYS Workbench to get a general sense of the behavior of plates subjected to point loads. We varied the plate parameters in a controlled way and performed analysis of variance in order to be able to make useful and accurate generalizations about the out-of-plane behavior of plates. To assist in the design of hardpoints on zero curvature chamfer panels, graphs are given to aid the initial decision making process.

To be able to use the graphs, it is important to understand the model set-up and how the controlling parameters were varied.

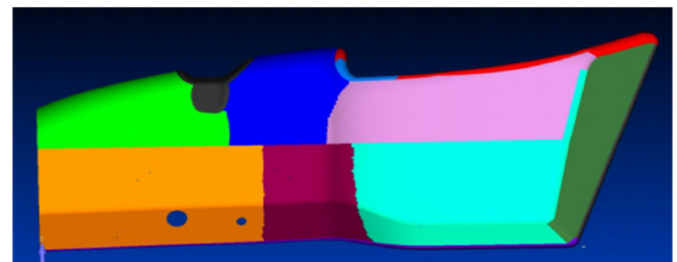


Figure 19. Chamfer Transition - Load applications show by blue dots

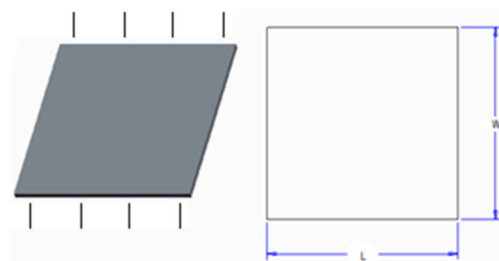


Figure 20. Plate Model for Chamfer Transition

We modeled the suspension load on a locally flat chamfer (See figure 20) by setting up a plate, fixed on two sides and free on the other two sides, with a point load acting on the center of the plate. The fixed sides are assumed to approximate the greatly increased stiffness at the fold angles at the edge of the chamfer. The parameter 'W' represents the width of the chamfer. 'L' is the length of the fixed edge of the chamfer along the monocoque. We kept the distance 'L' constant and increased 'W' for a particular plate thickness. The results are presented in Figures 21, 22, 23, 24.

The plots are stiffness-per-equivalent stress (stiffness/von Mises stress) for different chamfer width and thickness. To make the results general to any monocoque design, we normalized all the values with the length L of the monocoque. Since plates typically have a thickness-to-width ratio of approximately 0.1 or less, the final point on the graph for the "0.009 thickness plot" represents the maximum stiffness-per-equivalent-stress value for a plate. To use the plots, cross reference the chamfer width with the chamfer thickness and read off the stiffness-per-equivalent-stress. For example if we desire a stiffness-per-equivalent-stress that is 80% of the maximum value, we have the option of choosing a chamfer with a thickness that is 0.009 times the length of the monocoque and a width that is 0.036 times the length of the monocoque. The individual "thickness plots" are shown in Figures 23, 24, 25.

In general, for a particular thickness, the hardpoint stiffness-per-equivalent stress decreases as the width of the chamfer increases. To put it simply, as the width of the chamfer increases, the equivalent stress increases and the stiffness also decreases significantly. Therefore, the stiffness-per-stress decreases. For a particular chamfer thickness, the designer should opt for smaller chamfer widths. How small the width of the chamfer should be will depend on the suspension attachment sizes and options available to the designer.

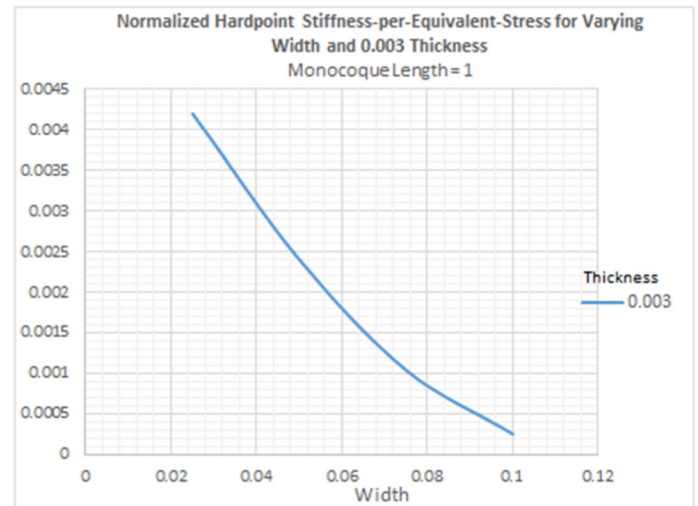


Figure 22. Normalized Chamfer transition stiffness-to-equivalent-stress for varying chamfer width and 0.003 thickness

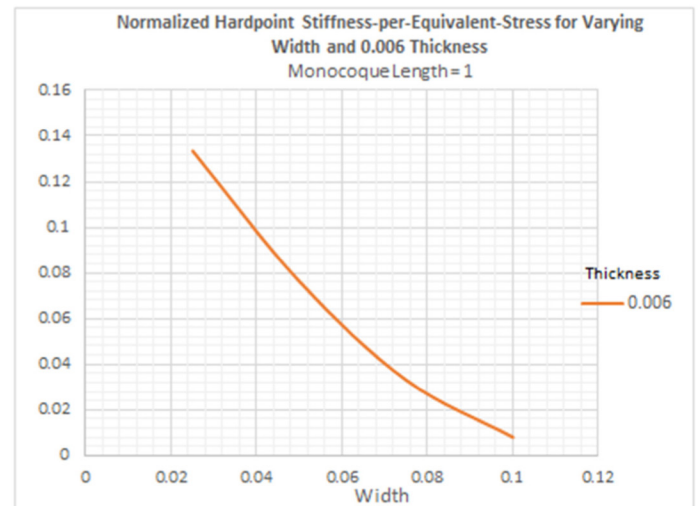


Figure 23. Normalized Chamfer transition stiffness-to-equivalent-stress for varying chamfer width and 0.006 thicknesses

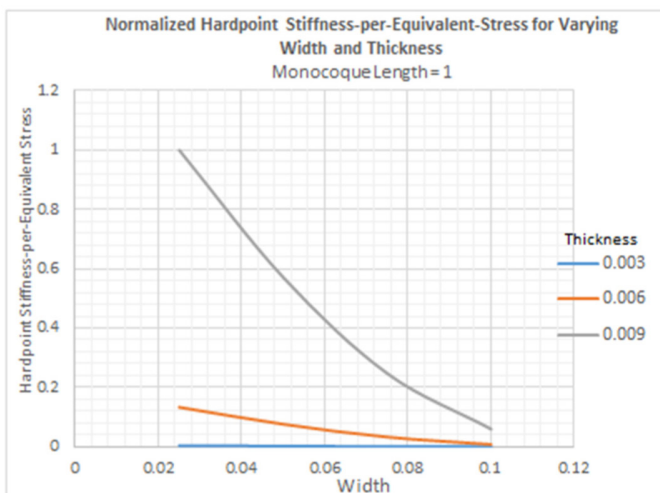


Figure 21. Normalized Chamfer transition stiffness-to-equivalent-stress for varying chamfer width and thickness

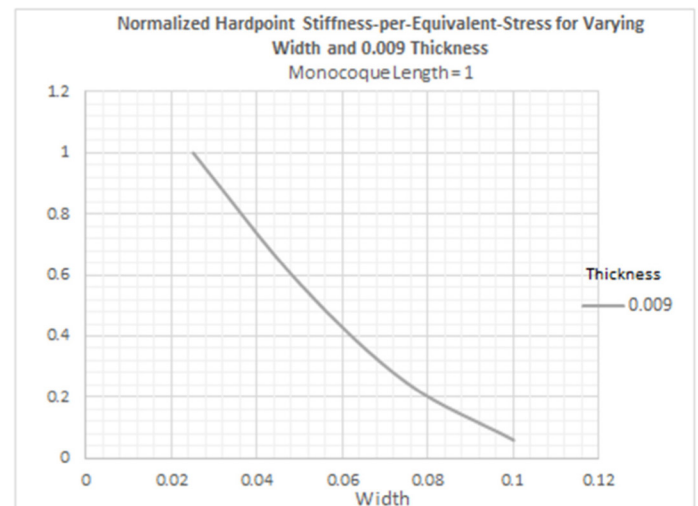


Figure 24. Normalized Chamfer transition stiffness-to-equivalent-stress for varying chamfer width and 0.009 thicknesses