Engineering Report May, 2003

LIGHTWEIGHT**SUV**FRAME

DESIGN DEVELOPMENT

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EXECUTIVE**SUMMARY**

Background

The Lightweight SUV Frame project is a research initiative of the Department of Energy and the Auto/Steel Partnership (A/SP), designed and engineered by Altair Engineering, Troy, Mich., with the key objective to reduce the baseline frame structure's mass by 25%.

This was to be accomplished through:

- development of an efficient steel SUV frame architecture
- application of High- and Advanced High-Strength steels and related manufacturing technologies
- maintenance of baseline structural performance.

This Phase I study focused mainly on frame system performance and mass reduction. Cost considerations were not a driving factor for this Phase I design effort, however cost was monitored throughout.

Analyses and advanced techniques for applying material, architecture and design were used to develop a non-traditional lightweight frame (Figure A). This new frame is 23% lighter than the baseline frame, while maintaining the baseline structural performance. This was achieved for a minimal cost increase of 31¢ per pound saved. Click **Lightweight Frame Concept** to view a a slide show highlighting the steps followed to design this concept.

Lightweight Frame Concept

- 23% Reduced Mass
- 50% Reduced Weld Mass
- 50% Reduced Weld Length
- Cost increase of just 31¢/lb. (68¢/kg) saved
- Maintains or improves performance
- Aggressive use of HSS or AHSS steels
- Streamlined, pioneering design

The new frame features aggressive use of High-Strength (HSS) and Advanced High-Strength (AHSS) steels, increasing usage by 59% over the baseline.

The Lightweight Frame Concept can replace the baseline frame (Ford Expedition/Lincoln Navigator ladder-style frame) without major assembly and packaging issues.

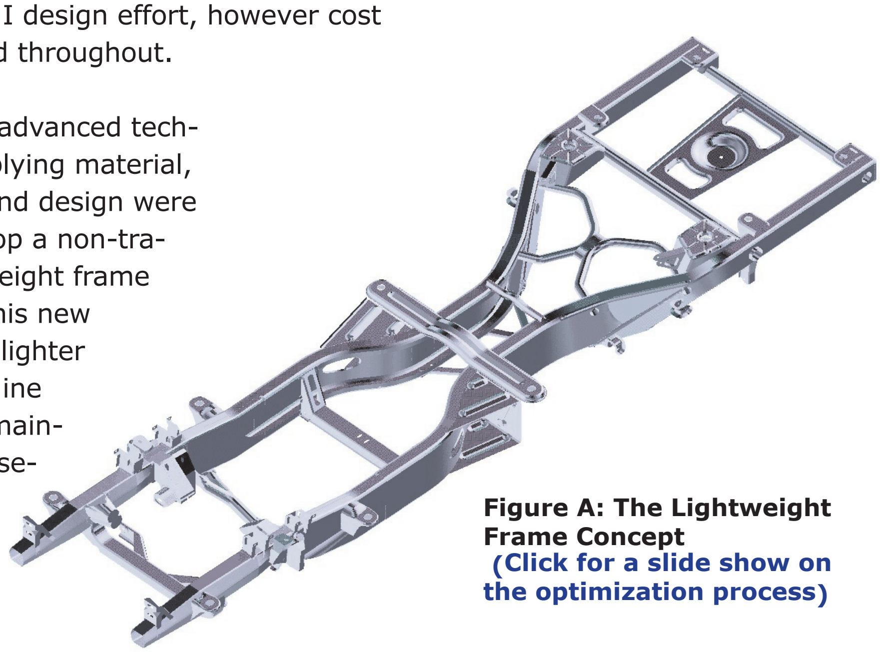


Figure A: The Lightweight Frame Concept
(Click for a slide show on the optimization process)

Altair's design engineering team used newly introduced tools in the field of concept design development that combine a complete packaging study with topology optimization analytical methods to formulate an efficient concept structure. This approach leads the design engineer to consider the design's most effective load paths, not just those from historical or competitive designs. This methodology has often shown radical departure from an incumbent design philosophy, producing a significantly more efficient design.

The topology exercise creates successive concept design iterations until an optimal balance of mass, architectural efficiencies and structural performance is reached.

The results of this project can be used as a guideline for engineers to develop lightweight structures.

The Lightweight Frame Concept, at this design phase, shows promising results for significant mass savings potential and quality structural performance. It is an excellent foundation for a Phase II design investigation. Phase II is a necessary next step to evaluate vehicle level performance, such as:

- Crash Management (Offset Barrier, Side Impact, Rear Impact)
- Body Mount Stiffness

Through Phase I and further phases of the program, the Department of Energy and the A/SP seek to deliver valuable research to assist OEMs in streamlining vehicle mass with new, pioneering steel vehicle structures. ❑

1.0 PROGRAM**INTRODUCTION**

Frame-based platforms are expected to maintain their position as the standard for large SUVs and pickups. This is mainly due to the economic benefits related to the scalability of the frame to accommodate various vehicle configurations. While improvements have been made, the basic architecture of the large SUV and truck frame has not changed much in the past 25 years. Due to the dependence of multiple overlapping platforms for a particular OEM on the configuration of a frame, it is difficult to make a departure from the traditional design philosophy without some design guidelines and a demonstration of the weight reduction and efficiencies that can be achieved.

The objective of this Lightweight SUV Frame Project, an initiative of the Department of Energy and the Auto/Steel Partnership, is to design a lightweight frame with the ultimate goal of achieving 25% weight reduction from baseline, while maintaining baseline structural performance. This would be achieved by applying state-of-the-art concept design and analytical methods, as well as advanced steel materials and manufacturing techniques. Cost considerations were not a driving factor for this Phase I design effort; however, it was monitored throughout.

The project deliverables include an optimized frame concept for a production SUV, known as the Lightweight Frame Concept (Figure 1). Also included in the deliverables is an electronic report documenting the process and conclusions of the first phase. This report may be used as a guideline by a frame engineer to develop improved frame designs with significantly reduced mass.

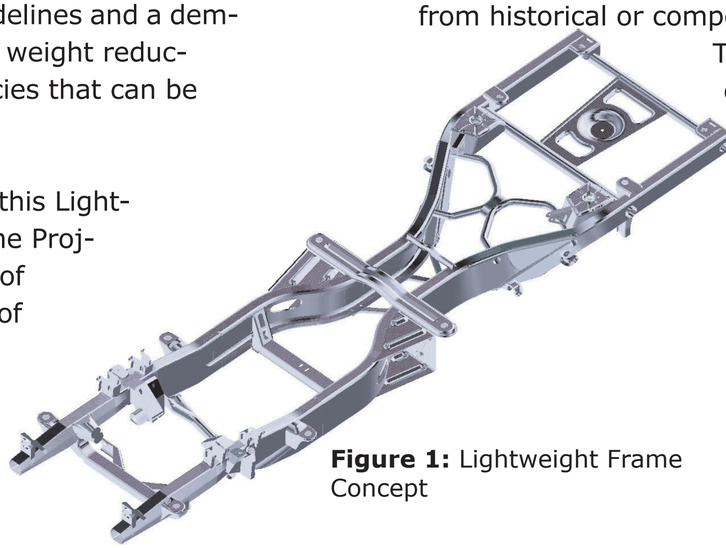


Figure 1: Lightweight Frame Concept

Altair's design engineering team used newly introduced tools in the field of concept design development. The tools combine a complete packaging study with topology optimization analytical methods to formulate an efficient concept structure. This approach leads the designer to consider the most effective load paths for the design, not just those from historical or competitive designs. This methodology has often shown radical departure from an incumbent design philosophy, producing a significantly more efficient design.

In tandem with the topology optimization, a rigorous review of baseline and imminent advanced manufacturing and materials technologies was applied to the new architecture to develop the most cost- and mass-effective design approach for the frame subsystems. High-Strength (HSS) and Advanced High-Strength (AHSS) steels were considered and applied for their proven lightweighting capabilities.

Before commencing the design process, a technology review was conducted to evaluate the baseline frame and to identify design architecture, material and manufacturing mass saving strategies that could be implemented to fulfill program objectives. ❑

2.0 TECHNOLOGY**REVIEW**

To achieve the aggressive 25% SUV frame weight reduction, it was imperative to adopt non-traditional materials, design and manufacturing. An industry technology review was conducted to determine strategies to reduce frame weight without sacrificing performance/durability. Appendix A lists the strategies that resulted from this review. A broad range of current and emerging technologies, all relevant to frame architecture, was investigated in order to assess mass savings potential.

During the process of designing a new lightweight frame, it was essential to consider the following domains to identify technologies that might help reduce mass:

- Materials
- Manufacturing
- Design Architecture

These technologies are often overlooked, primarily due to relatively high cost solutions, but they can provide interesting options for the frame design. Based on the technology review conducted for this program, mass reduction strategies were developed and applied to the design engineering effort. A summary of the technology review and the resultant strategies follows in succeeding sections.

2.1 Materials

The sheet steel industry has made significant advancements in the refinement and production of the steel materials in use today. In earlier years, automotive structures primarily used low-carbon steels. In the recent years, there has been a slow trend to the increasing use of High-Strength Steels such as HSLA, microalloy and bake hardenable. Today, an even broader range of steel materials is emerging, including several different types of Advanced High-Strength Steels (AHSS), including:

- Martensitic (Mart)
- Dual Phase (DP)
- Transformation Induced Plasticity (TRIP)
- Complex Phase (CP)

Compared to High-Strength Steels (HSS), AHSS offer high strength with better formability, greater energy absorption, post-form strengthening capabilities, and high strength-to-weight ratios. As reported by the American Iron and Steel Institute (AISI)[1], the principal differences between conventional HSS and AHSS are their microstructures. AHSS are multiphase steels, which contain martensite, bainite, and/or retained austenite in quantities sufficient to produce unique mechanical properties. Among other types of steels investigated were:

- Metallic (zinc) coated steels (replace hot-dipped wax)
- Boron steels (combined with heat treatment)
- Sandwich Material (steel/plastic/steel)
- Metal foam (to reinforce joints)

HSS and AHSS application was deemed the most effective and relevant material technology to achieve the SUV frame mass saving target. With the understanding that the architecture of the frame may deviate from the baseline's traditional ladder frame, the utilization of HSS and AHSS would prove beneficial in improving performance as well.

2.2 Manufacturing

As with the sheet steel industry, the manufacturing sector has made significant advancements in forming complex parts with minimal defects, producing parts with minimal scrap, and improving strength, stiffness and weight savings. Some of these advancements include:

- Hydroforming
- Tailor-Welded Blanks
- Tailor-Welded Tubes
- Butt-Weld (e.g., Rail Sections)
- Conical Tubes
- Roll Forming
- Extrusion
- Patch Technology
- Metal Foam
- Hot Stamping

The utilization of hydrofoming reduces mass by eliminating the need to overlap material for conventional welding methods. Incorporating tailor-welded blank technology with hydroforming could further assist in the frame mass reduction.

2.3 Design Architecture

The improvements accomplished in the materials and manufacturing sectors could be combined with design modifications to achieve effective mass reduction. Determining the proper frame configuration, or architecture, may greatly influence its performance. Based on improvements made through material and manufacturing technologies, the frame design could be modified to influence global structural performance. To achieve the structural performance goals of the project, it is advantageous to investigate the connection between the cross-members and the rails (i.e. joint interface) and to look at means of optimizing the frame structure. These goals are detailed following:

2.3.1 Joint Stiffness

Joint stiffness performance is critical to overall frame integrity. Therefore, it is important to maximize the stiffness of each joint. In order to better understand joint stiffness versus mass efficiency, the A/SP sponsored the Light Truck Frame Joint Stiffness Study[2]. The study produced a "Joint Stiffness Toolbox" that allows engineers to maximize the stiffness of many common frame joints. Of the joints studied, a round tube intersecting a rectangular tube, welded on both sides, provided the best overall stiffness-to-weight ratio. Consequently this type of joint may be applied, as appropriate, to the concept frame. To learn more about the Joint Stiffness Study, see Appendix B.

2.3.2 Optimization

Other equally important modifications that could greatly influence the overall frame performance while reducing mass are:

- Optimizing load paths
- Increasing section sizes
- Using more cross-members
- Incorporating lightening holes
- Down-gauging

Successive optimization analyses would be performed to determine the optimal architecture, gauges, section sizes and shapes. Finite element-based structural analyses and optimization software tools would be used to design and optimize structures. There are two types of optimization analyses, as described below:

Topology Optimization

Topology optimization generates an optimal material distribution for a set of loads and constraints within a given design space.

The design space can be defined using shell elements, solid elements, or both. In the classical topology optimization setup, global loads and boundary conditions are applied to acquire the load paths (i.e., optimal design structure) by solving the minimum compliance problem. Manufacturing constraints can also be imposed using minimum member size and draw direction constraints.

Gauge and Shape Optimization

General gauge and shape optimization problems can be solved by assigning variables to control the model. These variables include the shape, height and width of the frame members. Variables can also be assigned to properties which control the thickness, area, moments of inertia, stiffness, and non-structural mass of frame members.

Both of these methods were used throughout the project. The methodology used to apply these optimization techniques is detailed below. ❑

3.0 BASELINE**FRAME**

The A/SP Lightweight SUV Frame Project Team, which is made up of A/SP members, Altair Engineering and Oxford Automotive (see Page ii for a list of team members) selected the 1997-2002 Ford Expedition/ Lincoln Navigator for proof of concept. A/SP member representatives of Ford Motor Company supplied the necessary data to proceed with the program. The Expedition/ Navigator frame (Figure 2) consists of rails made with stamped open C-sections and stamped cross-members. The front third of each rail is boxed with a second stamped C-section. These architectural components resemble a ladder, with the straight stamped cross-members representing the rungs. The thickness of the individual frame components varies between 2.5 mm and 5.0 mm. The frame is mainly made from components stamped from low-carbon steel. Please refer to Appendix E for a complete component listing including weights and materials.

A series of analyses was conducted on the baseline frame, monitoring stiffness, natural frequencies and peak stress. This data was used to establish performance targets for the concept frame. Please refer to Sections 5.9 and 5.10 for more information.

The Project Team desired to look outside the conventional ladder frame design. Therefore, all of their ideas and suggestions were investigated to fully explore the feasibility of a lightweight chassis in alternate designs. A brainstorming session was held to identify and assess benefits and shortcomings of the baseline ladder frame and determine weight reduction strategies. The output from the brainstorming meeting is documented in Appendix A. An advantages/ disadvantages summary (Table 1) is the result of that exercise.

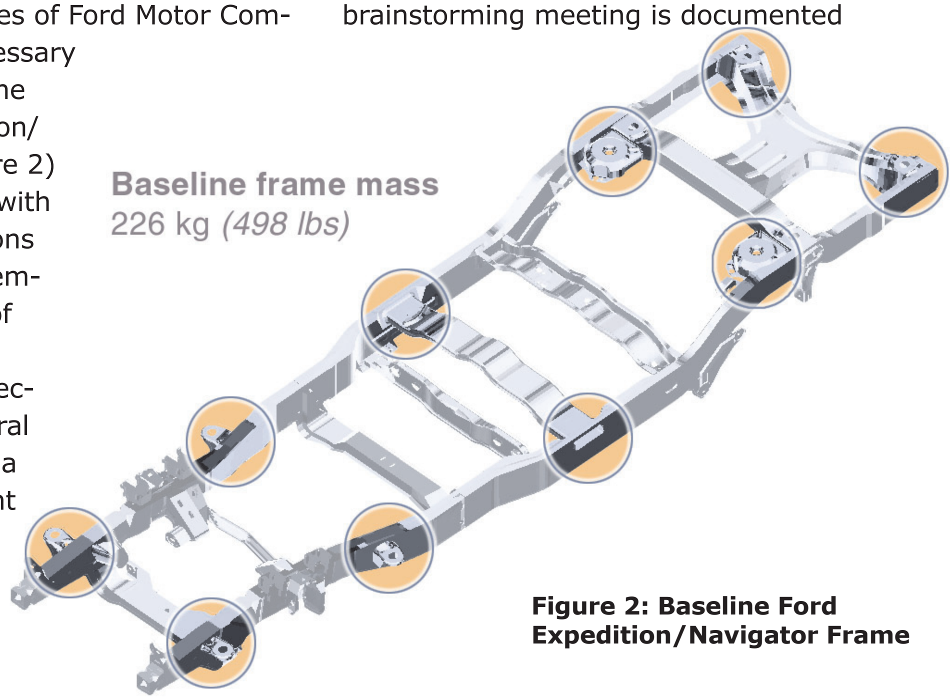


Figure 2: Baseline Ford Expedition/Navigator Frame

As the design engineering process ensued, these points concerning the baseline frame were used as guides towards developing an optimal frame design.

The new concept frame was intended to replace the baseline frame without major assembly and packaging issues (see Appendix G for the packaging comparison images). Therefore, the engine and suspension brackets were treated as carry-over components, since altering their design would result in

Table 1: Baseline Frame Advantages/Disadvantages

Advantages	Disadvantages
Packaging, particularly propshaft, fuel storage	Weight
Proven Safety	Dimensional Control
Existing Manufacturing Infrastructure	Number of Pieces
Flexibility for Additional Wheelbase	
Cost	
Vehicle Service (lift)	
Parking Brake Mounting	

complex suspension and powertrain packaging issues. Design changes were considered for the rest of the frame without altering the location of the fixed points (body, powertrain, and suspension mount locations, circled in Figure 2). The only major components that would require modification in the new frame configuration would be the fuel tank and exhaust systems. ❑

4.0 DESIGN**METHODOLOGY**

The purpose of this section is to give a basic knowledge of the methodology that will be used in assisting the Project Team to develop a lightweight frame structure. The image below is a generic visual aid to help understand the major steps in the optimization process. First, the design space is determined by creating a usable volume in which the part can be included (A: Design Space / Topology Optimization). A 2D or 3D Topology optimization is performed on this design space based on certain constraints and loading conditions.

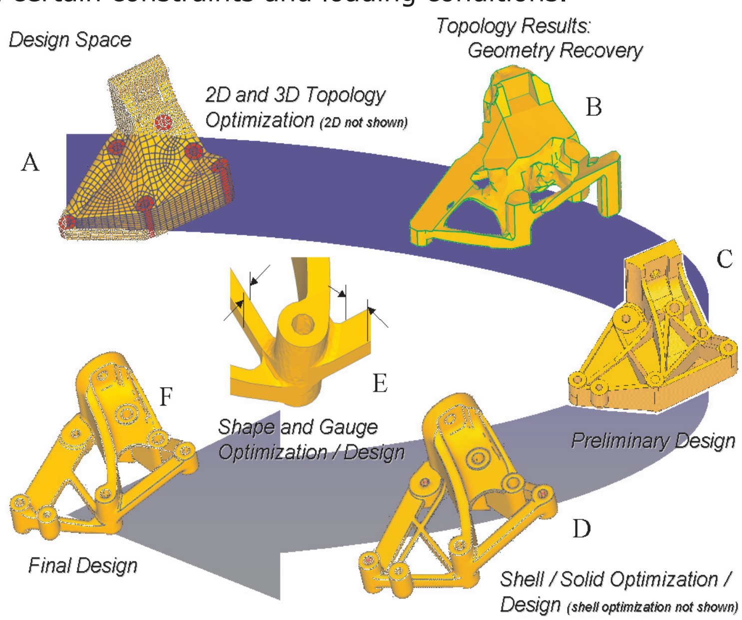


Figure 3: Generic Illustration of Design Process

Note: Although not shown in Figure 3, some structures can accommodate a preliminary 2-Dimensional feasibility study, which gives quick direction on the design configuration with a minimum amount of input data. The nature of this SUV frame structure lent itself to this 2-Dimensional study, and consequently it was the first action taken in the design process. Details of the 2D study for this project are in Section 5.1.

The resulting geometry is an initial rough concept that reveals where structure should be located (B: Geometry Recovery).

After the topology optimization is performed, the results must be interpreted per the materials and manufacturing processes that are chosen. A preliminary design is established (C: Preliminary Design), with the goal of maintaining, as closely as possible, the optimal material distribution derived from the optimization.

Once this design is established, a finite element (FE) model is created using shell or solid elements on which additional optimization analysis is performed. This analysis determines the locations and shape of the components as well as the location of the lightening holes. The results reveal a more refined load path and lighter structure (D: Shell / Solid Optimization and Design).

The optimization analysis results are interpreted in a new design, which is then prepared for gauge and shape optimization. This step in the methodology (E: Shape and Gauge Optimization and Design) varies the components' geometric variables (i.e., heights, widths, and gauges) to satisfy the performance targets.

Finally, the results of the gauge and shape optimization are interpreted by design engineers who consider all manufacturing and cost issues in order to develop a final design (F: Final Design). ❑

5.0 LIGHTWEIGHT SUV FRAME DESIGN DEVELOPMENT

The technology review previously mentioned identified the direction that Altair followed to design the new lightweight frame. The optimization analyses, as described in Section 4.0, Design Methodology, were conducted with OptiStruct®[4], in order to determine the structure with the best mass saving while maintaining the performance targets. These performance targets were established by evaluating baseline frame global stiffness (bending stiffness and torsional stiffness), modal response, and peak stress. The following sections review the optimization and performance evaluations that led to a final lightweight frame design.

5.1 2-Dimensional Topology Optimization

A preliminary 2-dimensional feasibility study was conducted to obtain a potential direction to pursue in order to achieve the mass and performance targets.

The package space for the frame topology optimization was initially represented in HyperMesh®[5], with a continuous layer of shell elements stretching along the tops of the baseline rails, from the first cross-member rearward. Load cases relevant to bending and torsion, along with the appropriate boundary conditions, were applied to the 2D model for optimization (see Appendix C for the details of the loading and boundary conditions). The model used for the topology optimization is shown in Figure 4.

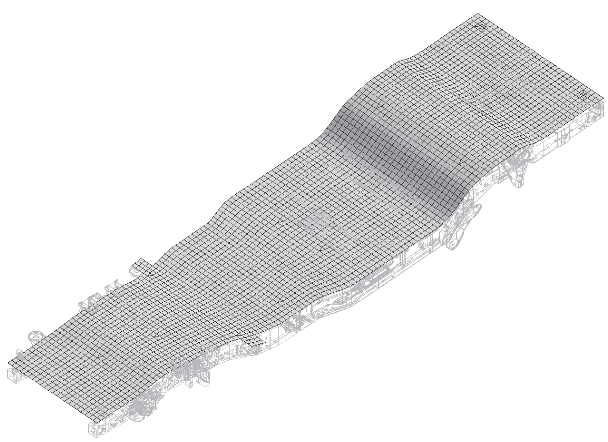


Figure 4: Two-Dimensional Model

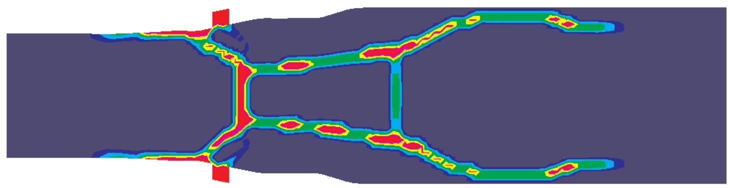


Figure 5: Density Plot of Topology Optimization Results

The 2D topology optimization revealed a preliminary architectural layout, Figure 5, indicating areas where material would be best placed, as shown by the density plot. The results also reveal the locations of cross-members, indicated by the bands stretching across the width of the model.

The results of the 2D topology show a non-traditional architectural layout, which indicates that the methodology chosen to conduct this study will lend itself to explore other avenues of frame design, without limiting the design to a typical ladder frame. The 3-dimensional topology optimization that was performed next further defined the frame layout.

5.2 3-Dimensional Design Space Definition

After the preliminary 2D feasibility study, the analysts continued to the next step of the process, 3D optimization. This step begins by defining the available package space that will be used for the frame design.

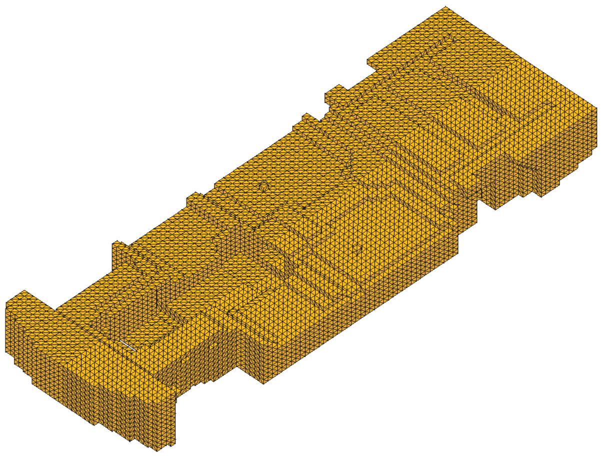


Figure 6: Three-Dimensional Design Space

CAD information about the baseline frame, suspension, attachments, powertrain and ground clearance was used to define the 3-dimensional volume package space for the 3D topology optimization to determine the frame's optimized load paths, or optimal design structure. The design space, shown in Figure 6, encompassed the maximum allowable volume, represented by a solid mesh. The volumes for the spare tire, suspension components, wheels, body mounts, and powertrain components were removed from the available package. The Project Team agreed to ignore the fuel and exhaust systems based on the assumption that these systems could be redesigned to accommodate the optimized frame.

5.3 3-Dimensional Topology Optimization of Design Space

A 3D topology optimization was performed on the design space to determine the load paths. The objective of the topology optimization was to minimize the mass of the structure under the constraints as defined by meeting or exceeding the following baseline targets:

- bending stiffness
- torsional stiffness

The stiffness of body mounts was not a target of the study but was monitored throughout the design effort. However, it should be investigated further in the next phase of the program. To review the boundary conditions, refer to Appendix C.

The results for the 3D topology optimization are shown in Figures 7 and 8. They may also

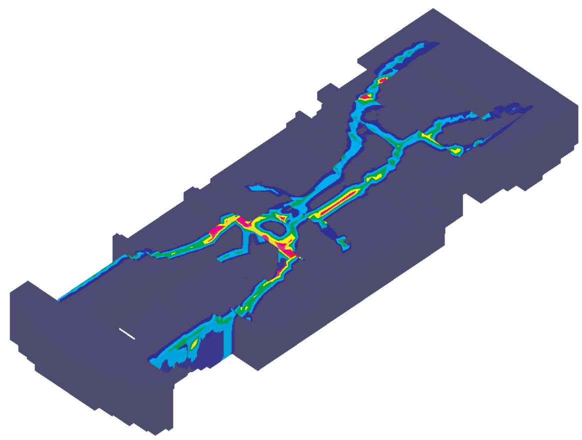


Figure 7: Topology Optimization Results: Material Distribution In Package Space

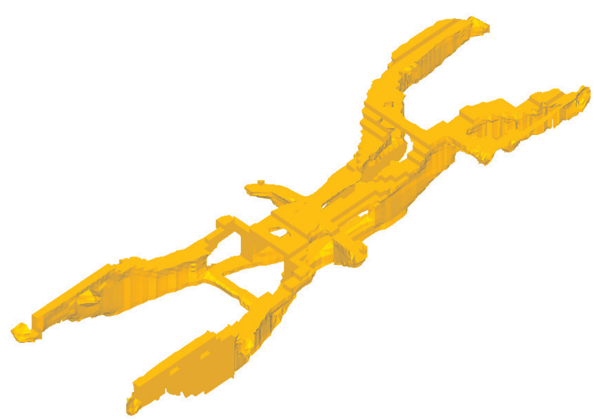


Figure 8: Geometry Recovery Topology Optimization Results: Load Paths

be seen in a HyperView3® player, which can be viewed by clicking this link: **3D topology optimization***. The rough hourglass-shaped structure (Figure 8) indicates where material is needed in order to meet performance requirements.

The results of this analysis do not indicate the type of cross-section or the thickness of the components, only the placement of material.

*Note: If you have not already installed a HyperView® player, the application is available on this Electronic Report CD.

5.4 Interpretation and Preliminary Design

The 3D topology results indicate the structural load paths for the frame and clearly show the location of the rails and cross-members. After completion of the topology optimization, the Altair design and engineering team interpreted the results. They applied

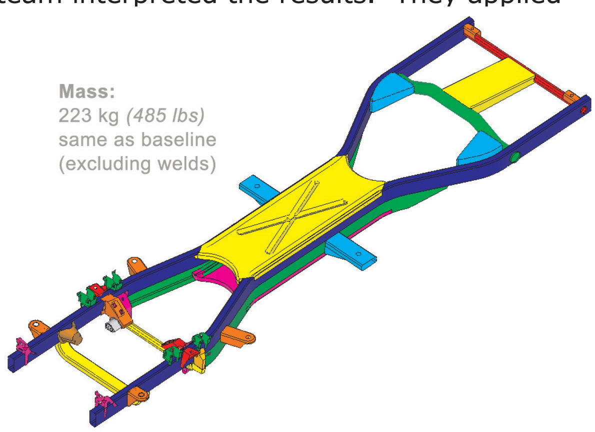


Figure 9a: Design Interpretation 1

the methods that were identified in the Technology Review phase of this project when developing the preliminary design. The team utilized basic design rules for the selected manufacturing method to develop a design that best represents the OptiStruct results.

While utilizing these guidelines, it became apparent that we would not be able to meet

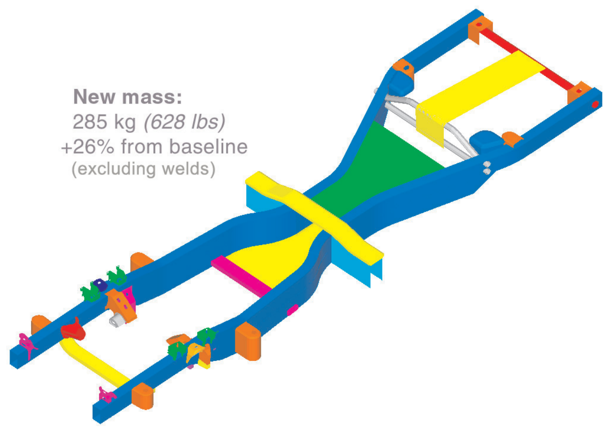


Figure 9b: Design Interpretation 2 —Preliminary Design

the OptiStruct results exactly, so the design and engineering team discussed which load paths were most important and which areas could be revised without negatively impacting the design. The design was then completed to best meet the structural and manufacturing objectives.

The analysts replaced the load path structure with components whose cross-sections, gauges and dimensions were interpreted based on the analysts' experience and the topology optimization results shown in Figures 7 and 8.

The location and number of cross-members, as well as vertical and horizontal rail alignment, were also interpreted by analysts from the topology optimization results of the 3D package space, while the baseline frame fixed points (body, suspension, and power train mounts) were maintained.

The analysts' interpretation of the topology results led to several concept designs and the two most promising designs are shown in Figures 9a and 9b. The initial thicknesses used for these designs were carried over from the baseline frame. The structural performance and frame mass of both designs were evaluated. Based on the results of the analysis, it was determined that the design shown in

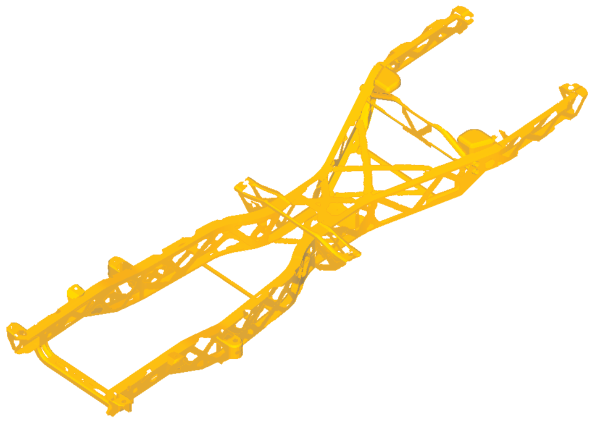


Figure10a: Shell Topology Results Concept 1

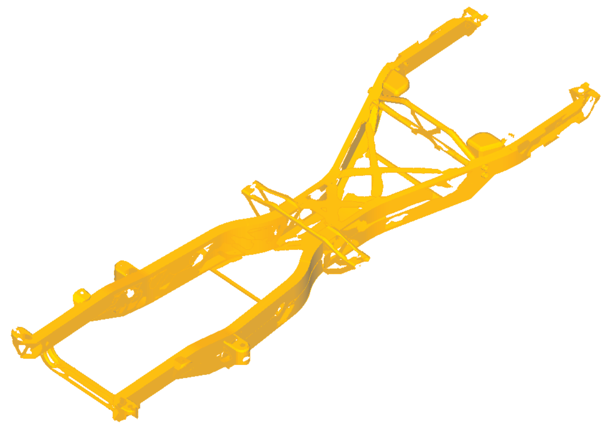


Figure10b: Shell Topology Results Concept 2
(Click image for HyperView®)

Figure 9b was the preferred starting point for the next stage of the optimization process. This design was modeled with 2D shells and this was the model used for the shell topology optimization.

5.5 Shell Topology Optimization

The results of the shell topology optimization effectively identified the areas where mass can be removed, based on the load paths for global stiffness loading conditions. This creates a more accurate representation of the optimized structure. Cross-members, or truss-like structures, will emerge, giving the frame more realistic attributes. This can be clearly seen by the voids in the rails shown in Figures 10a and 10b.

Two concepts were developed through the shell topology optimization. Concept 1, Figure 10a, appears promising from a mass reduction standpoint, but extensive crash analysis should be conducted in order to determine if this concept could meet structural performance. However, Phase I of this program did not allow that design concept to be pursued. According to the topology results, Concept 2, Figure 10b, offers a better chance to achieve the structural performance requirements, within the limits of this phase of the program, and be factored into a design that could be analyzed for structural performance.

5.6 Shell Topology Design

The new design took into account the manufacturing considerations from the technology review. In the initial design (Figure 9b), the rails were represented with one continuous closed section. When interpreting the shell topology optimization results, again using manufacturing and assembly guidelines, the analysts favored replacing the center rail portion with an open C-section, while maintaining a closed section for the front and the rear. Based on this interpretation, the front and rear portion could accommodate hydroformed components. Also, the solid center section of material initially defined in the analysis was removed from the center section plate, which is now represented by obvious X-shaped cross-members.

Design interpretation 3, however, needed to be fine-tuned using the gauge and shape optimization process in order to further reduce the frame mass.

The final shape and dimensions of the cross-sections will be determined by the gauge and shape optimization.

Figure 11: Shell Topology Results

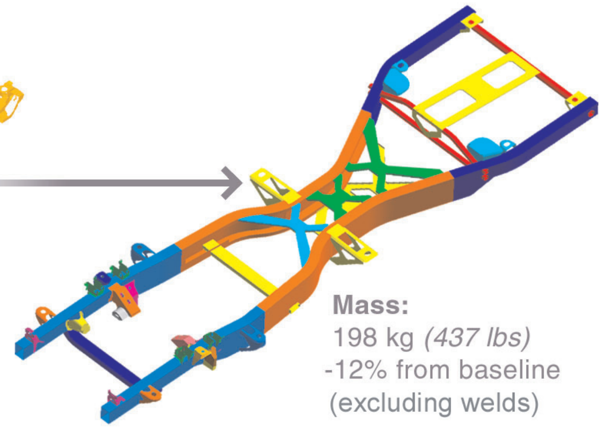


Figure 12 : Design Interpretation 3

5.7 Gauge and Shape Optimization

In the prior shell topology step, the resulting design interpretation (Design Interpretation 3, Figure 12) had 12% less mass than the baseline frame. To this point, the design process has applied the baseline frame component thicknesses to each design topology optimization. The objective of the next step in the design methodology, the gauge and shape optimization, is to squeeze more weight out of the frame by optimizing component material thickness and shape. The analysis is set up by minimizing the frame weight and its design variations under certain displacement, frequency and stress constraints. The design interpretation resulting from the shell optimization, Figure 12, was the starting point for the gauge and shape optimization.

Objective

A range of geometric variables (i.e., heights, widths, and material gauges) were applied to minimize the weight of the frame while maintaining the stiffness performance targets.

Constraints

The constraints imposed in the gauge and shape optimization process were the same as the constraints imposed in the topology optimization. That is, they should meet or exceed baseline frame performance with respect to bending and torsional stiffness.

Design Variables

In order to achieve an optimal shape or gauge, design variables must be defined for specific components. The design variables were defined as:

- Thickness of all frame components - lower boundary of 1.0 mm, upper boundary of 4.0 mm
- Dimensions (heights and widths) of the rail and cross-member cross-sections - the height and width could both vary in a 50-mm range.

Numerous combinations of component material thicknesses and dimensions will be assessed until the optimal gauge and shape was achieved for the least mass and best performance. Clicking Figure 13 will display an animation that graphically illustrates how the evaluation of these design variables might look.

Figure 13: Design Variables Evaluation (Click to View Animation)

To optimize mass, thicker material should be applied only where it is needed for structural performance. Therefore, the rails and some of the cross-members were subdivided as shown in Figure 14 so that the thickness could be varied along their lengths during optimization. While suggesting a better mass distribution, this subdivision also would provide direction as to preferred material thickness variation along the length of the rail. These subdivisions could be consolidated into three or fewer divisions of uniform thickness as typically found in ladder type frames. Later, as the structure is evaluated for performance, technologies such as tailor welded blanks, local reinforcements or doublers of suitable thickness were investigated to accommodate the need for extra thickness in a particular region, as identified in the optimization.

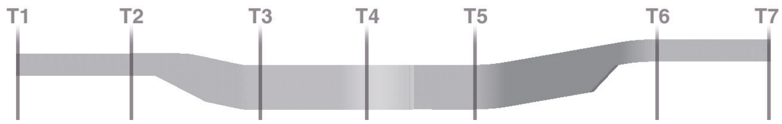


Figure 14: Rail Material Thickness Subdivisions

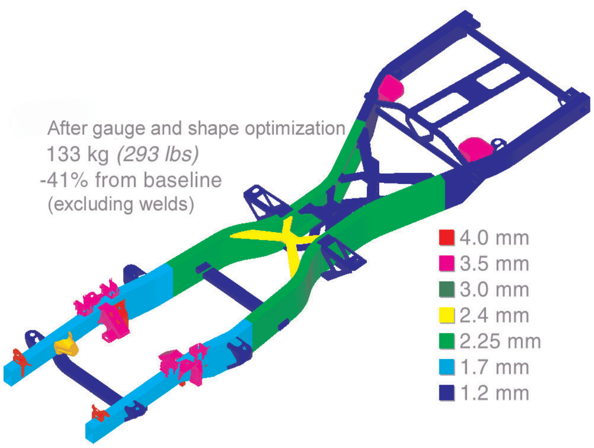


Figure 15: Design after Gauge and Shape Optimization

Gauge and Shape Optimization Results

The gauge and shape optimization results, Figure 15, showed that changing component thickness was more beneficial than changing the rail and cross-member dimensions. This is primarily due to the fact that the 3D shell topology had already closely predicted the optimal rail and cross-member dimensions.

The baseline rails are made of two overlapping C-sections with thicknesses of 2.5 mm and 3.0 mm. The gauge optimization analysis showed that the rails could be subdivided into three portions with thinner gauges: 1.2 mm for the rear portion, 2.25 mm for the center portion and 1.7 mm for the front. Several cross-members also were down gauged from 2.0 mm to 1.2 mm.

5.8 Lightweight Frame Concept

Up to this stage in the design methodology, all optimization results were factored into the design by analysts whose primary focus was on optimizing frame performance and material distribution rather than addressing manufacturing issues. Figure 15 shows the design as interpreted through this process.

The Lightweight Frame Concept seen in Figure 16 was derived by design engineers who interpreted all analyses results and manufacturing considerations, as well as their own experience. These interpretations caused the Lightweight Frame Concept to increase in weight. However, it is still 23% lighter then the baseline frame. The interpretation sometimes results in drastic design changes like that found in the cross-member area forward of the rear rails. The more complex cross-member structures first interpreted through the optimization process were replaced with a simpler tube structure, which is easier and less costly to manufacture.

As mentioned previously, the Light Truck Frame Joint Stiffness Study indicated that a round tube intersecting a rectangular tube, welded on

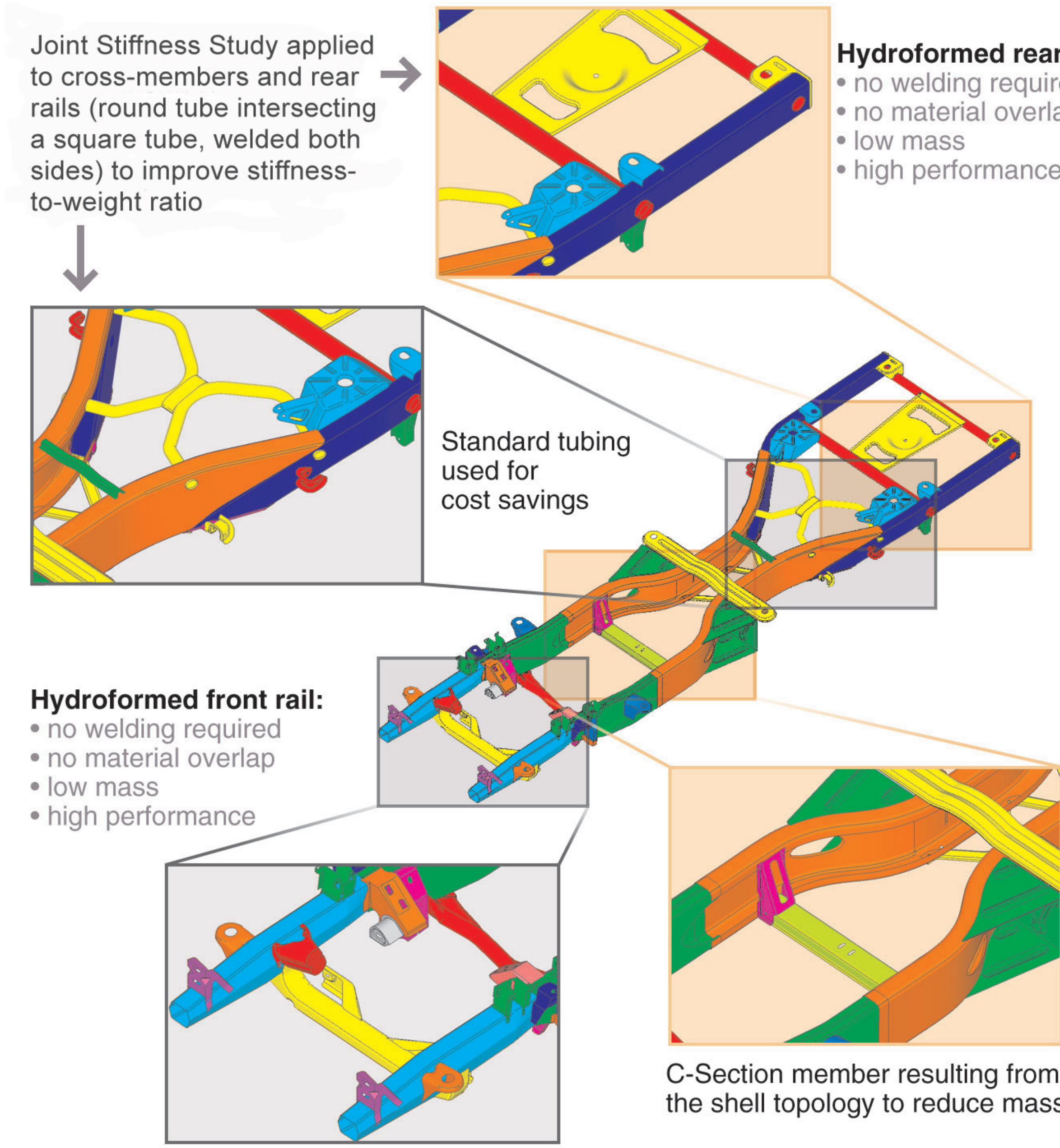


Figure 16: Lightweight Frame Concept (-23% lighter than Baseline frame)

both sides, provided the best overall stiffness-to-weight ratio. Therefore, this type of joint was utilized in most of the permanent cross-members, while maximizing the available cross-sections.

The shell topology analysis favored a closed front and rear rail section, which was represented by hydroformed tubes. By utilizing a hydroformed tube design, the overall mass of the rail section was reduced by eliminating the overlapping C-sections found in the original rail structures. Because of this change, the design engineers needed to ensure that the hydroformed rail design would maintain the baseline crash characteristics with regard to peak load and energy absorption. Consequently a separate study, detailed in Section 5.10, was conducted. It determined the final shape, thickness and material requirements to develop a lighter

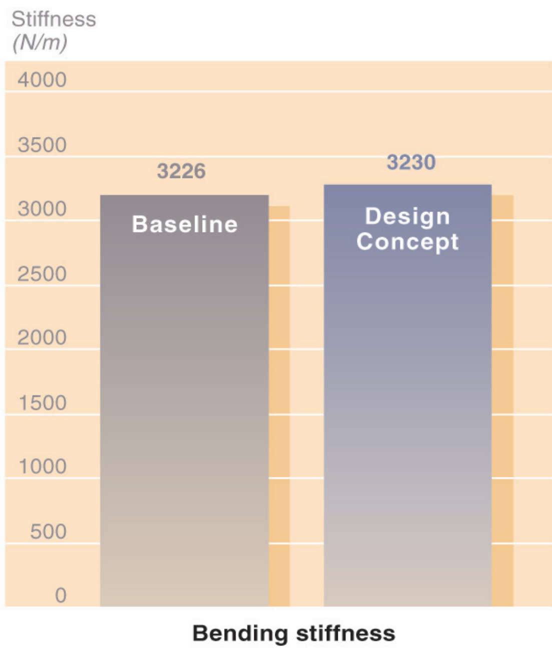


Figure 17: Bending Stiffness Results

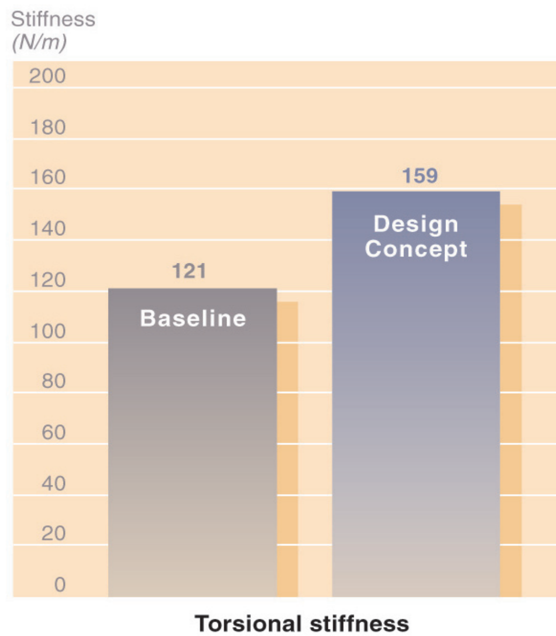


Figure 18: Torsional Stiffness Results

front rail section, while maintaining the same crash performance as the baseline rail.

At this point, the frame assembly Bill of Materials (BOM) was not finalized. In order to finalize the BOM, the peak stress analysis must be completed. See Section 5.9.4 for a discussion on material selection.

Because the Lightweight Frame Concept differs significantly from the post-shell topology design, which had successfully met performance targets, a new structural performance evaluation was required to ensure that baseline performance requirements were met.

5.9 Performance Evaluation

As mentioned previously, structural performance was judged by the bending and torsional stiffness results, modal responses and peak stress analysis. A summary of this evaluation follows.

5.9.1 Bending and Torsional Stiffness

The model was set up employing the simple-support method, wherein support structure does not contribute to frame stiffness test values. To review boundary conditions and load cases applied to both the baseline and proposed frame, please see Appendix C.

Figure 17 illustrates that the proposed frame design bending stiffness results of 3230 N/m meets baseline performance. Torsional stiffness results, graphed in Figure 18, show that the proposed frame design achieves 159 kNm/rad, improving performance by 31% over the baseline frame. This increase in performance is primarily due to the unique design of the frame.

5.9.2 Modal Responses

Modal responses were evaluated for natural frequencies of the frame, which are related to noise, vibration, and harshness (NVH) of the vehicle. Figure 19 shows that the optimized frame design exceeds the baseline twist mode performance by 34% (25.0 Hz) and exceeds the vertical bending mode by 3% (27.8 Hz). Twist and vertical bending modes have the most significance for ride and handling. Lateral bending is a mode monitored for evaluation of the overall frame performance. The Project Team agreed to monitor lateral bending mode performance, but it was not used as a target for this design phase. Lateral bending would be further investigated in a detailed design study. To review 3D animations of each modal response case, click **Twist**, **Vertical Bending** or **Lateral Bending.**

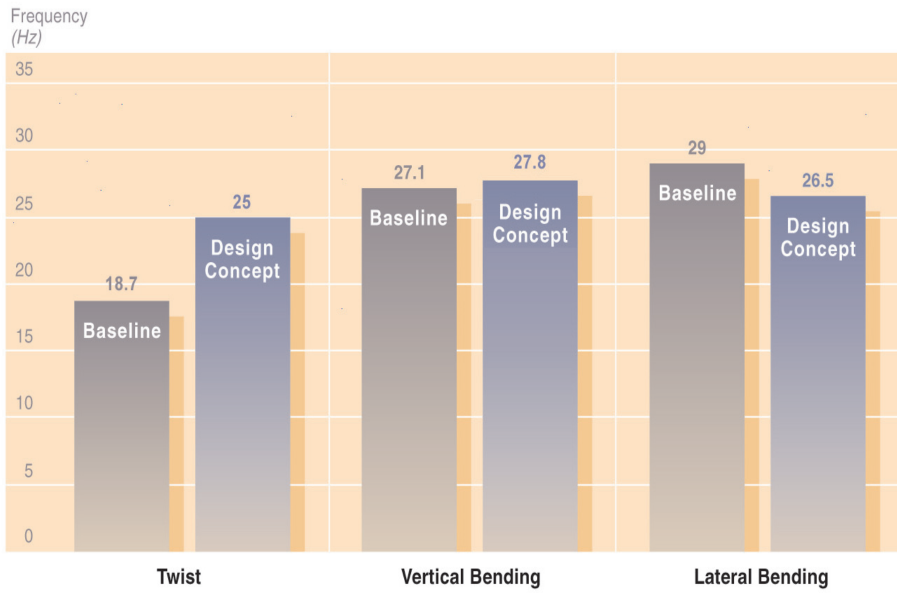


Figure 19: Modal Response Results

5.9.3 Peak Stress Analysis

The peak stress analysis was performed to determine the stress levels on the suspension brackets, body mounts and frame under three generic suspension loading conditions: 2G Twist, 3-2-1G Standing, and 3G Vertical, for both the front and rear suspensions.

5.9.4 Material Selection

The Project Team created a material matrix that the analysts would be able to choose from when assigning material grades to the frame (see Appendix D).

The materials selected for the Lightweight Concept Frame are shown in Appendix E. Parts 28-37, 39-41 and 46-48 were carried over from the baseline frame. Thus, the materials for these parts were also carried over from the baseline frame. The material selection process for the remainder of the parts commenced by considering peak stress. Figure 20 shows the stress contour in the frame under 3G Vertical loading. The close-up view of the body mount #1 and front rail details the stress distribution in that area. The stress distribution for each component, which is affected by the suspension load cases, is carefully reviewed. The area of each color has a stress range which is then compared with the material matrix list, and a selection is made based on the material yield strength.

For example, the results of the analysis show a stress range in body mount #1 (BODY MNT #1, Part ID 37 in the table located in Appendix E) between 270 MPa (39 ksi) and 320 MPa (46 ksi). Based on these stress values, an ESA-M1A35-C material grade would be selected. However, the body mount is a carry-over component and therefore its steel grade remains the same as in the baseline frame (i.e. ESA-M1A35-C). This shows that the methodology used to assign material grades to the frame’s components is valid.

This methodology was therefore used for the material selection of the remainder of the non-carry-over components. The results of the analysis show a stress range in the rail (S/M-Frame Frt RH, Part ID 1 in the table) between 370 MPa (54 ksi) and 475 MPa (69 ksi). In this case, an HSLA 420/480 was selected. This methodology was applied for each load case, which facilitated selection of the most appropriate steel grade for the other suspension components of the frame. However, the stress contours for these components are not documented. The material selection and the peak stress results are documented in Appendix E.

It should be noted that the material documented in Appendix E is only preliminary. Only after further studies, which would include structural and formability analysis along with physical testing, can the material grades be finalized. Phase II of this project would address the above issues.

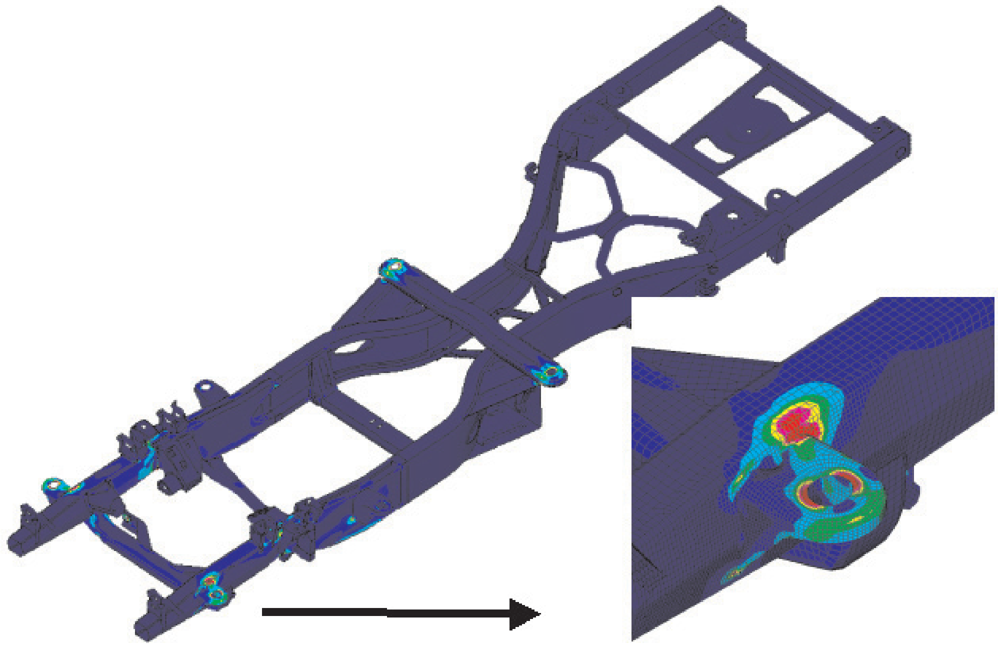


Figure 20: Stress Contour, 3G Vertical Loading

The Project Team reviewed this material selection for formability and crashworthiness considerations and made a change in material for some parts. Finally the Project Team, for corrosion protection reasons, agreed that all steel less than 2.0 mm in thickness should have a zinc coating. Thus, the Project Team selected an appropriate zinc coated sheet for all such parts.

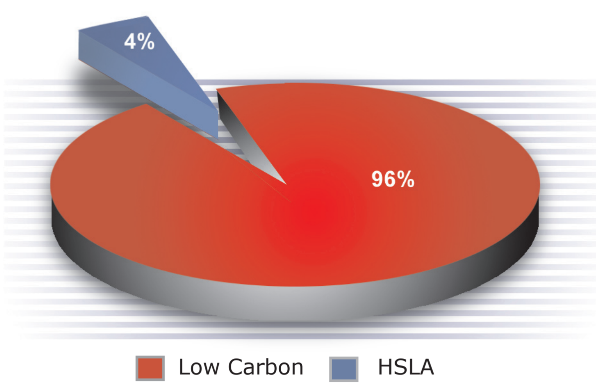


Figure 20a: Material Repartition in Baseline Frame

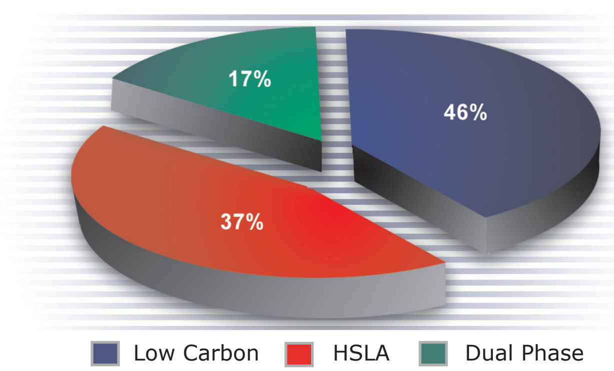


Figure 20b: Material Repartition in Lightweight Frame

Figures 20a and 20b show the repartition between low carbon steel and HSLA in the baseline frame and the proposed frame, resulting from the material selection described above.

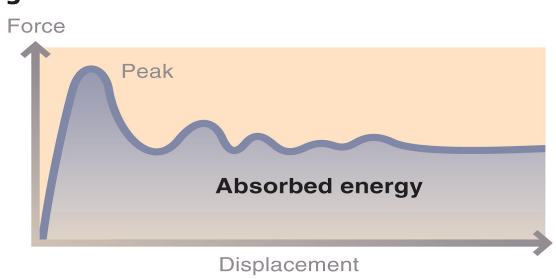
5.10 Rail Crush Study

A side-study was conducted with the objective to replace the baseline front rail with a lighter front rail while maintaining the baseline peak load and energy absorption. The baseline characteristics were monitored with a focus on matching the load curve by minimizing the difference in absorbed energy. Figures 21a and 21b show examples of how this monitoring process would be plotted. Actual results of the rail crush pulse comparison conducted for this study are shown in Section 5.10.2.

5.10.1 Baseline Frame Front Rail Crush Analysis

The baseline front rail section performance was evaluated through a rail crush analysis, using LS-Dyna 3D[6]. Only the front section of the frame was modeled. The front-end structure consists of the front rails, first cross-member, shock tower assembly and miscellaneous brackets. The front rails were truncated 250 mm from the center of the first cross-member and constrained. Because of the design symmetry, a front-end structure half-model was analyzed. The finite element model of the front section of the frame is shown in Figure 22.

Figure 21a: Crash Event Characteristics



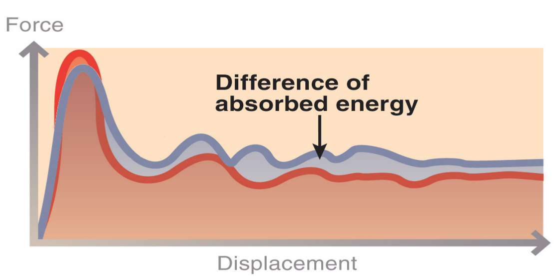


Figure 21b: Difference in Absorbed Energy

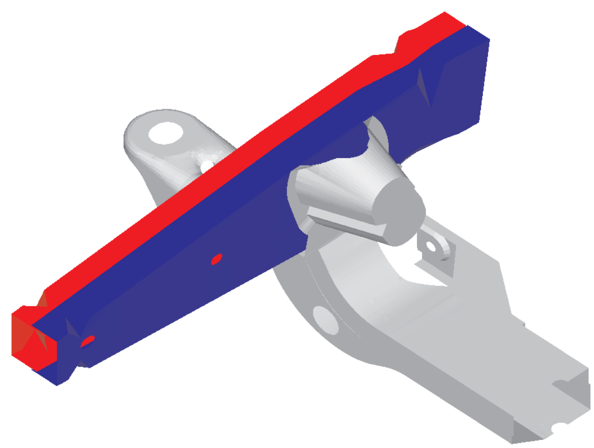


Figure 22: Rail Crush Crash Simulation Set Up

The Gross Vehicle Weight Rating (GVWR) for the 4x2 frame model, approximately 2606 kg (5746 lbs), was specified for the rail crush simulation barrier as a half model weight of 1303 kg (2873 lbs). The barrier was constrained in space in all but the axial, or x direction, which was defined at a 35 mph (56.3 km/h) initial velocity. The front-end structure of the frame was fixed in space at the rear as shown in Figure 23. Symmetry boundary conditions were specified at the center of the first cross-member.

The front portions of the baseline rails include crush initiators, which are mainly used to produce a progressive crush and ensure that crush zones begin at the desired location, while limiting peak force. The front rail sections are not uniform, but tapered, as shown in Figure 24.

Each of the rails is constructed of two stamped C-sections (Figure 25) made of 36 ksi low-carbon steel and seam welded at the top and bottom. The inboard C-section thickness (shown in blue) is 2.5 mm and the outboard C-section (shown in red) is 3.0 mm.

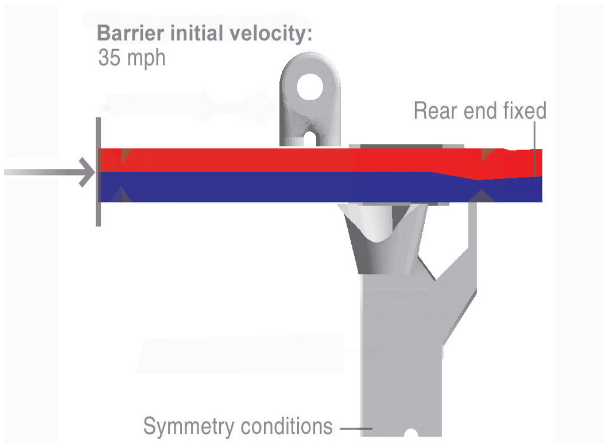


Figure 23: Front Rail Crush FEA Model

The front rail deformed shape, barrier intrusion, force-displacement curves, and the rail energy absorbed were all monitored for comparison purposes, as well as for investigation of gauge reduction and shape optimization.

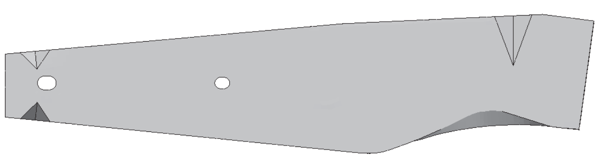


Figure 24: Baseline Front Rail Front Section View

The results of the baseline rail crush analysis (Figure 26) show a peak load of 175 kN and an average load of 90 kN.

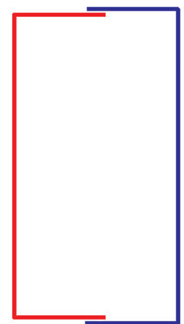


Figure 25: Baseline Front Rail Cross-Section View

5.10.2 Lightweight Rail Development

A Design of Experiment (DoE) analysis was performed to determine the combination of the parameters listed below that achieves maximum mass reduction in the front rails, while maintaining the baseline front rail crush performance. A DoE allows for inputs of numerous design variables having to do with material hardening curves and thickness, component shapes, heights and widths, applied in numerous configurations until the optimal combination of these is achieved.

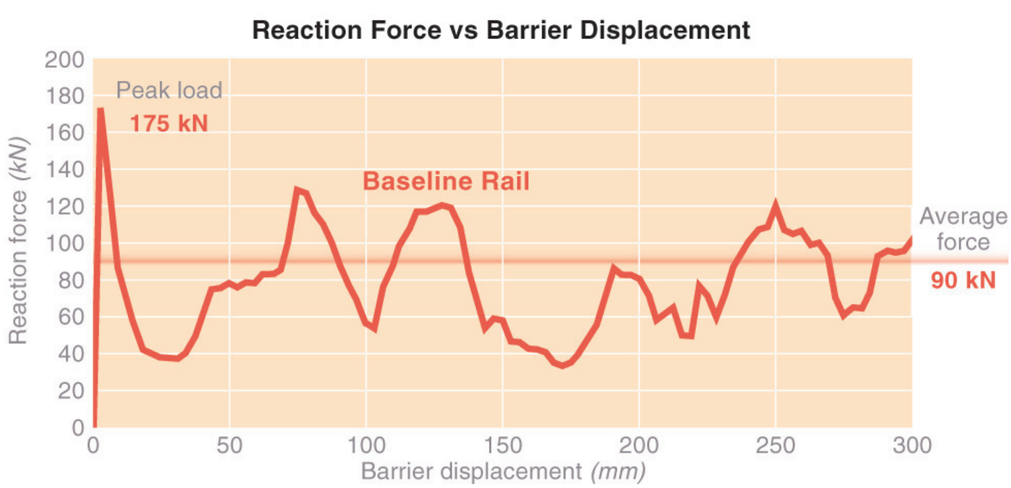


Figure 26: Baseline Frame Rail Crush Results

Material Properties

The baseline rails use 36-ksi low-carbon steel. In the DoE analysis, the yield strength could vary in a range between 36 ksi (250 MPa) and 85 ksi (500 MPa), with the assumption that a higher yield strength material would allow rail thickness down-gauging while maintaining a similar impact pulse. All materials represented in the Appendix D: Material List that fell within this yield strength range were considered for application in the rails.

Cross-Section Shape and Dimensions

The baseline rails are made of two C-Sections welded together to create a square cross-section which varies along the rail length (see Figure 26). In this lightweight rail study, the starting point was a closed square section. The nodes (see red points in Figure 27) were allowed to move independently, morphing into a shape that, according to the analysis, was the optimal structure to achieve the targeted performance.

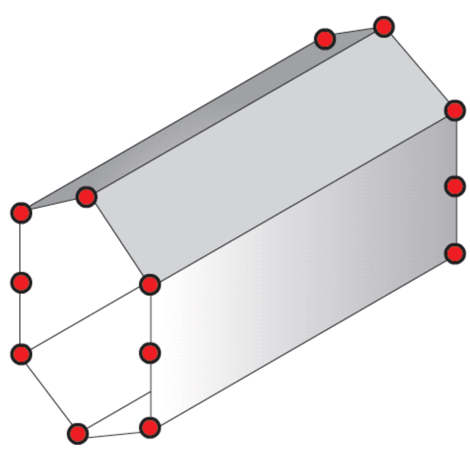


Figure 27: Front Rail Profile
(Click to View Animation)

Thickness

The baseline rail thicknesses were 2.5 mm for the inboard channel and 3.0 mm for the outboard channel. The design variables included a material thickness range of 1.5 to 3.5 mm.

Table 2 summarizes the design variables. The combination of performance improvements made through optimal material property selection and rail cross-section would allow a decrease in rail thickness, and thus a decrease in mass.

Rail crush event performance was judged by the peak load value and the energy absorbed during impact. The DoE analysis determined the combination of material, cross-section shape and dimensions that results in maximum mass reduction for a similar peak load and amount of absorbed energy, as previously illustrated in Figures 21a and 21b.

The DoE resulted in a hexagonal rail section, Figure 28, made of 2.25 mm HSLA420/480 (50 ksi) steel.

Note: Hydroforming this rail section might be challenging: additional work including forming simulation might be required to validate manufacturing concerns.

Table 2: Set Up Variables

Variables	Range
Material	36-85 ksi
Cross-Section	(Click To View Animation)
Thickness	1.5 to 3.5 mm

The optimized front rail geometry, which achieved an 18% mass reduction over the baseline design, was integrated into the new frame. A rail crush simulation was performed using the same setup used for the baseline, shown in Figures 22 and 23.

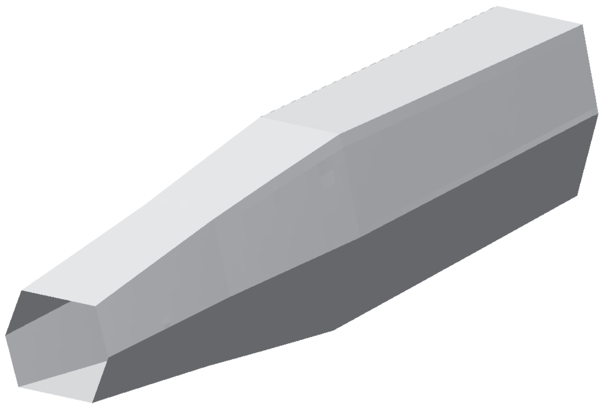


Figure 28: New Front Rail Resulting From the DoE

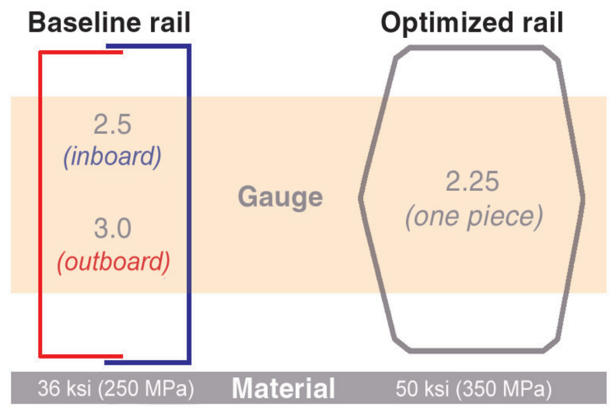


Figure 29: Baseline to Optimized Front Rail Comparison

The animation in Figure 31 gives a comparison between the baseline rail and the new rail crash performance. ❑

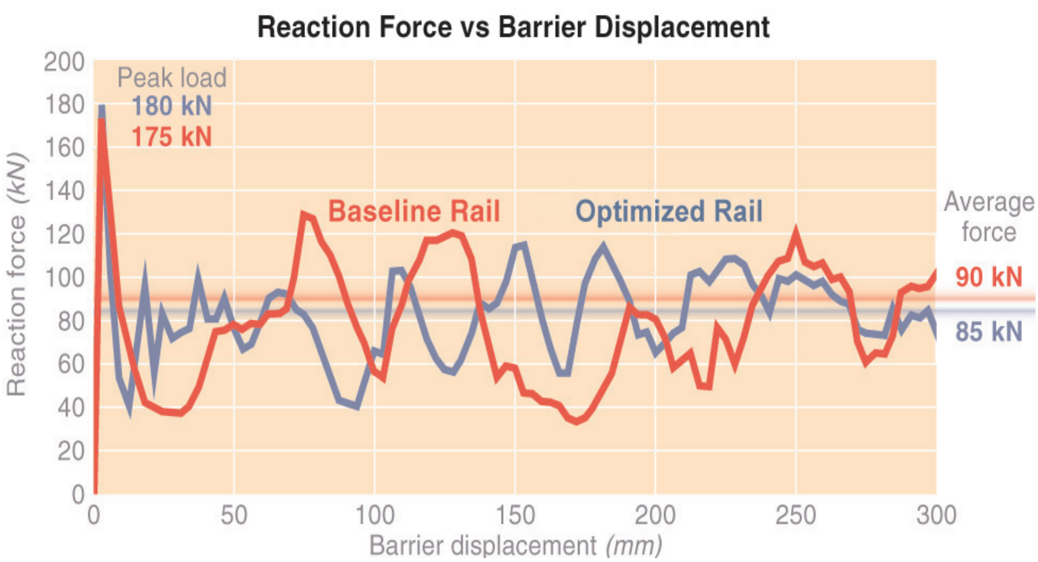


Figure 30: Rail Crush Performance Comparison

Figure 30 illustrates that the new lighter front rail performs similar to the baseline with +3% difference in peak load and -6% difference in average load.

Figure 31: Rail Crush Performance Comparison - Baseline Front Rail (left) and Lightweight Frame Front Rail (right)
(Click to View Animation)

6.0 COSTSTUDY

6.1 Background

In order to assess the potential cost impact of process and technology changes identified to reduce frame system mass, it was necessary to prepare a cost study that compared component and assembly costs of the baseline and concept frame systems. The cost study was not meant to be an exhaustive look at all elements of the design (or process) that contribute to overall cost, but rather a coarse estimate (using industry norms) to establish a reasonable means for cost comparison between old and new.

For the purposes of this study, our cost method is as follows:

- Product Cost is defined as a function of variable cost, tooling cost & capital cost.
- Variable Cost is defined as Piece Price + Packaging + Shipping:
  - Piece Price = (Materials + Labor)
  - Packaging = (Materials + Labor)
  - Shipping = (In + Out)

  Note: Materials + Labor include overhead (insurance, marketing, etc.) + profit + launch costs + amortized capital expenses.
- Tooling Cost is defined as all expenses not related to individual pieces, which are 100% directly billable to one part/project (e.g. molds, dies, assembly fixture, checking fixtures). In general, the customer pays for tooling.
- Capital Cost is defined as expenses related to manufacturing the individual pieces, but not 100% dedicated. Generally capital is expected to be paid for by the supplier. (e.g. presses, molding machines, buildings, etc.)

In order to establish this cost study, the following overall assumptions were made:

- 220,000 Frame Unit Annual Build
- Components will be produced outside of the assembly plant and delivered to the plant.
- Suppliers are already "production capable" (infrastructure and process in place) to produce stampings and/or "hydroform tube" components.
- Piece Cost for components will include variable and capital cost (amortized over (5) years). Tooling cost is managed by the customer and not included in these estimates.
- Frame assembly plant is already in operation, making a "like" product.
- Assembly costs include capital expenses to be amortized over (5) years, including capital cost improvements for hydroform transfer line system.
- Industry supply capacity is not considered in this study.
- No paint or wax coating costs were included for either the baseline or the proposed frame.

The cost study was conducted by Oxford Automotive. A summary of the process used and the results follows.

6.2 Process

6.2.1 Piece Cost Analysis

To establish piece cost estimates for the components that make up the baseline frame, Oxford Automotive used their industry expertise in the manufacture of chassis subframe components/subassemblies. From the supplied component detail drawings, the Team identified the most appropriate manufacturing methods/processes for developing each component and the resultant cost. The component materials specified in the baseline frame specification were used to evaluate material cost. An average material cost of 31¢/pound was used.

For the concept frame, Oxford utilized the methods defined above for those components that fit in with their current manufacturing infrastructure. For those components requiring a manufacturing process that was outside the scope of Oxford's current capabilities, Oxford utilized the resources and expertise of their current supply base in developing those cost estimates. An example would be the manufacturing of hydroform tube components. As stated above, it is assumed that the infrastructure to produce hydroform parts is resident within the supply base, and the cost differential between stamped components and hydroform tube parts is contained within the piece cost estimates.

Material types for the concept frame were selected from the A/SP "Materials Matrix Sheet" and matched to the performance requirements of the component, based on the structural analysis. Cost per pound pricing for these materials was supplied by Project Team members, to use for these estimates.

6.2.2 Assembly Cost Analysis

Considering that Oxford is not a supplier of frame systems, it was necessary for them to have industry knowledge/input in order to develop a reasonable assembly plant process in which to base their cost estimates. Oxford interviewed representatives from OE (Original Equipment) frame suppliers and toured assembly facilities to gain a better understanding how current frame system assemblies are processed.

Oxford Automotive created a virtual assembly plant plan and process in which to assemble the components of the baseline frame (Appendix F-Baseline Frame Assembly). This process and plan served to identify assembly, handling and specialty equipment needs, human resources and facility layout and space required to process the baseline frame, along with the associated costs. All assembly plant tool and equipment costs were derived from Oxford Automotive's current subframe assembly process experience.

Once the baseline frame process and assembly plant definition were established, Oxford Automotive evaluated the resultant changes that would be required in equipment, human resources and plant layout to assemble the concept frame. The revision to the plant layout (Appendix F - Lightweight Frame Concept Assembly) identifies those production cells that were reduced and/or combined as a result of the concept frame's component construction/integration. The floor space requirement was reduced as a result of the production cell changes.

Table 3 identifies the basic assessment for assembly plant definition for processing the baseline and concept frames.

Plant Requirements	Baseline	Concept
Assembly Line:	Appendix F	Appendix F
Operators	24	20
Robots:	104	72
MH Robots	13	4
MIG Robots	38	48
Sub Robots	53	
Floor Space	20,000 sq. ft.	20,000 sq. ft.
Hydroform Line	—	Turn Key System

Table 3: Assembly Plant Definition

Transfer Line Assumptions:

- turn-key system includes integrated plc system
- lift and carry system
- 36″ lift system
- Includes overhead structure, foot paths, etc.

6.3 Cost Results

Following in Tables 4, 5 and 6 are the cost analysis results:

Table 4: Stamping and Hydroforming Component Costs

	Baseline	Concept Frame
Annual Sales (220,000 units)	$58.4 M	$72.2 M
Tooling	$19.4 M	$17.5 M
Capital	$1.8 M	$2.6 M
Component Cost	$265.35/vehicle	$328.18/vehicle

Table 5: Assembly Costs

	Baseline	Concept Frame
Annual Sales (220,000 units)	$13.7 M	$7.6 M
Tooling	$29.2 M	$22.3 M
Capital	$5.2 M	$2.6 M
Assembly Cost	$62.29/vehicle	$34.56/vehicle

Table 6: Complete Frame Assembly Costs

	Baseline	Hourglass Frame
Annual Sales (220,000 units)	$72.1 M	$79.8 M
Tooling	$48.6 M	$39.8 M*
Capital	$5.2 M	$2.6 M
Total Frame Cost	$327.65	$362.72*

**Proposed frame tooling costs do not include hydroform tooling or equipment.*

7.0 LIGHTWEIGHT**FRAME**CONCEPT

The Project Team's primary challenge was to design a new lightweight SUV frame for the Ford Expedition/Lincoln Navigator while maintaining the performance requirements of the baseline frame. The new concept frame was intended to replace the baseline frame without major assembly and packaging issues. By utilizing analyses and advanced techniques relative to material, architecture and design, Altair designed a unique, non-traditional frame, which is 23% lighter and meets or exceeds the baseline frame performance (Figure 32). **Click Lightweight Frame Concept to view the new frame architecture in a Hyperview® player.**

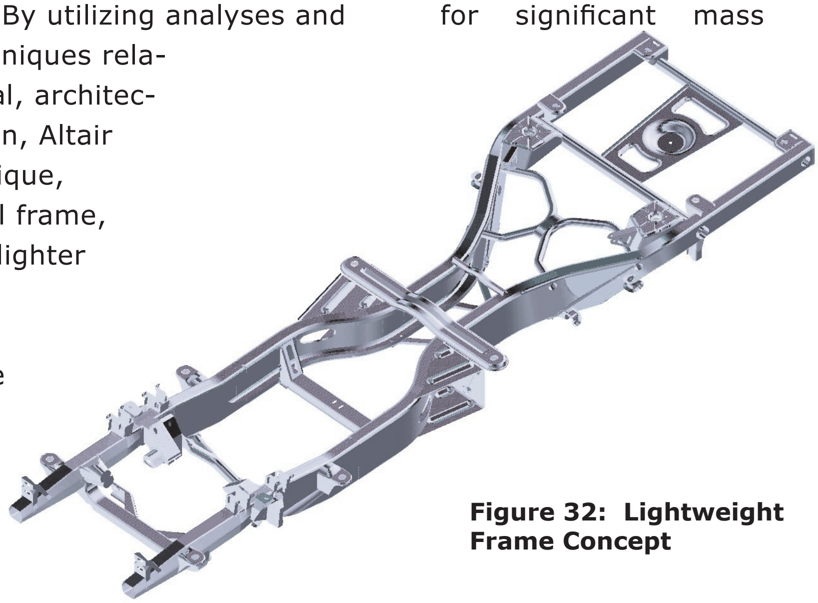


Figure 32: Lightweight Frame Concept

The frame is 100% steel, using hydroformed and stamped parts and the total overall frame weld length was reduced by 50% (see Appendix E). Also, the Joint Stiffness Tool Box was applied throughout the design process, incorporating many joint configurations that provide the best overall stiffness-to-weight ratio to achieve the resulting efficient Lightweight Frame Concept.

Appendices E and G provide images and descriptions of the components that make up the lightweight frame design, including material grades, gauges and manufacturing processes. Additionally, it provides information comparing packaging and mass to the baseline.

The lightweight frame design that results from this phase shows promising results for significant mass savings potential and quality structural performance and is an excellent foundation for a next-phase design effort. The Phase I study focused mainly on frame system performance. More in-depth investigation is necessary in order to evaluate full vehicle level performance, such as:

- Crash Management (Offset Barrier, Side Impact, Rear Impact)
- Body Mount Stiffness ❑

8.0 FUTURE**DEVELOPMENTS**

Successful design optimization and excellent performance results proved that significant mass reduction can be achieved with this unique lightweight frame design. Though there are some vehicle level engineering challenges yet to be met, as mentioned in Section 7.0, the Altair design engineering team is confident that these can be addressed effectively by progressing to a Phase II Design Development and Analysis stage. Once a Phase II concept design has been finalized, a Phase III Validation and Prototype Build is projected to prove out the concept and provide a working prototype and final engineering report. Altair and the Auto/Steel Partnership look to provide this valuable research as a roadmap to the automotive industry for developing lightweight, innovative steel frame structures that not only maintain or improve performance, but can be applied, in the near-term, to a manufacturer's SUV product development. ❑

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Program Technical Transfer Dispatch #6, May 2001
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[2] Light Truck Frame Joint Stiffness Study, July 2001
www.altair.com

Resources

[3] HyperView®, Altair Engineering Hyper-Works® Software

[4] OptiStruct®, Altair Engineering Hyper-Works® Software

[5] HyperMesh®, Altair Engineering Hyper-Works® Software

[6] LS-Dyna 3D, Livermore Software Technology Software

Appendices

APPENDIX A: Weight Reduction Strategies

Materials

1. Use HSS – higher strength (e.g. 340 and 420 MPa) and lower carbon (better weldability and formability).
2. Use AHSS (e.g., DP 600).
3. Provide a weight by item list for examination by the Project Team.
4. Concentrate on materials for the side rails since they have the greatest mass.
5. Replace the wax coating.
6. Use metallic (zinc) coated sheet.
7. Use Boron steels and heat treatment.
8. Use steel/plastic/steel sandwich material.

Manufacturing

1. Use hydroformed rails.
2. Use tailored blanks (e.g., rails).
3. Use tailored tubes.
4. Use one-piece rails.
5. Butt-weld rail sections together.
6. Use conical tubes.
7. Replace heavy welds using magnetic pulse/laser/laser assisted arc welding.
8. Use roll formed sections.
9. Use extruded sections.
10. Use patch technology.
11. Use metal foam.

Design

1. Include the bumper beam when determining frame properties.
2. Improve the bumper to frame connection (e.g., welding).
3. Down-gauge the rear end of the frame.
4. Use closed sections at the rear of the frame.
5. Use cross-bracing and tubes for tire carrier.
6. Evaluate the fuel tank cross-members.
7. Down-gauge the brackets using HSS/AHSS/UHSS.
8. Reduce the depth and mass of the center section of the rail.
9. Shape the rail depth to the moment diagram.
10. Increase the section sizes.
11. Use lightening techniques.
12. Scallop free edges.
13. Use stiffer connections (e.g., tube-to-tube)
14. Use metal foam to stiffen joints.
15. Use more cross-members.
16. Position the cross-members to triangulate between the rails.
17. Optimize load path.

APPENDIX B – LIGHT TRUCK FRAME JOINT STIFFNESS STUDY

Light Truck Frame Joint Stiffness Study, July 25, 2001
sponsored by the Auto-Steel Partnership
authored by Altair Engineering

An example of the toolbox developed in the Joint Stiffness Study is shown below. Values shown in yellow (size and thickness variables) can be adjusted to view the stiffness and mass effects (in red). General design rules and observations are listed on the right for each type of joint.

Joint #1: Tube Through Tube

Stiffness Calculations

Shape Animations	Design Variables (mm)	Input (mm) Case 1	Case 2	Test Joint	Design Window Min(mm)	Max (mm)
	Thickness Crossmember	2.5	2.5	2.5	2	6
	Side Rail	2.8	2.8	2.8	2	6
	Shape Variables (mm)					
animations\\...	Crossmember Diameter	57	57	57	50	100
animations\\...	Side Rail Height	125	125	125	75	150
animations\\...	Side Rail Width	125	125	125	75	150

Output

Loading Animations	Stiffness Calculations	Case 1	Case 2	Test Joint	Units
animations\\...	Kx Bending Stiffness (Mx)	3.235	3.235	3.477	kN-m/deg
animations\\...	Ky Torsion Stiffness (My)	3.271	3.271	12.715	kN-m/deg
animations\\...	Kz Fore/Aft Stiffness (Mz)	3.507	3.507	3.662	kN-m/deg
	Mass	4.801	4.801	4.801	kg

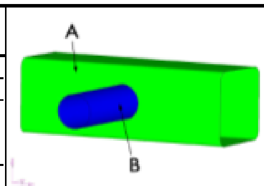


Illustration Key:
A= Side Rail
B = Crossmember

Design Rules:
* Make the crossmember diameter as large as possible.

* Make the crossmember as thick as possible.

* Thickness and diameter of the crossmember should be increased together if possible.

Joint Observations:
* The crossmember is the most important part of this joint. The thickness of the crossmember is 3X more sensitive than the thickness of the side rail.

* The maximum stress is in the crossmember at the connection to the inner side rail for all 3 Stiffness Cases.

* The outer weld (the crossmember to outer side rail) could be a partial weld because this section of the joint has low stress.

Notes:
* Modify the input values (yellow) based on your design criteria. There are two columns in which in to input and evaluate data, case 1 and case 2. The calculated stiffness will be displayed in red.

* Design variables are listed in order of influence on stiffness.

* Click on the animation to the left of the variables and loading conditions to see an animation of respective shape variable or loading condition.

* The mass calculation is based on 150 mm extension of joint members from the side rail to crossmember interface. (The crossmember is 150mm from the joint interface to the end of the crossmember). This caluclation is to serve as a reference, not the absolute value.

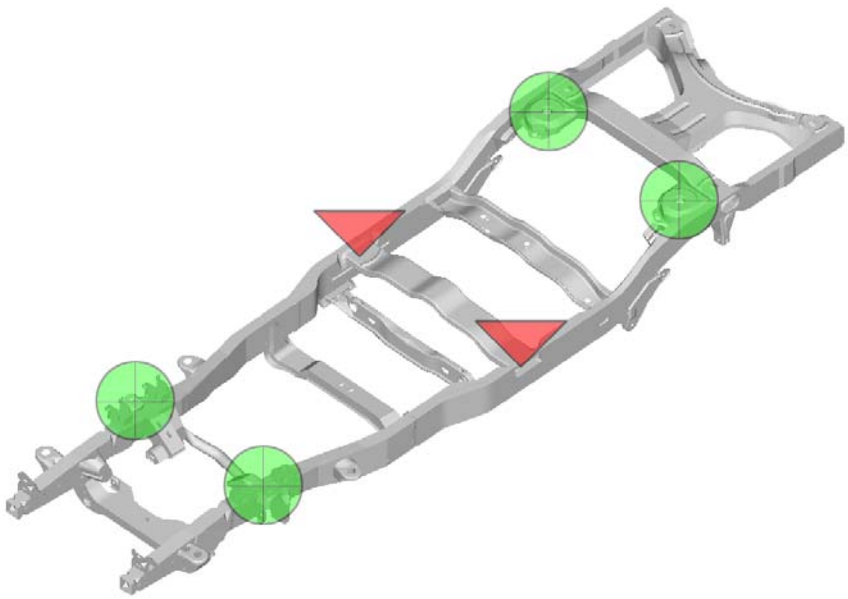
* The password to unprotect cells in this spreadsheet is: steel.

Yellow cells are user input data.

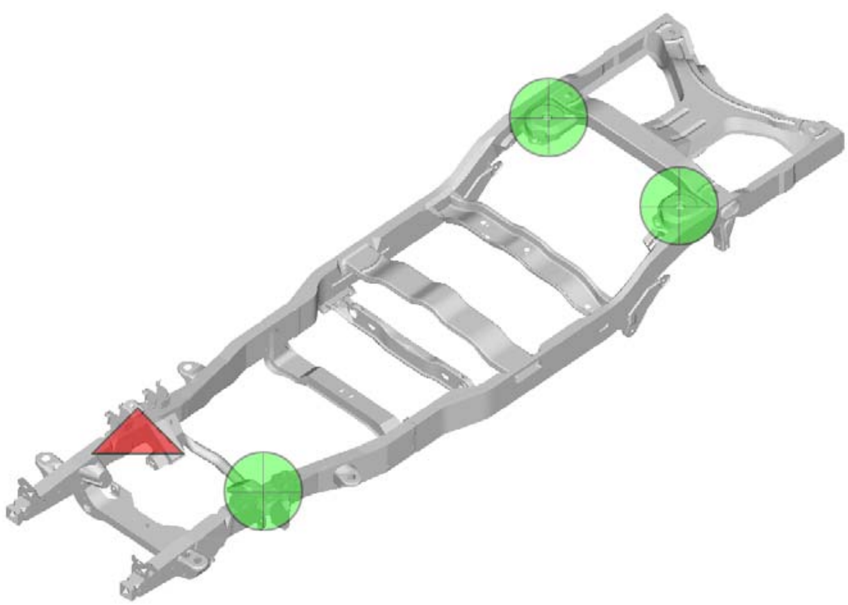
Red cells are calculated numbers.

APPENDIX C – FE ANALYSIS - BOUNDARY CONDITIONS

Bending Stiffness



Torsional Stiffness



These pictures show the boundary and loading conditions used for the baseline frame in order to determine the bending and torsional stiffness targets. These loading conditions were carried throughout the project for the optimization analyses.

APPENDIX D: MATERIAL LIST

HOT ROLLED SHEET (BARE)

STEEL DESIGNATION	EQUIVALENT SAE SPECIFICATION	AVAILABLE WIDTH (mm)	AVAILABLE THICKNESS (mm)	TYPICAL OR MEAN YIELD STRENGTH (MPa)	TYPICAL OR MEAN TENSILE STRENGTH (MPa)	TYPICAL OR MEAN ELONGATION (%)	TYPICAL "n" VALUE	TYPICAL "r" VALUE
MILD 205/280 DQ	SAE J2329 HR2	610-1829	1.397-9.525	234	331	42	0.2	1.1
MILD 205/280	SAE J2329 HR1	610-1829	1.397-9.525	269	386	39	0.19	
CMn 205/340	SAE J2329 HR1	610-1575	1.778-5.842	276	393	38	1.09	
CMn 230/360	SAE J2329 HR1	610-1575	1.778-5.842	290	393	38		
CMn 250/380	SAE J2340 HR300S	610-1575	1.778-5.842	310	393	35		
CMn 280/410	SAE J2340 HR300Y	610-1575	1.778-5.842	338	434	30		
CMn 350/460	SAE J2340 HR340X	610-1575	1.778-5.842	421	483	29	0.16	
CMn 380/470	SAE J2340 HR340Y	610-1575	1.778-5.842	421	490	29	0.16	
CMn 420/480	SAE J2340 HR420X	610-1575	1.778-5.842	503	572	28	0.12	
HSLA 250/340	SAE J2340 HR300S	610-1829	1.778-12.7	303	386	35	0.17	
HSLA 280/370	SAE J2340 HR300S	610-1829	1.778-12.7	331	407	35	0.17	
HSLA 315/390	SAE J2340 HR300Y	610-1829	1.778-12.7	352	434	33	0.17	
HSLA 350/448	SAE J2340 HR340X	610-1829	1.778-12.7	355	448	34		
HSLA 400/500	SAE J2340 HR380X			433	503	30		
HSLA 420/480	SAE J2340 HR420X	610-1829	1.778-12.7	476	531	27	0.15	
HSLA 490/550	SAE J2340 HR490X	610-1575	1.778-6.35	531	600	26	0.13	
HSLA 560/620	SAEJ2340 HR550X	610-1575	1.778-6.35	586	676	22	0.12	
DP 340/590	SAE J2340 HR600DL2	1524 max.	2.50-6.00	340	590			
DP 340/600	SAE J2340 HR600DL2	1524 max.	2.50-6.00	340	600			
FORD "A" ESA-M1A33-C	SAE J2340 HR210A			221 (min)	324 (min)	20 (min)		
FORD "B" ESA-M1A35-C	SAE J2340 HR250A			248 (min)	345 (min)	20 (min)		
FORD "C" SAE J1392 (340YHF HSLA)	SAE J2340 HR340Y			340 (min)	450 (min)	20 (min)		

APPENDIX D: MATERIAL LIST CONTINUED

COLD ROLLED SHEET (BARE OR ELECTROGALVANIZED)

STEEL DESIGNATION	EQUIVALENT SAE SPECIFICATION	AVAILABLE WIDTH (mm)	AVAILABLE THICKNESS (mm)	TYPICAL OR MEAN YIELD STRENGTH (MPa)	TYPICAL OR MEAN TENSILE STRENGTH (MPa)	TYPICAL OR MEAN ELONGATION (%)	TYPICAL "n" VALUE	TYPICAL "r" VALUE
BH180/330	SAE J2340 CR180B BARE OR EG	708-1600	0.43-2.01	205	330	41	0.20	
BH 210/365	SAE J2340 CR210B BARE OR EG	708-1650	0.60-2.03	235	365	39	0.19	
BH 250/375	SAE J2340 CR250B BARE OR EG	708-1650	0.60-2.03	255	375	37	0.18	
BH 280/415	SAE J2340 CR280B BARE OR EG	610-1650	0.48-2.03	300	415	36	0.17	
BH 300/445	SAE J2340 CR300B BARE OR EG	708-1650	0.60-2.01	320	445	35	0.17	
HSLA 250/400	SAEJ2340 CR280X BARE OR EG			285	400	35		
HSLA 350/515	SAE J2340 CR340X BARE OR EG	610-1515	0.48-2.03	370	515	29		
HSLA 450/525	SAE J2340 CR420X BARE OR EG			470	525	27		
HSLA 500/715	SAE J2340 CR490X BARE OR EG			540	715	20		
DP 340/590	SAE J2340 CR600DL2 BARE OR EG	610-1515	0.48-2.16	370	665	23		
DP 550/690	SAE J2340 CR700DH BARE OR EG	610-1515	0.48-2.16	592	746	17		
DP 630/965	SAE J2340 CR700DH BARE OR EG	610-1515	0.48-2.16	592	746	17		
DP 550/980	SAE J2340 CR950DL BARE OR EG	610-1515	0.48-2.16	634	1015	13		
Mart 700/900	SAE J2340 CR1000M BARE OR EG	610-1515	0.48-2.00	910	1045	5		
Mart 860/1100	SAE J2340 CR1100M BARE OR EG	610-1515	0.48-2.00	1020	1178	5		
Mart 10301300	SAE J2340 CR1300M BARE OR EG	610-1515	0.48-2.00	1183	1392	6		
Mart 1200/1500	SAE J2340 CR1500M BARE OR EG	610-1515	0.48-2.00	1350	1596	6		

COLD ROLLED SHEET (ELECTROGALVANIZED OR HOT DIP GALVANIZED)

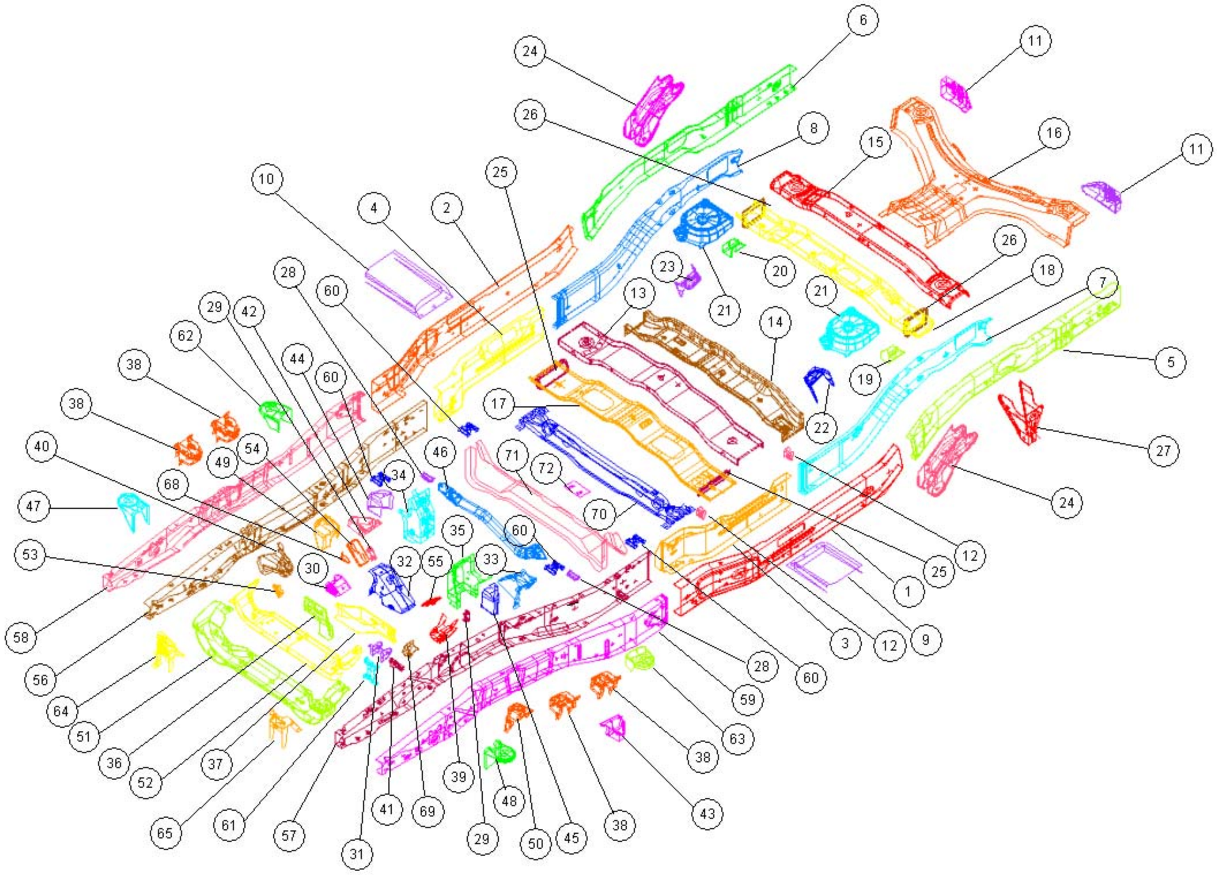
STEEL DESIGNATION	EQUIVALENT SAE SPECIFICATION	AVAILABLE WIDTH (mm)	AVAILABLE THICKNESS (mm)	TYPICAL OR MEAN YIELD STRENGTH (MPa)	TYPICAL OR MEAN TENSILE STRENGTH (MPa)	TYPICAL OR MEAN ELONGATION (%)	TYPICAL "n" VALUE	TYPICAL "r" VALUE
DR 210/350	SAE J2340 CR210A EG OR HD	760-1575	0.70-1.50	220	360	40		

COLD ROLLED SHEET (HOT DIP GALVANIZED) OR HOT ROLLED SHEET (HOT DIP GALVANIZED)

STEEL DESIGNATION	EQUIVALENT SAE SPECIFICATION	AVAILABLE WIDTH (mm)	AVAILABLE THICKNESS (mm)	TYPICAL OR MEAN YIELD STRENGTH (MPa)	TYPICAL OR MEAN TENSILE STRENGTH (MPa)	TYPICAL OR MEAN ELONGATION (%)	TYPICAL "n" VALUE	TYPICAL "r" VALUE
MILD 205/270	SAE J2329 CR3 HD	610-1650	0.55-2.50	170	303	48	0.25	
MILD CQ	SAE J2329 CR1 HD	1828 max.	0.60-2.50					
MILD 245/350 DQ	SAE J2329 CR2 HD	1828 max.	0.60-2.50	255	358	35	0.19	1.52
HSLA 280/345	SAE J2340 CR250A HD	610-1830	0.61-0.91	310	385	38	0.23	
HSLA 340/420	SAE J2340 CR300S HD	610-1525	0.70-1.80	344	420	27	0.18	
DP 300/500	SAE J2340 CR500DL HD	1524 max.	0.80-1.50	300	500			
DP 350/500	SAE J2340 CR500DL HD							
DP 340/600	SAE J2340 CR600DL 2 HD	1651 max	0.70-1.80	370	630	24		
DP 350/600	SAE J2340 CR600DL1 HD	1651 max.	0.70-1.80	350	600			
DP 500/800	SAE J2340 CR800DL HD			500	800			
MILD 185/310	SAE J2329 HR3 HD	810-1825	3.03-3.51	185	310		0.212	
MILD 205/280 DQ	SAE J2329 HR2 HD	610-1829	1.397-9.525	234	331	42	0.2	1.1
MILD 300/365	SAE J2329 HR1 HD	810-1825	1.52->3.51	303	366		0.178	
CMn 300/380	SAE J2340 HR300X HD	810-1825	1.52->3.51	307	381		0.179	
CMn 315/410	SAE J2340 HR300S HD	810-1825	2.03-3.51	317	414		0.174	
CMn 335/465	SAE J2340 HR300Y HD	810-1825	2.03-2.49	337	467		0.160	
CMn 365/430	SAE J2340 HR340X HD	810-1825	3.51 and up	365	431		0.162	
HSLA 370/470	SAE J2340 HR340Y HD	810-1825	2.03-3.00	370	474		0.164	
CMn 390/555	SAE J2340 HR380Y HD	810-1825	2.03-3.51	392	557		0.152	
HSLA 395/460	SAE J2340 HR380X HD	810-1825	2.03-3.51	399	463		0.166	
CMn 455/520	SAE J2340 HR420X HD	810-1825	3.51 and up	458	524		0.144	
HSLA 465/515	SAE J2340 HR420X HD	810-1825	2.03-3.51	468	516		0.146	
HSLA 480/530	SAE J2340 HR420X HD	810-1825	2.03-2.49	480	530		0.143	
HSLA 630/675	SAE J2340 HR550X HD	810-1825	2.51-3.00	632	676		0.112	

APPENDIX E – COMPONENTS SUMMARY

Baseline Frame Components - Exploded View



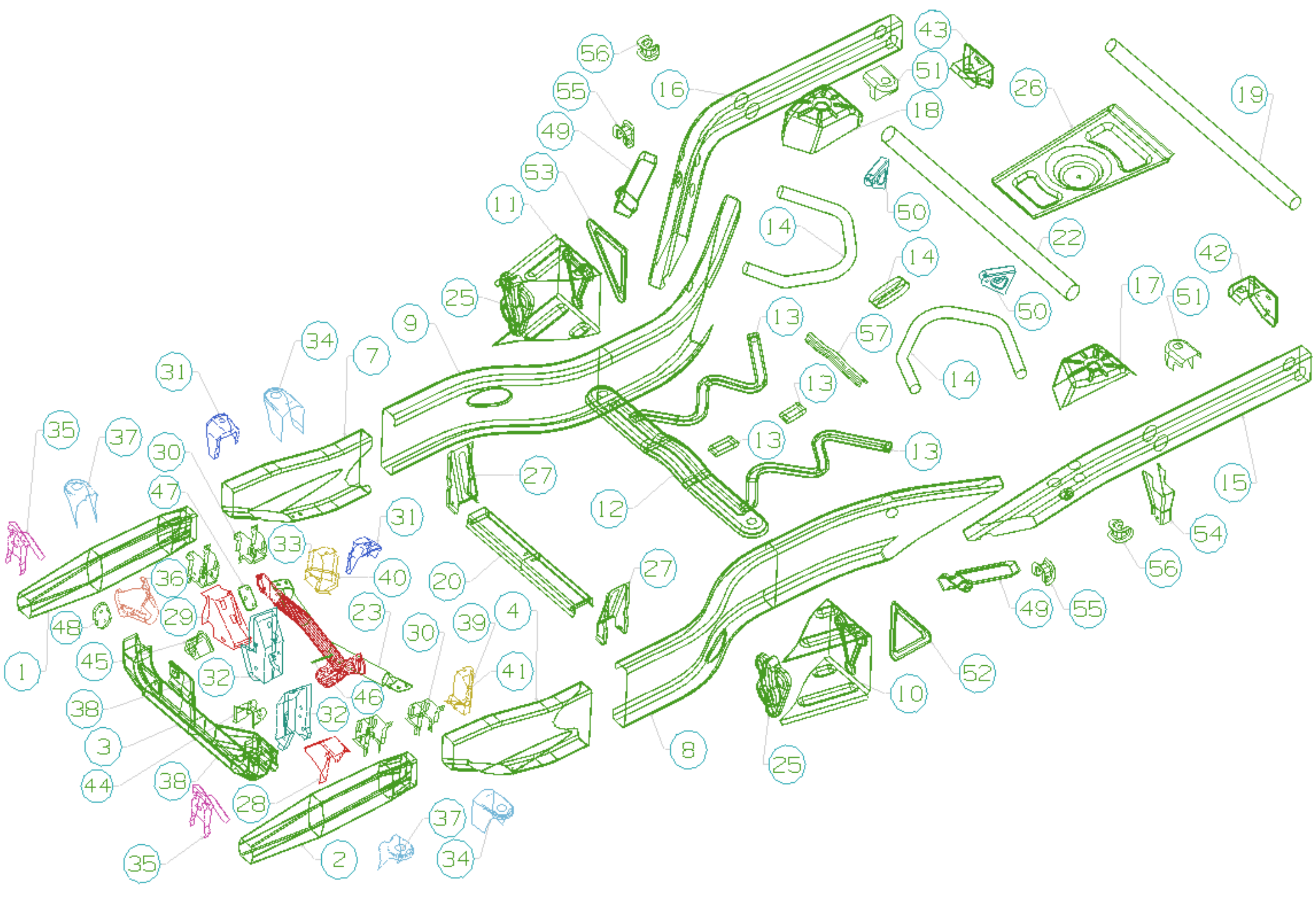
APPENDIX E – COMPONENTS SUMMARY CONTINUED

Baseline Frame Components – Part Information

Part ID	NAME	THICKNESS* (mm)	MASS (kg)	# PCS REQ.	Material	TYPE of TOOLING
1	CENTER S/M RH	2.50	8.700	1	Grade A - ESA-1A33-C	Stamped
2	CENTER S/M LH	2.50	8.700	1	Grade A - ESA-1A33-C	Stamped
3	CENTER INNER S/M RH	2.50	4.650	1	Grade A - ESA-1A33-C	Stamped
4	CENTER INNER S/M LH	2.50	4.650	1	Grade A - ESA-1A33-C	Stamped
5	REAR S/M LH	3.00	7.850	1	Grade A - ESA-1A33-C	Stamped
6	REAR S/M RH	3.00	7.850	1	Grade A - ESA-1A33-C	Stamped
7	REINF-FRAME S/M RR LH	2.50	5.000	1	Grade A - ESA-1A33-C	Stamped
8	REINF-FRAME S/M RR RH	2.50	5.000	1	Grade A - ESA-1A33-C	Stamped
9	TORSION BAR BRKT LH	3.00	0.700	1	Grade B - ESA-1A35-C	Stamped
10	TORSION BAR BRKT RH	3.00	0.700	1	Grade B - ESA-1A35-C	Stamped
11	REINF FR C/M RR LH/RH	4.20	1.450	2	Grade B - ESA-1A35-C	Stamped
12	PARKING BRAKE CBL BRKT LH	3.90	0.125	2	Grade B - ESA-1A35-C	Stamped
13	#3 CROSSMEMBER UPPER	2.50	4.800	1	Grade C - SAE J1392 340 YHF HSLA	Stamped
14	#4 CROSSMEMBER	2.50	6.500	1	Grade B - ESA-1A35-C	Stamped
15	#5 CROSSMEMBER UPPER	2.50	5.000	1	Grade B - ESA-1A35-C	Stamped
16	#6 CROSSMEMBER	2.50	12.000	1	Grade B - ESA-1A35-C	Stamped
17	#3 CROSSMEMBER LOWER	2.50	5.100	1	Grade B - ESA-1A35-C	Stamped
18	#5 CROSSMEMBER LOWER	2.50	5.100	1	Grade B - ESA-1A35-C	Stamped
19	BRKT-RR SUSP JNC BM LH	3.00	0.300	1	Grade B - ESA-1A35-C	Stamped
20	BRKT-RR SUSP JNC BM RH	3.00	0.300	1	Grade B - ESA-1A35-C	Stamped
21	REAR COIL SPRING BRKT LH/RH	3.50	2.500	2	Grade B - ESA-1A35-C	Stamped
22	REAR SHOCK BRKT LH	3.50	0.600	1	Grade B - ESA-1A35-C	Stamped
23	REAR SHOCK BRKT RH	3.50	0.600	1	Grade B - ESA-1A35-C	Stamped
24	REAR CONTROL ARM BRKT	4.00	4.000	2	Grade B - ESA-1A35-C	Stamped
25	#3 C/M ADAPTER BRKT	3.00	0.300	2	Grade B - ESA-1A35-C	Stamped
26	#5 C/M ADAPTER BRKT	3.00	0.300	2	Grade B - ESA-1A35-C	Stamped
27	TRACK BAR BRKT	3.00	1.750	1	Grade C - SAE J1392 340 YHF HSLA	Stamped
28	TRANSMISSION CROSSMEMBER SPACER	3.50	0.125	2	Grade B - ESA-1A35-C	Stamped
29	PLT-BRKT LCA MTG ANC	4.00	0.075	2	Grade B - ESA-1A35-C	Stamped
30	AXLE MOUNTING BRKT (ENGINE BRKT)	3.50	0.550	1	Grade B - ESA-1A35-C	Stamped
31	AXLE MOUNTING BRACKET (#1 C/M)	3.50	0.350	1	Grade B - ESA-1A35-C	Stamped
32	ENGINE MOUNTING BRKT RH	3.50	1.850	1	Grade B - ESA-1A35-C	Stamped
33	ENGINE MOUNTING BRKT LH	3.50	1.300	1	Grade B - ESA-1A35-C	Stamped
34	REAR LCA BRKT RH	4.00	2.850	1	Grade B - ESA-1A35-C	Stamped
35	REAR LCA BRKT LH	4.00	2.850	1	Grade B - ESA-1A35-C	Stamped
36	BRKT-FRT SUSP LWR RH	3.50	1.200	1	Grade B - ESA-1A35-C	Stamped
37	BRKT-FRT SUSP LWR LH	3.50	1.900	1	Grade B - ESA-1A35-C	Stamped
38	UCA BRKT	3.50	0.800	4	Grade B - ESA-1A35-C	Stamped
39	REAR PINION MOUNT BRACKET	4.00	0.600	1	Grade B - ESA-1A35-C	Stamped
40	BRKT-STRNG IDL ARM	3.00	1.500	1	Grade B - ESA-1A35-C	Stamped
41	MONOCULAR STEERING GEAR SPACER	5.00	0.250	1	Grade B - ESA-1A35-C	Stamped
42	FRONT JOUNCE BRKT RH	3.50	0.750	1	Grade B - ESA-1A35-C	Stamped
43	FRONT JOUNCE BRKT LH	3.50	0.750	1	Grade B - ESA-1A35-C	Stamped
44	REINF-RH JOUNCE BUMPER	3.50	0.850	1	Grade B - ESA-1A35-C	Stamped
45	REINF-LH JOUNCE BUMPER	3.50	0.850	1	Grade B - ESA-1A35-C	Stamped
46	CROSSMEMBER FRT INTERM	3.00	3.500	1	Grade B - ESA-1A35-C	Stamped
47	BRKT FRM TO RAD RH	3.00	0.750	1	Grade A - ESA-1A33-C	Stamped
48	BRKT FRM TO RAD LH	3.00	0.750	1	Grade A - ESA-1A33-C	Stamped
49	FRONT SHOCK BRKT RH	3.50	0.900	1	Grade B - ESA-1A35-C	Stamped
50	FRONT SHOCK BRKT LH	3.50	0.900	1	Grade B - ESA-1A35-C	Stamped
51	C/M-FR FRT LWR	3.50	7.850	1	Grade B - ESA-1A35-C	Stamped
52	C/M-FR FRT UPR	3.50	5.000	1	Grade B - ESA-1A35-C	Stamped
53	TAPPING PLATE	3.50	0.100	1	Grade B - ESA-1A35-C	Stamped
54	REINF-BRKT ENGINE RH	3.50	0.250	1	Grade B - ESA-1A35-C	Stamped
55	REINF-BRKT ENGINE LH	3.50	0.200	1	Grade B - ESA-1A35-C	Stamped
56	FRONT INNER SIDEMEMBER RH	2.50	10.500	1	Grade A - ESA-1A33-C	Stamped
57	FRONT INNER SIDEMEMBER LH	2.50	10.500	1	Grade A - ESA-1A33-C	Stamped
58	FRONT OUTER SIDEMEMBER RH	3.00	11.400	1	Grade A - ESA-1A33-C	Stamped
59	FRONT OUTER SIDEMEMBER LH	3.00	11.400	1	Grade A - ESA-1A33-C	Stamped
60	SPACER-FRAME SIDE MEMBER REINF	3.50	0.250	6	Grade B - ESA-1A35-C	Stamped
61	BINOCULAR STEERING GEAR SPACER	5.00	0.650	1	Grade B - ESA-1A35-C	Stamped
62	BRKT-FRAME TO BODY RH	3.00	0.700	1	Grade B - ESA-1A35-C	Stamped
63	BRKT-FRAME TO BODY LH	3.00	0.700	1	Grade B - ESA-1A35-C	Stamped
64	BRKT FRT BMPR RH	4.00	1.000	1	Grade C - SAE J1392 340 YHF HSLA	Stamped
65	BRKT FRT BMPR LH	4.00	1.000	1	Grade C - SAE J1392 340 YHF HSLA	Stamped
66*	FRONT SHOCK BRKT RH	3.50	N/A	1	Grade B - ESA-1A35-C	Stamped
67*	FRONT SHOCK BRKT LH	3.50	N/A	1	Grade B - ESA-1A35-C	Stamped
68	ENGINE MOUNT REINF.	3.00	0.050	1	Grade B - ESA-1A35-C	Stamped
69	STA BAR REINF.	3.50	0.200	1	Grade B - ESA-1A35-C	Stamped
70	F75A-5D034-AA	4.00	10.300	1		Stamped
71	XL34-5059-CB	3.00	7.500	1		Stamped
72	XL34-5059-CB-PLATE	3.50	0.200	1		Stamped
	Welds (Total Length = 51,000 mm)		3.000			Stamped
Total Mass (kg)			226.0			

APPENDIX E – COMPONENTS SUMMARY CONTINUED

Lightweight Frame Concept Components – Exploded View

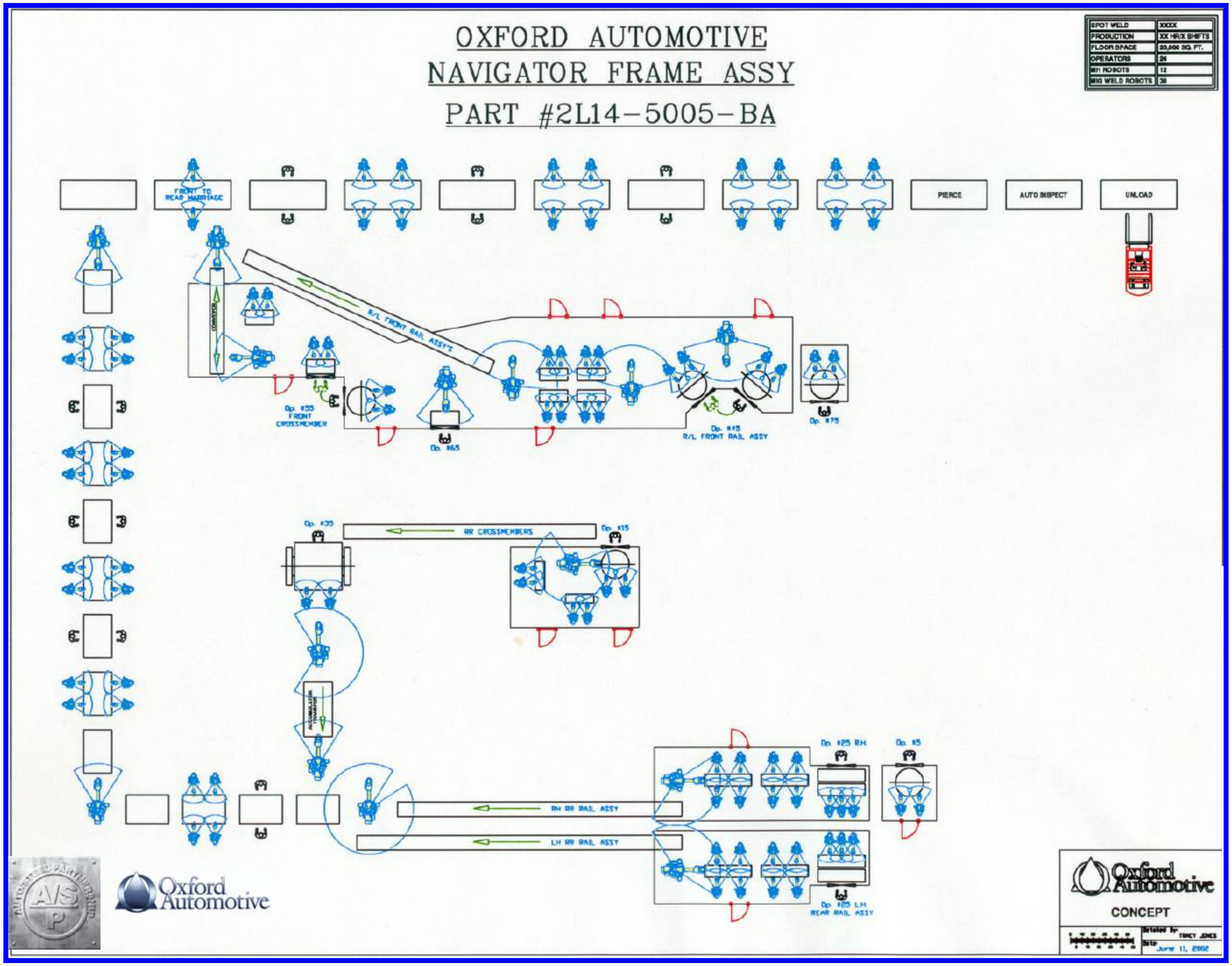


APPENDIX E – COMPONENTS SUMMARY CONTINUED

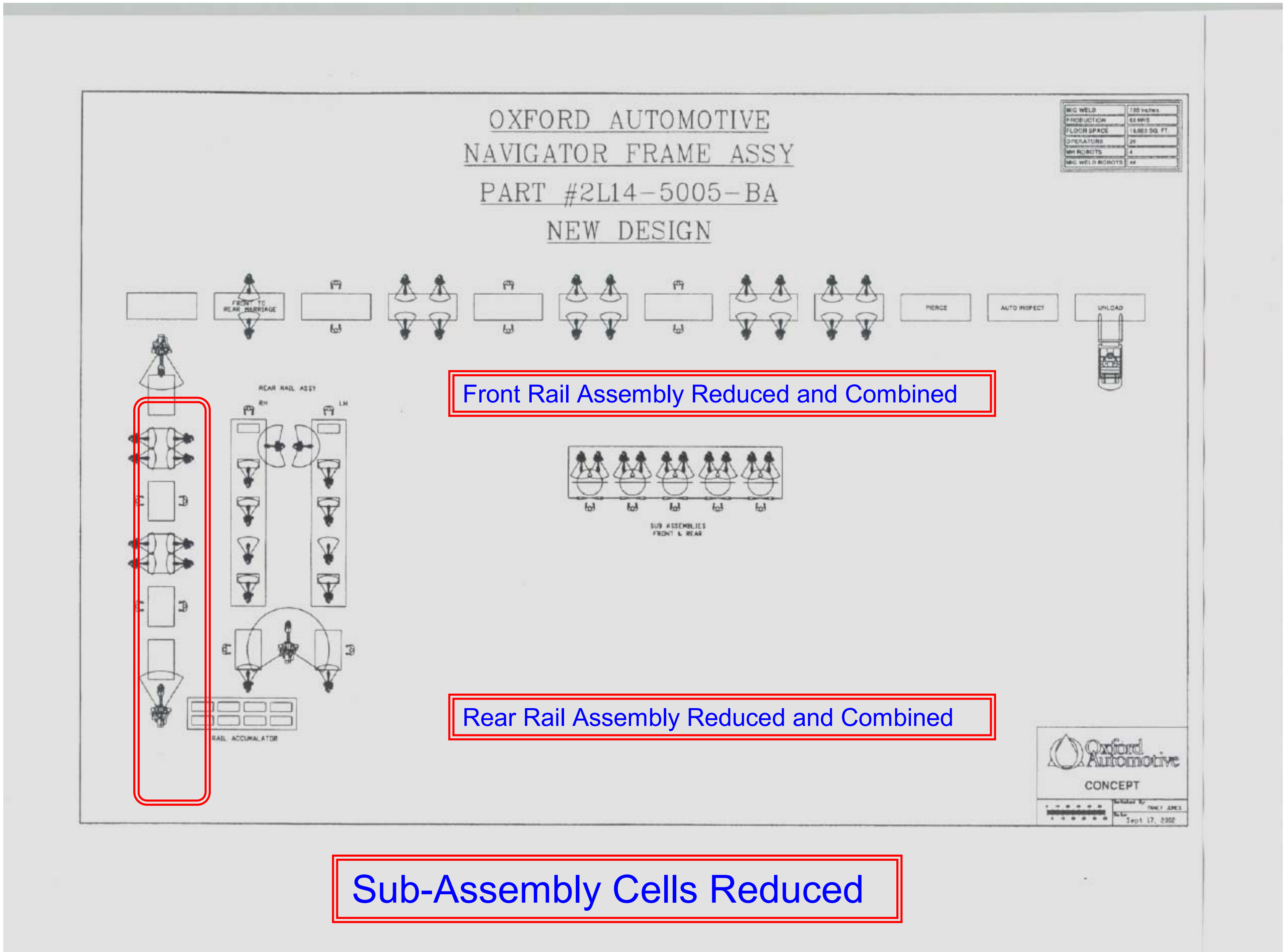
Lightweight Frame Concept Components – Part Information

Part ID	NAME	THICKNESS [mm]	MASS (kg)	# PCS REQ	Material	FEA YS (MPa)	TYPE of TOOLING
1	S/M-Frame Frt RH	2.25	7.500	1	HR HSLA 420/480	425	Hydroform
2	S/M-Frame Frt LH	2.25	7.500	1	HR HSLA 420/480	425	Hydroform
3	C/M #1	2.00	7.100	1	HR MILD 205/280 DQ	TBD	Hydroform
4	S/M-FRAME CTR EXT LH	2.25	9.300	1	HR MILD 205/280 DQ	225	Hydroform
7	S/M-FRAME CTR EXT RH	2.25	9.300	1	HR MILD 205/280 DQ	225	Hydroform
8	S/M-FRAME CTR LH	2.90	21.600	1	HR HSLA 350/448	TBD	Stamped
9	S/M-FRAME CTR RH	2.90	21.600	1	HR HSLA 350/448	TBD	Stamped
10	REINF C/M #3 LH	1.40	2.350	1	EG DP 550/690	TBD	Stamped
11	REINF C/M #3 RH	1.40	2.350	1	EG DP 550/690	TBD	Stamped
12	C/M #3	1.40	3.650	1	HDG CR HSLA 340/420	333	Stamped
13	REINF ASY-FRAME CTR LWR	1.20	3.750	1	HDG CR MILD 245/350 DQ	TBD	Hydroform
14	REINF ASY-FRAME CTR UPR RR	1.50	3.350	1	HDG CR MILD 245/350 DQ	TBD	Hydroform
15	S/M-FRAME RR LH	1.20	6.550	1	HDG CR DP 340/600	TBD	Hydroform
16	S/M-FRAME RR RH	1.20	6.550	1	HDG CR DP 340/600	TBD	Hydroform
17	BRKT-SPRING MTG LH	3.50	3.600	1	HR DP 340/590	TBD	Stamped
18	BRKT-SPRING MTG RH	3.50	3.600	1	HR DP 340/590	TBD	Stamped
19	C/M #6	1.20	2.400	1	HDG CR MILD CQ	TBD	Hydroform
20	C/M #2	1.25	1.550	1	HDG CR MILD 245/350 DQ	TBD	Stamped
22	C/M #5	1.30	2.950	1	HDG CR MILD 245/350 DQ	TBD	Hydroform
23	C/M #4	2.25	4.750	1	HR HSLA 420/480	TBD	Hydroform
25	TORS BAR MNT LH	2.00	1.110	2	CR DP 340/590	TBD	Stamped
26	TIRE CARRIER	1.20	2.700	1	HDG CR MILD 245/350 DQ	TBD	Stamped
27	C/M #2 BRKT	2.00	0.760	2	HR MILD 205/280 DQ	TBD	Stamped
28	ENG MNT BRKT LH	3.50	1.300	1	ESA-M1A35-C	199	Stamped
29	ENG MNT BRKT RH	3.50	1.850	1	ESA-M1A35-C	324	Stamped
30	UCA MNTS	3.50	0.800	2	ESA-M1A35-C	186	Stamped
31	SHOCK MNT BRKT	3.50	0.900	2	ESA-M1A35-C	TBD	Stamped
32	LWR CNTL ARM MT	4.00	2.850	2	ESA-M1A35-C	497	Stamped
33	JNC BUMPER RH	3.50	0.850	1	ESA-M1A35-C	45	Stamped
34	BODY MNT #2 BRKT	3.00	0.700	2	ESA-M1A35-C	250	Stamped
35	BUMPER BRKT FRT	4.00	1.000	2	SAE J1392	56	Stamped
36	IDLER ARM MNT	3.00	1.500	1	ESA-M1A35-C	34	Stamped
37	BODY MNT #1	3.00	0.750	2	ESA-M1A35-C	310	Stamped
38	C/M #1 Reinf	2.00	0.900	2	HR HSLA 560/620	292	Stamped
39	JNC BUMPER LH	3.50	0.850	1	ESA-M1A35-C	45	Stamped
40	JNC BRKT REINF RH	3.50	0.750	1	ESA-M1A35-C	137	Stamped
41	JNC BRKT REINF LH	3.50	0.750	1	ESA-M1A35-C	137	Stamped
42	REAR BUMPER BRKT LH	3.00	1.350	1	HR HSLA 560/620	TBD	Stamped
43	REAR BUMPER BRKT RH	3.00	1.350	1	HR HSLA 560/620	TBD	Stamped
44	AXLE MNT LH	3.50	0.350	1	HR HSLA 560/620	TBD	Stamped
45	AXLE MNT RH	3.50	0.550	1	HR HSLA 560/620	TBD	Stamped
46	REINF. BRKT ENG LH	3.50	0.200	1	ESA-M1A35-C	29	Stamped
47	REINF. BRKT ENG RH	3.50	0.250	1	ESA-M1A35-C	29	Stamped
48	REINF-TAPPING PLATE	3.50	0.100	1	ESA-M1A35-C	TBD	Stamped
49	REINF.-FRAME S/B CAP	2.50	0.680	2	HR HSLA 560/620	TBD	Stamped
50	SHOCK-MNT-BRKT	3.50	0.650	2	HR HSLA 250/340	TBD	Stamped
51	BODY_MNT_4	3.00	0.700	2	HR HSLA 315/390	TBD	Stamped
52	CM3-REINF-LH-2	1.20	0.400	1	HDG CR MILD 245/350 DQ	TBD	Stamped
53	CM3-REINF-RH-2	1.20	0.400	1	HDG CR MILD 245/350 DQ	TBD	Stamped
54	TRACK_BAR_BRKT	3.00	0.870	1	HR HSLA 250/340	137	Stamped
55	LWR-R-CNTRL-ARM-BRKT	2.50	0.290	2	HR DP 340/590	350	Stamped
56	UPR-R-CNTRL-ARM-BRKT	2.50	0.290	2	HR DP 340/590	426	Stamped
57	C/M #7	1.80	0.470	1	HDG HR MILD 205/280 DQ	TBD	Stamped
	Welds (Total Length = 24,000 mm)		1.500				
Total Mass (kg)			174.4				

APPENDIX F – ASSEMBLY LINE CONFIGURATIONS –BASELINE FRAME

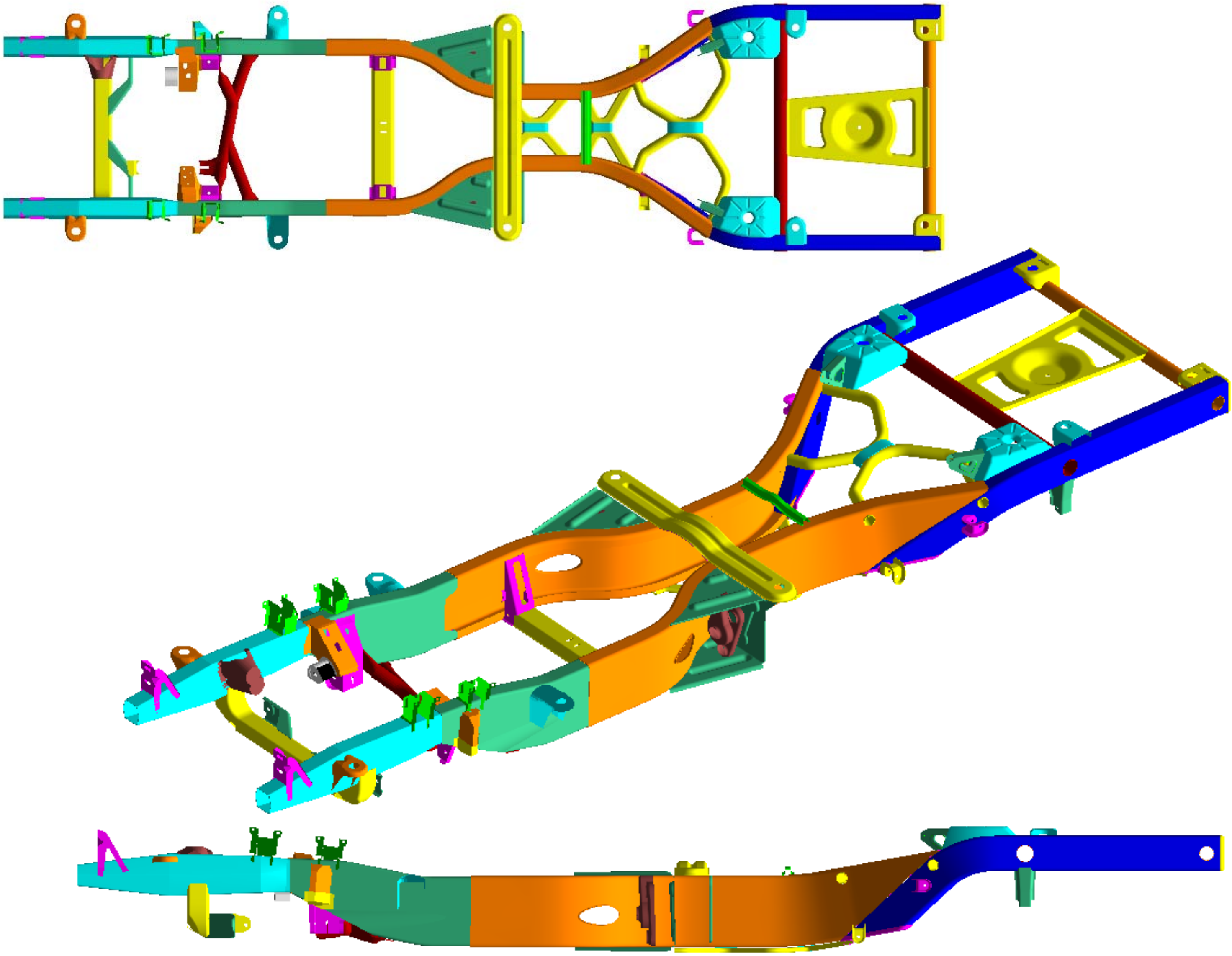


APPENDIX F – ASSEMBLY LINE CONFIGURATIONS – LIGHTWEIGHT FRAME



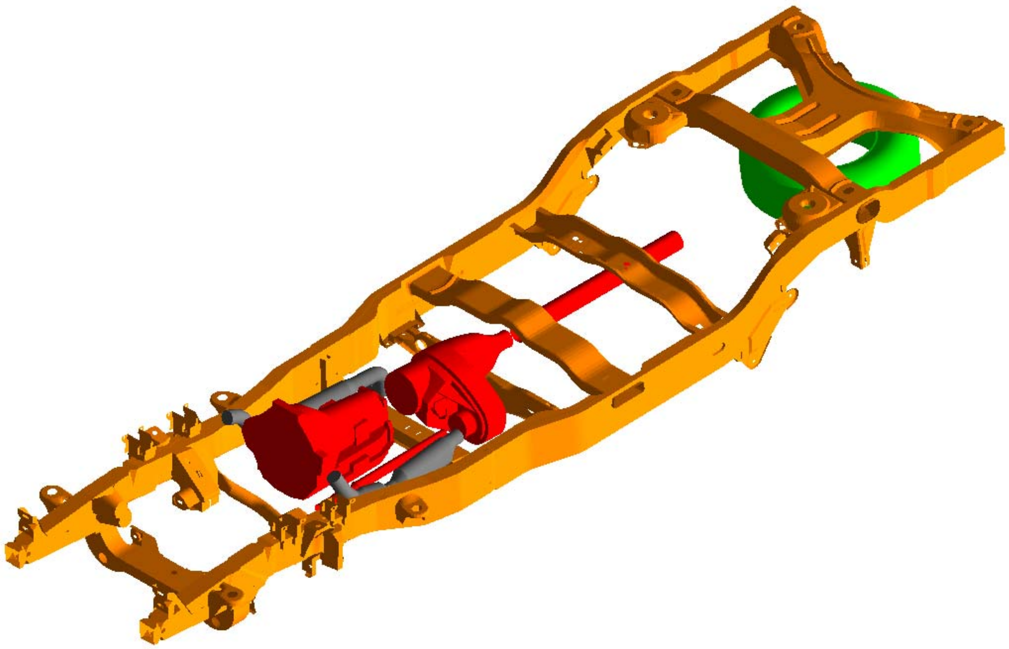
APPENDIX G – LIGHTWEIGHT FRAME CONCEPT

Lightweight Frame Images

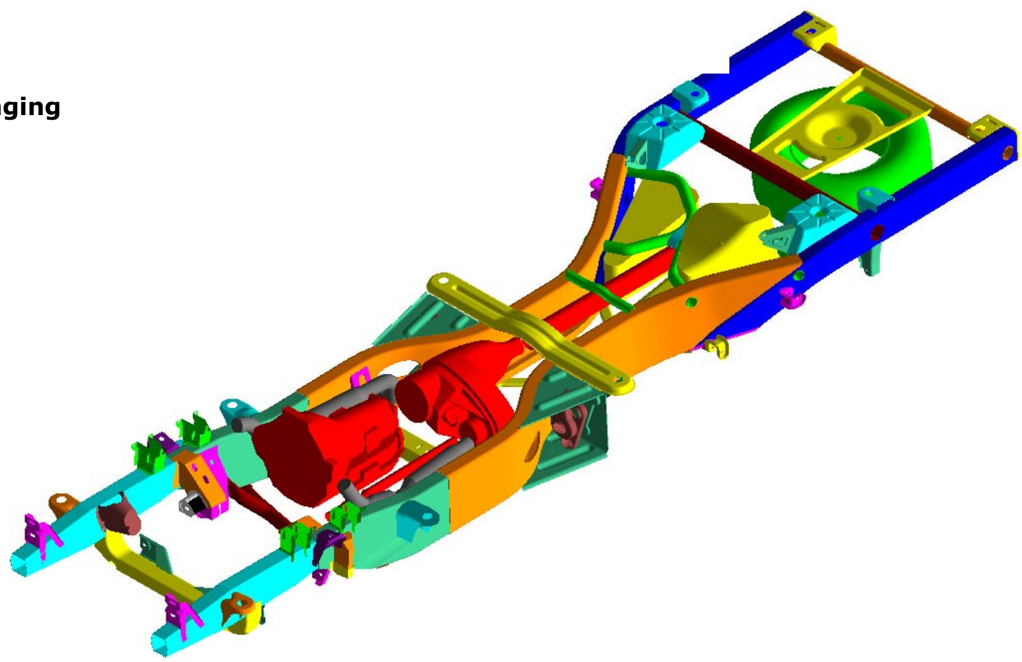


APPENDIX G – LIGHTWEIGHT FRAME CONCEPT

Packaging Comparison to Baseline Frame



Baseline Frame Packaging



Lightweight Frame Packaging