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Design of a Recumbent Seating System

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TABLE OF CONTENTS

ABSTRACT

Future space shuttle missions presented by NASA might require the shuttle to rendezvous with the Russian space station Mir for the purpose of transporting astronauts back to Earth. Due to the atrophied state of these astronauts, a special seating system must be designed for their transportation. The main functions of this seating system are to support and restrain the astronauts during normal reentry flight, and to dampen some of the loading that might occur in a crash situation. Through research, the design team developed many concept variants for these functional requirements. By evaluating each variant, the concepts were eliminated until the four most attractive designs remained. The team used a decision matrix to determine the best concept to carry through embodiment. This concept involved using struts to support during reentry flight and a spring-damper/shock absorber system to dampen crash landing loads.

The embodiment design process consisted of defining the layout of each of the main functional components, specifically, the seat structure and the strut structure. Through the use of MCS/pal 2 the design was refined until it could handle all required loads and dampen to the forces specified. The auxiliary function carriers were then considered. Following the design of these components, the complete final layout could be determined.

It is concluded that the final design meets all specifications outlined in the conceptual design. The main advantages of this design are its low weight, simplicity, and large amount of function sharing between different components. The disassembly of this design could potentially present a problem because of time and size constraints involved. Overall, this design meets or exceeds all functional requirements.

FIGURE LIST

TABLE LIST

INTRODUCTION

During the past few years, a series of joint missions between the Russians and the Americans have been proposed. The proposed missions involve the United States' space shuttle docking with the Russians' space station Mir. The space shuttle would rendevous with the Mir to replace three astronauts that had been living in the space station for a period of ninety days. Since these astronauts are significantly weakened due to the extended stay in a micro gravity environment, a special seat must be developed for the space shuttle to accommodate the three returning astronauts.

The special seat, known as a recumbent seating system, is meant to protect the astronauts from harmful levels of acceleration during normal flight and during emergency conditions. Normal conditions involve accelerations up to 12.5g while emergency landing conditions have a maximum acceleration of 20g. Since humans voluntarily take approximately 12g, a special seat must be designed to protect these astronauts during reentry flight. Since three astronauts will be coming down from the Mir, the seating system should accommodate three passengers.

The couch structure should withstand all the normal loads and remain intact upon landing. Since the shuttle has a limited area, this seat must be specially designed to fit in the middeck and can only attach to certain points inside the shuttle. This report proposes a design solution for a shuttle reentry couch. First, a brief description of the conceptual design process is given. This section presents the specification list, function structure, and the conceptual design process. The second section presents the embodiment design process. Embodiment design involves finalizing the specifics of the conceptual design. The final section discusses possible enhancements which could be integrated into the seat.

CONCEPTUAL DESIGN

The main objective of the design process is to develop a recumbent seating system (RSS) that restrains up to three fully suited space shuttle crew members. The RSS is located in the middeck and must support and protect astronauts that have experienced muscle atrophy due to an extended stay in orbit. The micro gravity environment causes muscular atrophy resulting in a 15-20% loss of muscle strength [Gannon, 1993]. Due to this weakened state, the astronauts require a special seat to protect them.

Specifications

The first step of conceptual design is to develop a complete list of specifications. These specifications encompass the functional requirements and constraints for the recumbent seating system. The specification list for the RSS, shown in Appendix A, is not meant to hinder the design team, but to aid the team in understanding the design problem. Several specifications are basic design wishes of NASA [Mongan, 1993]. For example, the seat must accommodate the weight and dimensions of three fully suited 95th percentile American males and 5th percentile oriental females for reentry and landing. This specification is based simply on NASA's desire for a seat to bring down three astronauts from the space station Mir.

The seat must protect these astronauts during the return flight to Earth. The tolerated acceleration levels are shown in Figure A-3. These levels are the acceptable accelerations the astronauts may experience [Sanders, 1987]. The astronauts must be adequately constrained preventing them from moving dangerously about the cabin during the flight.

Many of the specifications supplied by NASA were dictated by configuration of the middeck floor in the space shuttle. For example, maximum dimensions of the RSS are based on the configuration of the middeck (see Figure A-1). Also, the attachment points and their allowable force loading are established by the supporting structure underneath the middeck floor.

NASA also specified the normal and emergency conditions on the space shuttle. These specifications are in the form of load factors and are presented in Tables A-1 and A-2. These load factors are the maximum factors that are possible during normal and emergency circumstances.

Function Structure

The function structure is a visual tool that shows the primary functions of a design problem. The function structure for the RSS is shown in Appendix B. The energy flow is depicted by the solid lines and the material flow is shown by the double line. As shown by the function structure, the recumbent seating system has four sub functions. The first sub function is to secure the passengers. The next sub-function is to support the passengers during normal operational loads. The middeck floor is subjected to these loads during the reentry and landing portions of the flight. The next major sub-function is to support and reduce emergency landing loads. These are loads that occur during a crash situation. The last sub function is release the passenger. As seen by the function structure, energy comes into the system in the form of motion and leaves the system as motion and dissipated energy.

Solution alternatives

After the function structure is completed, the design team proposes solutions to each of the major functions seen in the function structure. Care was taken so that little bias is present in the process at this point. For the first function of secure passengers, a cushioned seat with a 5-point safety harness was the only solution variant (see Appendix C). This configuration is typical for similar applications such as jet plane seat, helicopter seats, and current space shuttle seats [Singley, 1972]. Similarly, the last function of release passenger had the same solution of a cushioned seat and 5-point safety belt.

The solution variants to support during normal operational loads and reduce emergency landing loads also shown in Appendix C. These various ideas were the result of research and brainstorming by the design team. As a sub-function of the two main sub-functions, the connection points were also considered by the design team. These floor connection points resulted in many different solution variants as seen in Appendix C.

Filtering solutions. These solution variants lead to 48 different combinations. However, some of the solutions can be quickly eliminated due to incompatibility or unfeasibility.

For the function of support normal operational loads, all of the solution variants were eliminated except for struts. The solid material is eliminated because it violated the weight constraint. Even though~~t~~ a solid material could resist all of the load, the size and weight would be too high. The design team eliminated the liquid medium because of weight, size and environment compatibility problems. The liquid would be heavy, difficult to store, and present potential problems in the micro gravity environment.

The gaseous medium is eliminated due to time constraints because an air support takes too long to fill with gas. The magnetic field is eliminated due to weight [Bell, 1988]. Connecting the seat directly to the floor is removed because it was not compatible with the second major sub-function. After the design team eliminated the unusable alternatives, struts are the only remaining function solution.

Several solution variants to the function of support and reduce during emergency load condition are also easily eliminated. First, the motion of the passengers is eliminated due to geometry constraints. The restriction imposed on the operational space shown in Figure A-1 leaves approximately 6 inches to move in the forward direction. A solenoid, or magnetic damping, is removed due to the power constraints imposed by the space shuttle [Bell, 1988]. Particle damping is eliminated because this involves extruding particles such as sand. This concept proves to be heavy and presented possible compatibility problems with the struts. Finally, active damping was eliminated due to the complexity of the setup and the power constraints.

At this point, a dominance matrix is performed to determine the type of connection the RSS would use to attach to the floor. The matrix as well as the decision criteria is shown in Appendix C. After this matrix, the design team chooses a clevis joint based on its ability to handle the stress concentrations.

Combining solutions to form concepts. After the aforementioned solution variants were eliminated, four combinations remain. A depiction and proof of feasibility of each of these concepts are shown in Appendix D. The first combination was struts with crushable material. This concept uses crushable material to dissipate the kinetic energy of the seat [Ellis, 1961]. The main disadvantage of this concept is that the struts are bulky thus creating future storage problems.

The second idea that remained is the struts with a spring damper. This concept is similar to a shock absorber that is found on a typical automobile. This concept is commercially available and is compatible with a number of different configurations. The main disadvantage of this concept is the possible complexity involved in the design.

The third concept that remained after the elimination process is the pressfits. Pressfits are high pressure interfaces between two surfaces. This interface resists motion until a large enough force is applied. Once the high force is reached, the surfaces begin to move with damping provided by friction caused by the contact. The main drawback of this design is the weight. All the material must be constructed out of metal thus causing the high weight.

The fourth combination was the use of airbags. This concept is similar to the airbag found in the steering wheel of an automobile. This idea proved to be quite complex and only dampened in one direction.

Judging the Concept Variants

The design team uses a decision matrix to judge the remaining four concepts (see Appendix E). The criteria used to judge the concepts is chosen on the basis of its relevance to the initial design problem. Since the acceleration felt by the astronaut is directly related to the health of the astronaut, this criterion receives a relatively high weight. The reliability of the device encompasses complexity of the design as well as how likely the device is to work the same each time. The setup time and the weight are chosen because this device is to be used on the space shuttle.

The ranking from the decision matrix indicates that the spring-damper system is the best possible route. This concept is readily available and provides the highest possible damping because the shock absorber can oscillate while the other three concepts required that the motion be dampened in one motion.

EMBODIMENT DESIGN

The embodiment design process for the RSS begins with the selection of specifications that are crucial to achieving a satisfactory design. The constraints chosen as most important included geometric, force and operational specifications.

The first important specification involves the geometry of the seat and the operational space. The seat is confined to the space outlined in Figure 1 [Mongan, 1993]. The struts supporting the seat can only attach to the points shown in this figure. Points 1-8 can take all tension and compression loads while points 9-12 can take only 80 pounds in tension and compression. In addition, all points have a maximum shear rating of 5000 pounds. These attachment points are crucial to the design of the strut configuration which, in turn, determines the layout of the entire seat.

The specification stating that the weight of the RSS must be less than 180 pounds becomes important when deciding on materials for the different components. This constraint also forces the design to be simple and use the least amount of material possible.

The center of gravity of the RSS with occupants must be less than 16 inches above the middeck floor. As the center of gravity is lowered, the design improves because less bending moment is created by the couch. A lower bending moment reduces the stresses in the struts and in the couch supports.

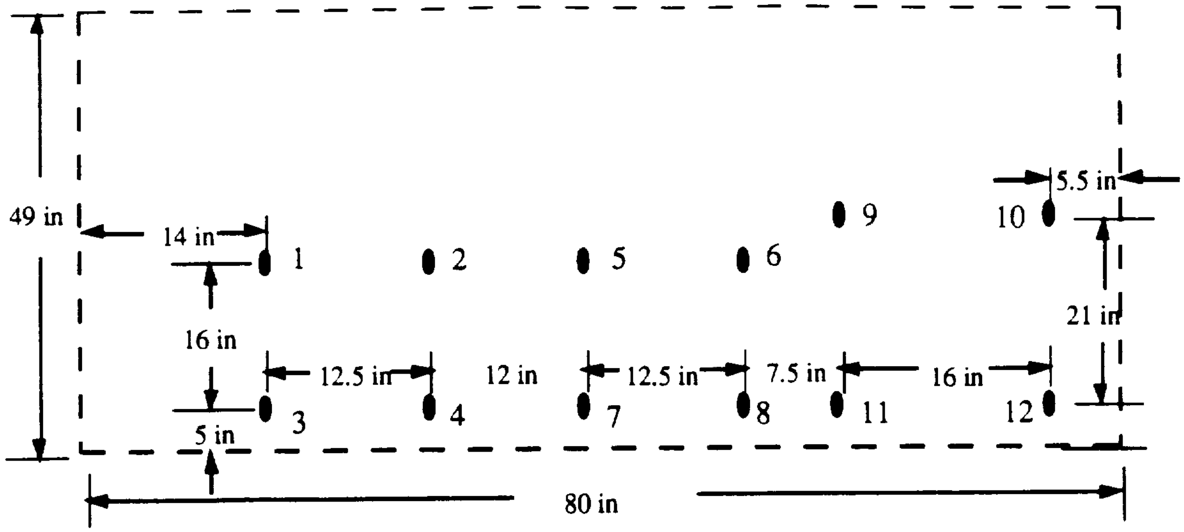


Figure 1. Operational space of the middeck area.

The functional requirement stating that the RSS must dampen the maximum 20g load factor to 11.2g governs both material selection and damper selection. This requirement also contributes to the geometry of the struts and seat supports.

The ability of the RSS to accommodate different size users from the 95th percentile American male to the 5th percentile oriental female while in full suits with parachutes is also a crucial specification. This requirement helps define the configuration of the seat surface and the geometry of the seat supports.

The specification concerning disassembly dimensions is ignored in the first part of the embodiment design. This simplifies the design structure. Suggestions for disassembly are discussed after presentation of the final design.

Main Functional Components

The RSS is divided into two major components that are treated separately. In the analysis of each component, optimization of configuration and selection of materials are both addressed. The main components are seat structure, and struts with spring/dampers. These components are shown relative to each other in Figure 2. This figure also defines planes of view that are referred to throughout the embodiment design process.

Seat Structure. Analysis of the seat structure begins by dividing the seat into sub-components that include the seat frame, seat panels, cushioning, and restraints. Each sub-component is considered separately and optimized before the whole structure is combined.

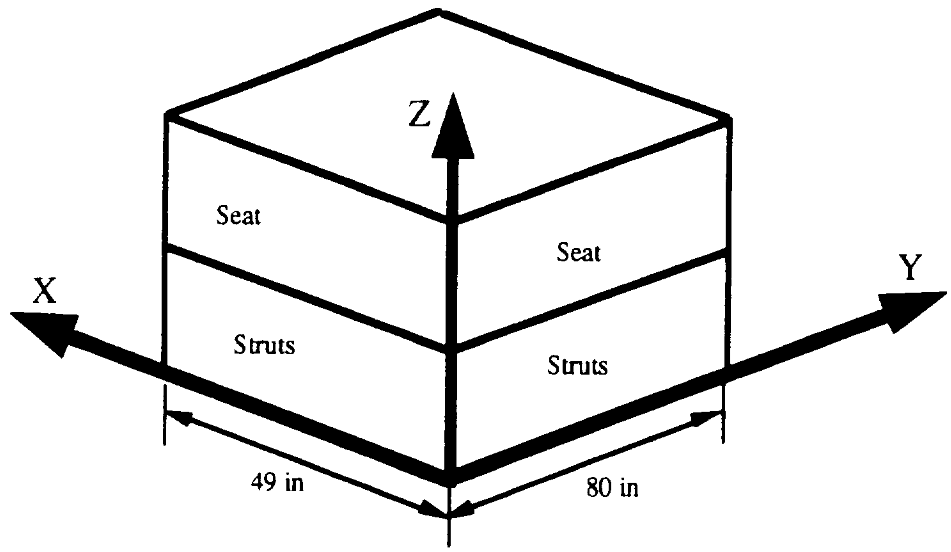
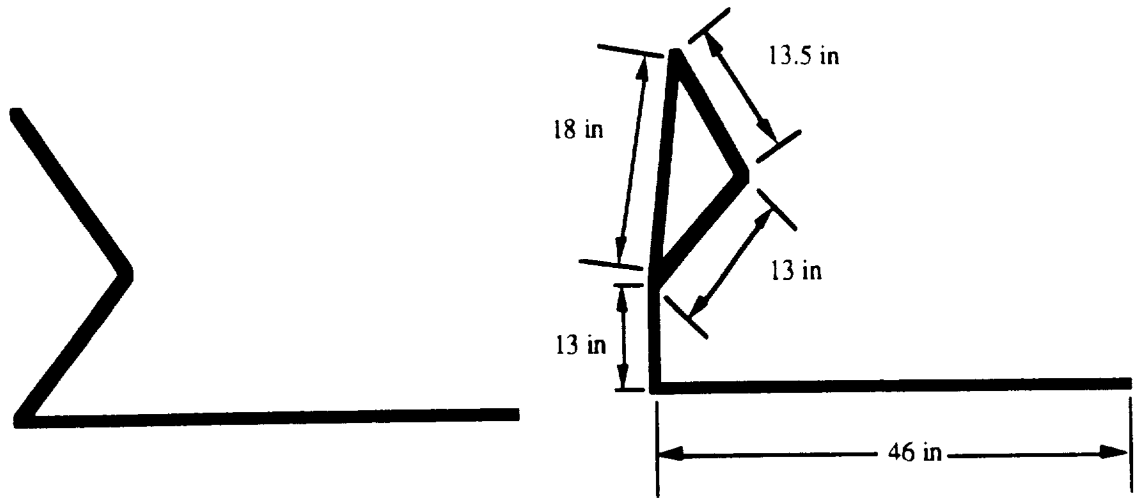


Figure 2. Three dimensional view of main components.

Since the operational space provides ample room in the z (upward) direction and the y (left-right) direction, the x (forward-aft) direction proves to be the most important dimension for analysis. For this reason, determination of the seat frame configuration began with the x-z plane.

The design must accommodate the dimensions of both the largest male and smallest female while they are in the pressure suits with parachutes. Calculations for these dimensions are included in Appendix F. The initial design for the x-z plane is shown as Figure 3a. While this design is adequate, a drawback exists in the pinch point at the hips of the passengers. This configuration does not make efficient use of the operational space because the passengers hips will not be able to reach the end of the back support.

The final concept for the x-z plane view is shown in Figure 3b. The addition of a flat bar at the base of the design provided optimum use of the operational space by allowing the passengers hips to reach the end of the back support. Should the seat ever rotate counter clockwise, the bar supporting the lower leg will hit the middeck lockers in a uniform fashion, thus increasing the safety of the design. Analysis and feasibility proofs of the decisions leading to this final design appear in Appendix G.



a. Initial profile b. Final profile

Figure 3. Designs for the x-z plane view.

Next, the design team analyzed the framework of the back support. The main purpose of this structure is to support the back and to allow easy connection to the struts and the restraints. Three different profiles were considered, and are shown below in Figure 4.

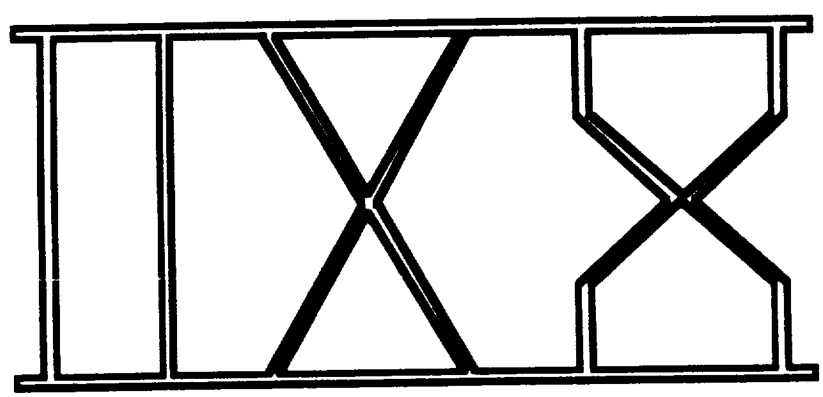


Figure 4. Variants for the Back Support of the RSS

Of these three profiles, the second configuration is optimal. This design provides better support underneath the passengers' center of gravity than the first concept. The third concept, while very similar to the second, has added stress concentrations at the angled joints.

Final revisions made to the optimum design are shown in Figure 5. Since the passenger restraints must be attached to the back structure, bars are added for easier connection. The bar added across the structure also improves the support of the passengers' center of gravity should the maximum downward load of 12.5g occur. The final width of the back support is 80 inches reflecting the shoulder breadth of three 95th percentile American males in space suits (26 inches each) and 1 inch clearance between each passenger for comfort.

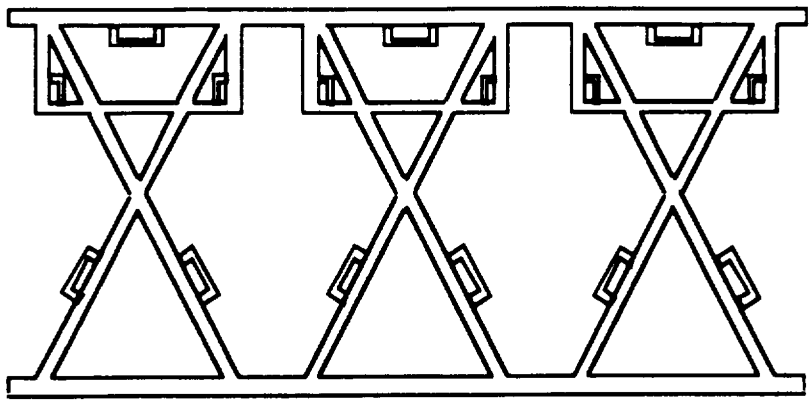


Figure 5. Final design for back support.

Next, the vertical structure located behind the passengers hips is considered. During the maximum loading of 20g in the forward direction, this section of the seat will be subjected to loading as the passengers slide towards it. For this reason, the configuration of this portion is made similar to the configuration of the back section by employing crossing bars to provide the maximum support (see Figure 6).

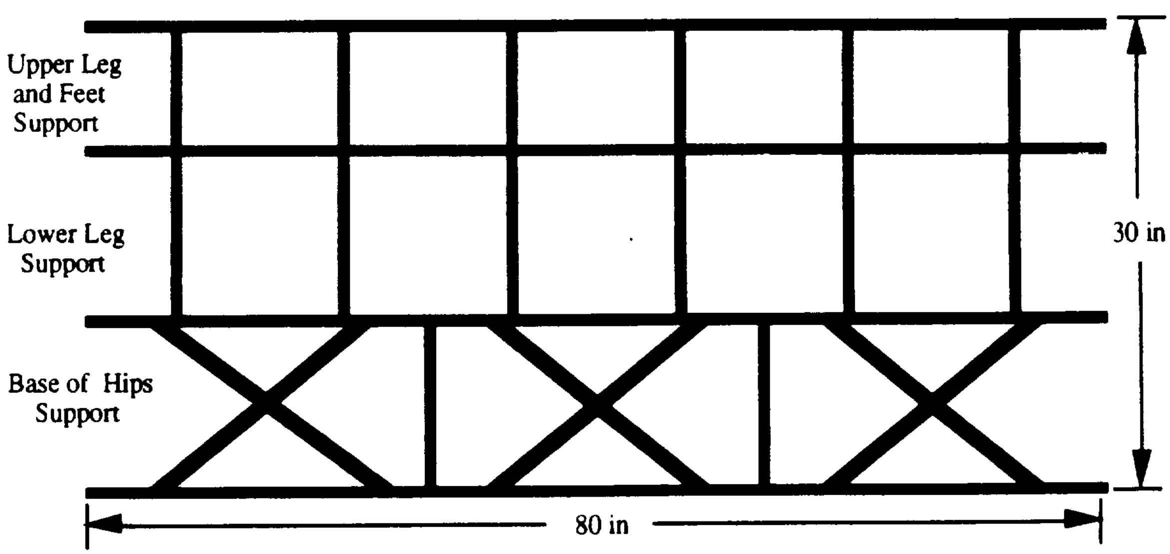


Figure 6. Design of Vertical Structure of the Seat

The area of the frame supporting the lower and upper legs is subjected to only minimal loads because little weight is distributed over these areas. Due to this fact, and the wish to conserve as much material volume as possible, the bars in this section are straight (see Figure 6).

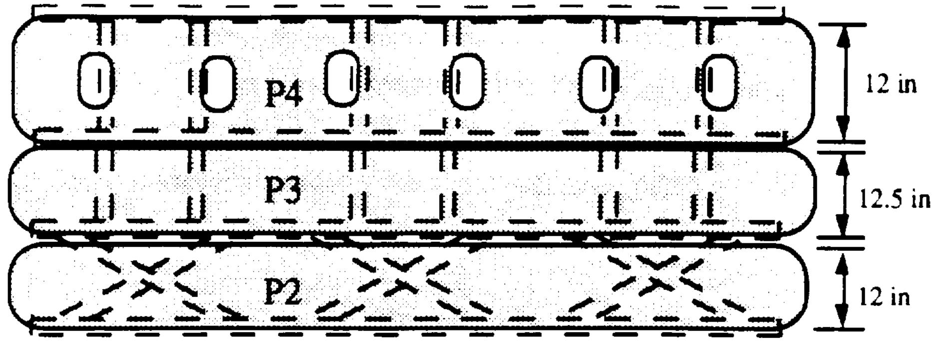
Since the complete framework of the seat has been determined, attention is given to the actual bars the make up the structure. All bars that connect to the seat panels or struts should have a square cross-section to allow for the easiest and most secure connections. The cross-sectional dimensions of the bars that make up the rest of the framework will be discussed later in the embodiment design.

The material chosen for the seat framework has three main requirements. First, it must have the capability of carrying loads in both the axial and transverse directions. Given the volume of material that is required, it must also have a relatively low density in order to meet the weight constraints. Finally, the strength of the material must be high enough to not fail under the loads that will be experienced.

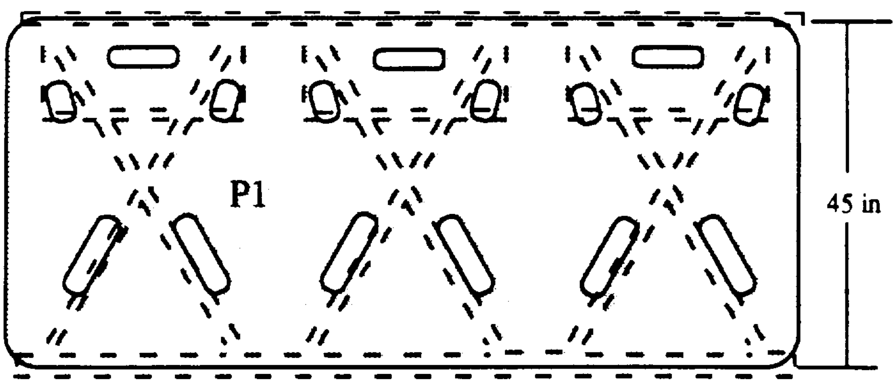
Three materials are initially considered: boron epoxy, aluminum, and carbon steel. The approximate volume of material needed is calculated in Appendix G as 530 in^2. Since carbon steel has an approximate density of 0.28 lb/in^2, the total weight of this structure will be approximately 148 pounds [Juvinall, 1991]. Carbon steel is rejected as a possible material because this value is over 80% of the allowable weight of the entire RSS. The largest load that the structure will experience in the downward direction is 12.5g. This force creates a transverse load on the back support of 13,500 pounds. Since the boron epoxy has a maximum transverse tensile strength of 8.9 ksi , the back support will not achieve the required 1.4 safety factor [Lee, 1991]. The approximate weight of the structure if 7075 aluminum is used as the material (density of approximately 0.1 lb/in is 53 lbs [Juvinall, 1991]. Since this material has an acceptable weight, and the yeild strength is 78 ksi, aluminum is chosen.

Seat Panels. The design of the seat panels begins with four main considerations. First, the geometry must efficiently provide adequate support of the passenger's body over the seat frame, while minimizing material volume. Second, the panels must be strong enough to support the maximum loadings. Third, the panel material must have a relatively low density to minimize the weight of the RSS. Fourth, the panel geometry and material must allow for simple and secure attachments to the seat framework and the overlaying seat cushion.

Since the final seat configuration has been specified, the dimensions of the panels required are easily obtained. Since the passenger restraints must connect to the seat framework, holes are added to allow the belts to pass through the panels. There will be a total of four panels. One panel supports the body on the horizontal support structure and three support the legs on the vertical support structure. Diagrams of each of the panels with dimensions are shown in Figure 7.



Vertical seat panels



Horizontal seat panel

Figure 7. Dimensions of seat panels.

The width of each panel is detemined from the total seat width of 80 inches. This choice eliminates any risk of injury to the passengers by getting pinched in gaps between panels. The dimensions of each panel are chosen from the dimensions of each section of the seat structure (see Figure 3). These panel dimensions provide suficient support to all areas of the seat where high inertials loads will be acting.

To determine the thickness of each panel, the team considers various materials with high strength-to-weight ratios . The materials to be examined are aluminum and fiber-reinforced composites (graphite epoxy and aramid epoxy). The aluminum alloy provides significant support, but has a significantly higher density than the composites. The high impact strength of aramid epoxy composite compares favorably with aluminum and graphite epoxy. Aramid epoxys also have a relatively low density of approximately 0.055 lb/in^3 [Lubin, 1982]. Finally, aramid epoxy is chosen as the panel material since the material is commonly used in aerospace panel applications [Lee, 1991]. Araid epoxy passes NASA's requirement for off gasing.

Using aramid epoxy, the thickness of panels P4 and P3 is set at 0.1 inches to provide adequate leg support. The thickness of panels P2 and P1 is 0.125 inches because these areas are subjected to higher loads. Epoxy glue is used to connect the seat panels to the seat framework. Epoxy glue has a shear strength of 6500 psi [Avallone and Baumeister, 1978]. This type of permanent connection creates much smaller stress concentrations at the joining interface compared to other fixed connections, such as rivets or bolts.

Cushioning. The cushioning for the RSS is selected from padding materials that help support the passengers in a secure and comfortable position during the reentry flight. One type of cushioniong is a constant stiffness material, such as neoprene, isoporene, or flexible polyurathane foam. This type is often used in car seats. It works well in low-speed impacts and elastically deforms around the passenger creating a form-fitting support.

A second type of cushioning is a constant force material, such as expanded foam polymers, balsa wood, or hone: .ıb aluminum. This type of material yields at an approximately constant stress and works well for high-speed impacts. Also, constant force material helps distribute the load more uniformly onto the seat panels [Daniel, 1989].

To insure crash-worthiness, a combination of the two cushioning materials is chosen [Farenthold, 1993]. A layer of the constant stiffness material placed on top of a layer of the constant force material utilizes the advantages of both materials. The double-layered cushioning acts like a spring/damper system and attennuates some of the energy created by shock loading. Neoprene is chosen for the top layer while the bottom layer is chosen as balsa wood. This decision is based on off-gassing constraints.

To keep the center of gravity low, the cushion thickness is chosen as one inch. This thickness provides adequate cushioning and support for the passengers. Also, the double layered cushioning has a woven nylon covering that attaches the cushion to the seat panels.

Restraints. The passenger restraint chosen for the RSS is specified by NASA as a 5-point seat belt [Singley, III, 1972]. The seat belt is connected to the horizontal seat framework at the points shown previously in Figure 5. Additional restraints on the passenger's feet are required to prevent the legs from shifting during the reentry, flight, and landing. These feet straps are to be attached to the points shown on the lower leg section of the framework in Figure 8. This location accomodates the largest male and smallest female that will use the seat. The material chosen for the all restraints is nylon fabric. The restraints are sewn around the seat framework.

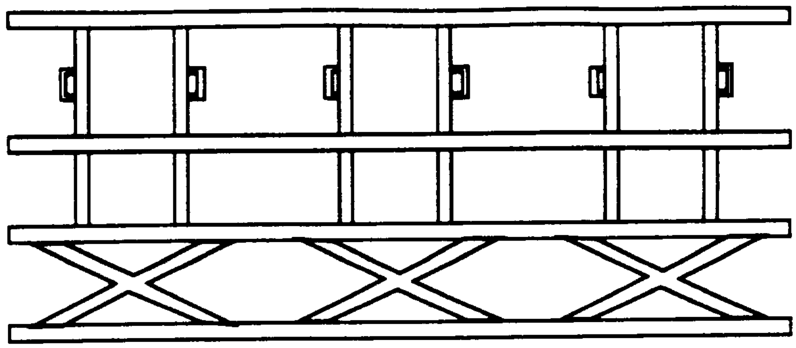


Figure 8. Location of restraining belt connections on the vertical seat structure.

Struts. To determine the optimal strut configuration, several assumptions and initial calculations must be made. First, the maximum height, h, of the seat is approximately 6.5 inches. This height is dependent on the center of mass for the person, estimated center of mass for the seat, and the maximum height of the total center of mass. The calculation for the center of mass for a person in a crouched position is shown in Appendix F. [Damon, 1966]. Using this information along with the system center of gravity of 16 inches, the approximate seat height is found. The calculations in Appendix F use the simple relation for center of mass shown in Equation 1. M_T is the total mass and m is the mass of each individual component.

$$M_T\bar{z} = \sum_{i=1}^{n} m_i\bar{z}_i \qquad (1)$$

The second calculation that was made was to calculate the length of the seat (see Figure 3b). As stated earlier, the length, L, of the seat is 46 inches.

Before the configurations of the struts were analyzed, the design team made one final assumption. The crucial factor in determining the strut configuration is the 20g crash loading. Consideration is given to this forward load because the 20g crash loading is significantly larger than the other loads. While the other loads are important, the struts will probably handle the normal loads. However, if this criteria is not met, the struts can be easily altered to handle the other flight loads.

To analyze the various configurations presented in this section, MSC/pal 2 is used. MSC/pal 2 is a stress analysis software package sold by MacNeal-Schwendler Corporation. The software uses a finite element method to calculate displacements, forces, and stresses for two and three dimensional systems. After the simulation is performed, pal 2 gives information ranging from axial forces in each member, force at each connection point, stresses, and percent yield.

For points 1-8, four different configurations for the x-z plane, or side view, are given consideration in the embodiment design process (see Figure 9). For the second variant, the shear rating exceeds the allowable 5000 pounds at most of the connection points thus eliminating the variant. The third and fourth variations are eliminated because the configurations require additional struts for stability. The first variant is accepted because it requires the fewest supporting struts, and uses the simplest and most direct connections.

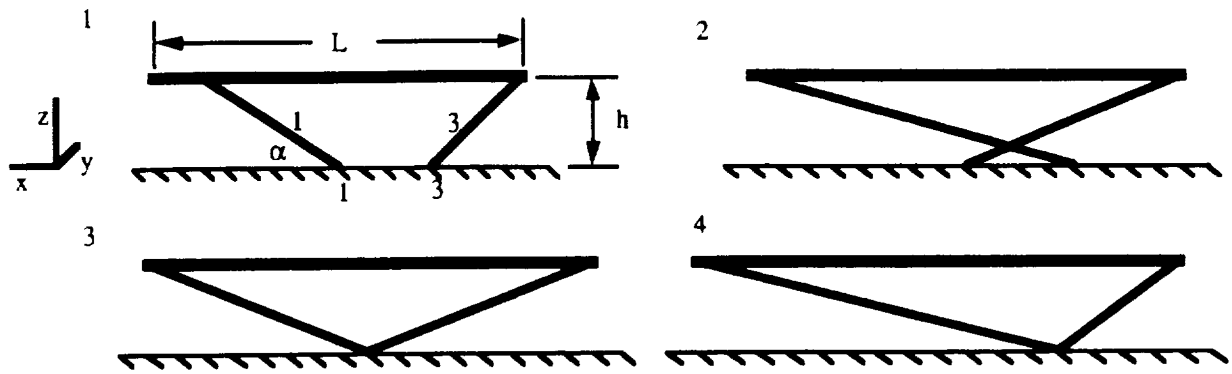


Figure 9. Strut configurations for the side view (x-z plane).

This first configuration is the most feasible and advantageous of the four choices. This choice minimizes the weight and is the most stable setup. The height, h, of the struts was determined from the center of mass and the total length, L, was determined by the body dimensions. To finalize the geometry in this configuration the actual angle, α, of struts 1 and 3 must be determined. The strut number corresponds to the floor connection point to which it is attached. Using pal 2, strut 1 is analyzed for a range of different angles. The coding for this simple two dimensional model is included in Appendix H. The simulation results for an arbitrary horizontal loading, shown in Figure 10, demonstrate that the angle, α, should be minimized to reduce the axial stress in the bar. The same analysis is also be applied to strut 3. As a result, the struts extend from the floor connections to the farthest points on the seat base.

The next view to be analyzed is the rear view. Four different arrangements are considered for analysis. These variants appear in Figure 11. These side struts prevent motion of the RSS left to right and provide support for compressive and tensile loading above points 9-12. These side struts also reduce the amount of bending stress in the main struts contained in the x-z plane. The third and fourth variants are eliminated because these configuratons do not provide symmetrical support in both directions. The first and second designs are both acceptable, but the first is chosen over the second because it minimizes size and weight.

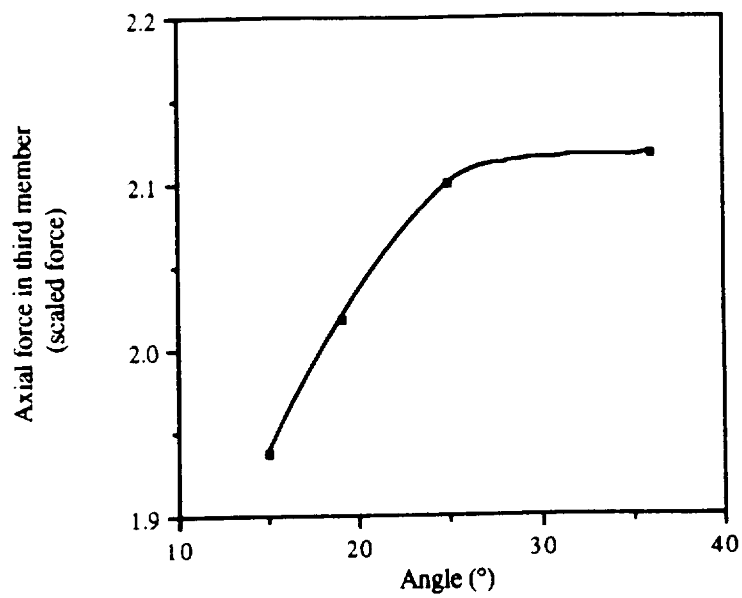


Figure 10. The force in the strut increases as the angle increases.

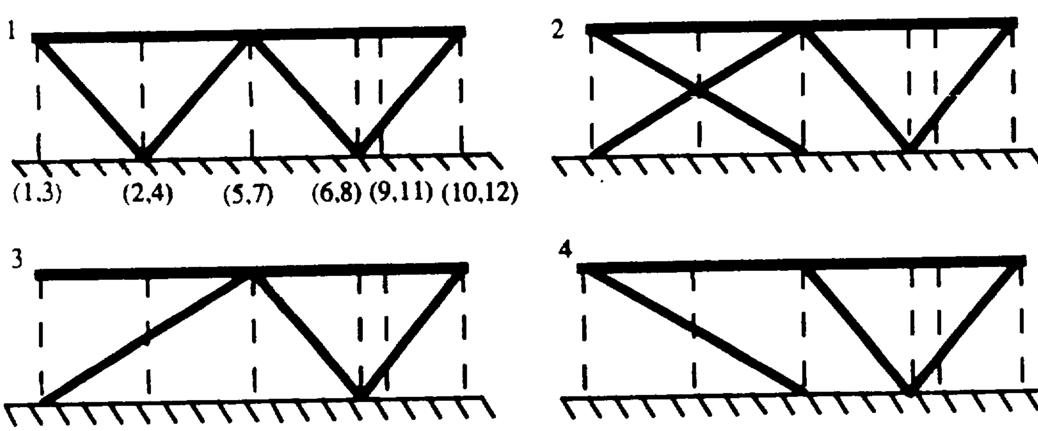


Figure 11. These side struts help carry the side load and any moments.

Now that the general configuration for the struts is chosen, the three dimensional setup is put into pal 2. Both the struts and the underlying seat structure were input into pal 2 so a more accurate simulation could be made. As mentioned earlier, the thickness of the back support tubing is also analyzed using pal 2. Since pal 2 gives the force in each member, an acceptable material and cross-section is then chosen. The team begins with the struts of hollow tubing with an outer diameter of 1.6 inches and an inner diameter of 1.4 inches. Intitially, this simulation contained only circular tubing. Hollow tubing is chosen because a hollow tube handles bending moments better than a solid tube [Gere, 1990]. Hollow tubing also minimizes the use of material. The initial dimensions are chosen by estimating a required area for handling a 20g forward load. Initially, the entire structure is constructed of hollow aluminum tubing. Aluminum 7075 is chosen because of its low density and its common use in aircraft and reentry vehicles [Lubin, 1982]. Aluminum also has uniform properties in all directions.

The criteria used to judge the struts is percent yield. Percent yield is the ratio of maximum stress to yield strength. For example, if the stress in the element is half the yield stress, the percent yield is 50%. Until each member does not fail this criteria, the dimensions and material are iterated This data insures that the struts do not fail. Also, the team made sure that the forces at the floor connection points were not greater than the allowable levels.

The final coding used for pal 2 is included in Appendix I. A three dimensional depiction of the model inputted into pal 2 is shown in Figure I-1. After running the initial simulation, it became evident that several of the elements were failing in shear. Struts 1, 2, 5, and 6 were failing because of a large axial stress. The couch wanted to rotate counter clockwise due to the asymmetric configuration of the struts. To remedy this problem, two steps can be taken. Either the material can be changed or the cross-section of the struts can be changed. In order to minimize the weight of the design, the design team opted to change the material. The material in these four struts, as well as their crossbars, is changed to boron-epoxy. Boron-epoxy is a readily available composite material commonly used in aerospace applications [Lee, 1991]. Boron-epoxy also passes NASA's specifications for off gasing [Lubin, 1982]. The composite has a unidirectional tensile strength of 220 x 10^3 psi. In comparison, aluminum 7075 has a tensile strength of 83 x 10^3 psi. Also, the density of boron-epoxy is 0.075 lb/in^3. The density of aluminum is approximately 0.100 lb/in^3. The main drawback of the composite strut is that material carries primarily only axial loading. This property disallows the use of bolts and normal connections between the boron-epoxy and the rest of the structure. Pal 2 is then used to simulate the new structure. A factor of safety of 1.4 was used to insure that failure would not occur [Mongan, 1993]. The forces, as well as the percent yield, that resulted in these struts due to the 20g loading are shown in Table 1. The shear force at the 8 floor points are shown in Table 2.

Table 1. Forces in Struts for 20g Foward Load

Strut	Axial Load (Lb)	Percent yield
1	-1133	31%
2	-7685	53%
3	617	20%
4	-1014	54%
5	-2171	53%
6	-8406	84%
7	757	43%
8	-1061	15%

Table 2. Shear in Floor Points 1-8

Floor Point	Foward 20g	Downward 12.5g	Side 3.3g
1	56	256	1825
2	3422	1235	1883
3	596	378	159
4	978	1454	3
5	1100	2895	2051
6	3112	1850	1970
7	730	915	70
8	990	4369	177

During this iteration process, the team tried different configurations that used points 9-12 on the mid deck floor. However, the forces in these member results in tensile and compressive load that exceed the allowable load of 80 pounds. The shear pins discussed in the conceptual design did not allow the structure to dampen or deform without exceeding the maximum shear loading of 5000 pounds. As a result, these members are eliminated and the rest of the structure was fortified to compensate for the reduction of supports. Also, two struts are added to the right corners to help compensate for the loss of points 9-12. The final configuration for the struts is shown in Figure 12.

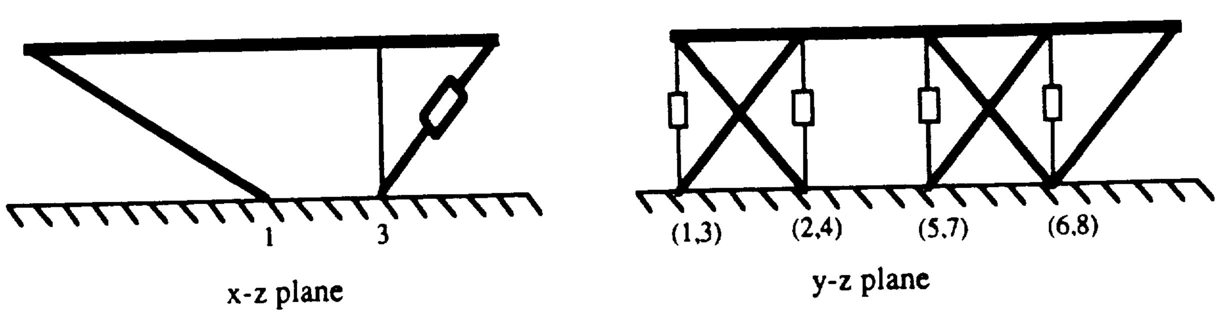


Figure 12. The configuration for each of the struts.

The final cross-section, shown in Figure 13, handles the maximum loading of 20g.

For the struts, the outside diameter is 1.6 inches and the inside diameter is 1.4 inches. Since the vertical and horizontal seat structure is square tubing, the cross-section of the tubing in the back support is different. The square cross-section allows easy connection to the panels and struts. To convert the dimensions from the circular tubing to the square tubing, the moment of inertias were set equal (Equation 2 and 3).

$$I_{o,circle} = \frac{\pi}{4}\left(d_o^4 - d_i^4\right) = \left(1.6^4 - 1.4^4\right) = .133in^4 \qquad (2)$$

$$I_{o,square} = \frac{1}{12}\left(h_o^4 - h_i^4\right) = \left(1.6^4 - h_i^4\right) = .133in^4 \qquad (3)$$

The resulting inside dimension of the square cross-secton is 1.5 inches. However, since the beams carrying a large amount of moment due to the asymmetric setup, the inside dimension is decreased to 1.4 inches to insure a higher degree of safety.

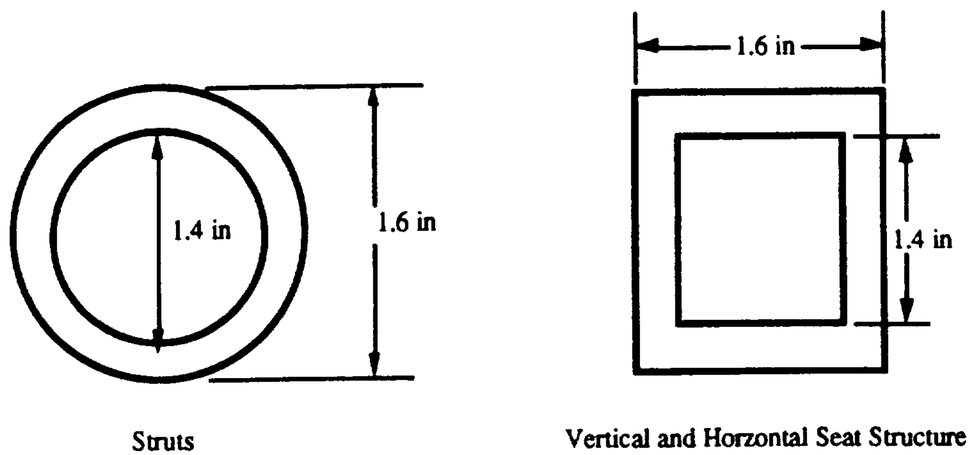


Figure 13. The final cross-section of each of the struts.

To fully test the structure, the maximum loading in the other directions are applied to the structure in pal 2. The results from these further simulations are also shown in Appendix I.

<u>**Shock absorbers.**</u> The design of the shock absorbing system of the RSS requires the fulfillment of three criteria. First, it must possess the capablity of sustaining the maximum force transmitted without the risk of collapse. Second, it must possess suffient energy absorbtion capacity to reduce the occupant's velocity to tolerable deceleration levels. Finally, it must fit any size constraints of the unit. From a finite element analysis using pal 2, the maximum axial loading on the supporting struts connected to floor points 1, 3, 5, and 7 occurs under a downward 12.5g condition. The three struts connecting to points 1, 3, and 5 undergo a maximum loading of 1500 lbs., while the strut connecting to point 7 undergoes a maximum loading of 4500 lbs. An uneven distribution of shock

loading on these supporting struts calls for shock absorbers of different capacities.

The selection of the proper shock absorbers requires the sizing 1 high capacity shock absorber and 3 lower capacity shock absorbers. The high capacity shock absorber requires an energy absorbing capacity of at least 13500 in-lbs. The lower capacity shock absorber requires an energy absorbing capacity of at least 4500 in-lbs. An acceptable stroke distance for the shock absorbers for the RSS configuration is no more 2.5 inches. The proper mounting of the shock absorbers requires the following special order requirements : (1) double acting dampers, (2) tension-compression shock absorbing, (3) light weight version, (4) pin connections at mounting points, and (5) a total body length of 7 inches.

From a search in manufacturer's catalog information, Taylor Devices Inc. supplies two shock absorber models from their h-series that meet the required specifications. A heavy -duty model with a maximum reaction force of 8000 lbs. and a max. energy absorbing capacity of 19200 in-lbs [*Thomas*, 1992]. Also, a smaller heavy-duty model with a maximum reaction force of 5000 lbs. and a maximum energy absorbing capacity of 8000 in-lbs.

Auxillary Functional Components.

In the conceptual design, clevis joints were selected to attach the struts to the middeck floor. To conserve weight, the material chosen for the clevis joint is aluminum. However the pin through the clevis joint must be strong to take all of the load on the strut. Therefore, the pin will be 4340 steel. Since the floor attachments are limited to a two inch diameter, the clevis joint will be bolted down with one bolt as shown in Figure 14. Part A of the clevis joint is attached to the bottom of the struts with epoxy glue. The dimensions of the clevis joint are based on the maximum forces that will be present in the members. These calculations are shown in Appendix J.

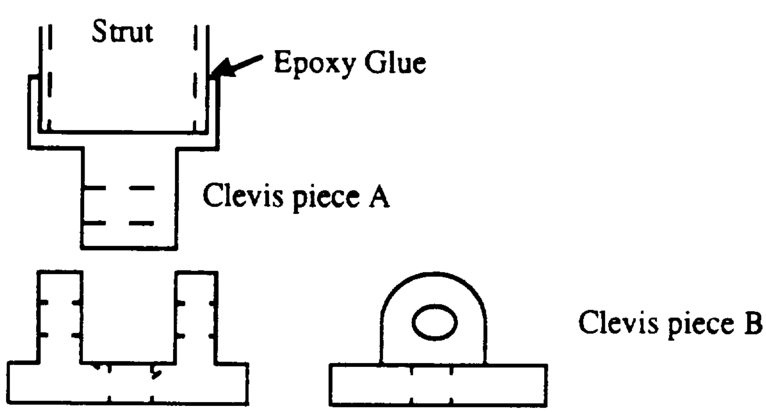
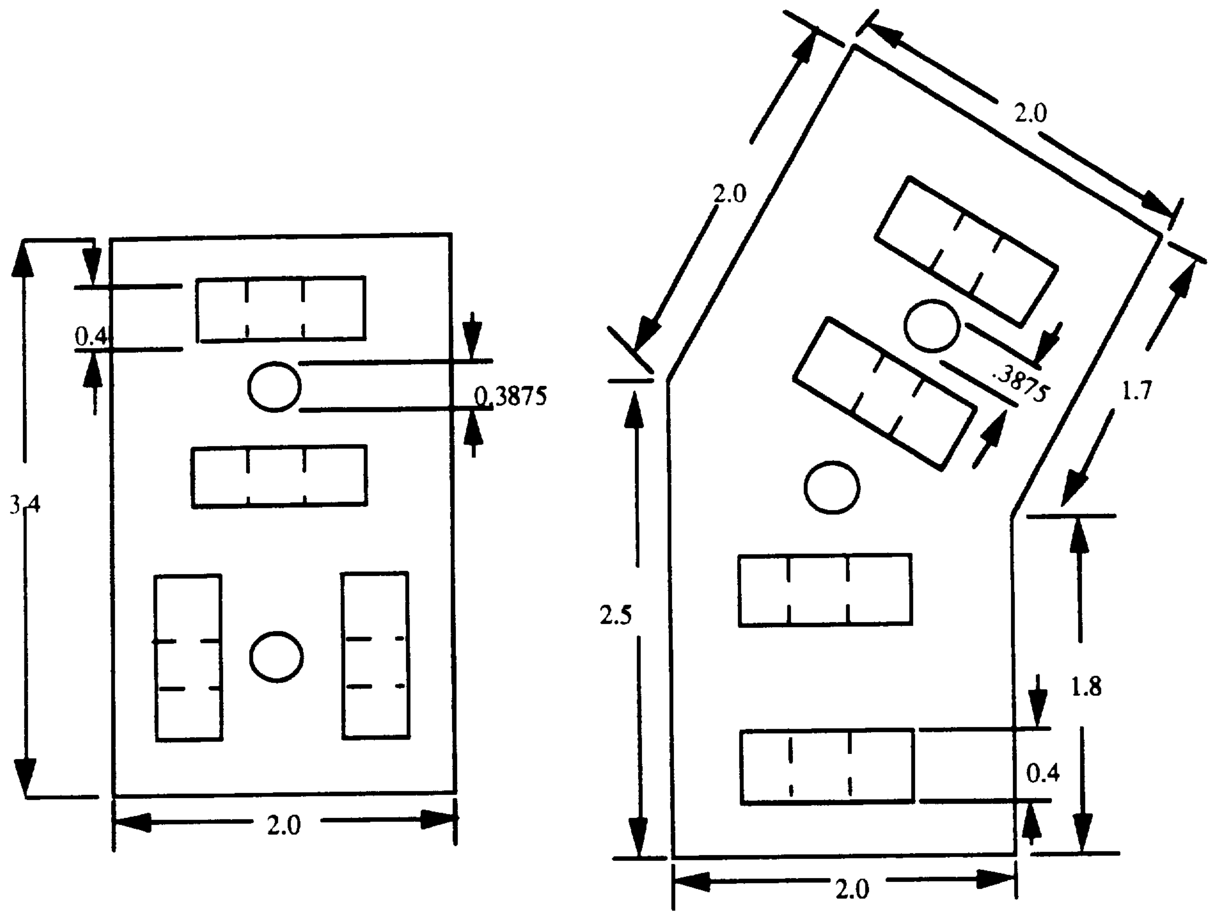


Figure 14. Clevis joints for points 1-5, and 7.

On the floor points 6 and 8, two clevis joints are needed because two struts contact the floor at these points. The configuration for these clevis joints are shown in Figure 15.



Clevis Joint for Point 6

Clevis Joint for Point 8

Figure 15. Clevis joints for points 6, 8.

To allow for motion of the dampers, clevis joints are used to attach the struts from floor points 3,4,7, and 8 to the frame. These clevis joints have the same dimensions as the clevis joints on the floor shown in Figure 10x. For struts connecting to points 1,2,5 and 6 a fixed connection is needed. The struts are made of boron epoxy and the frame is made of aluminum so designing the struts and frame as one piece is not possible. The solution to this problem is shown in Figure 16. The boron epoxy strut will fit inside the aluminum cylinder, and will be held in place with and epoxy glue.

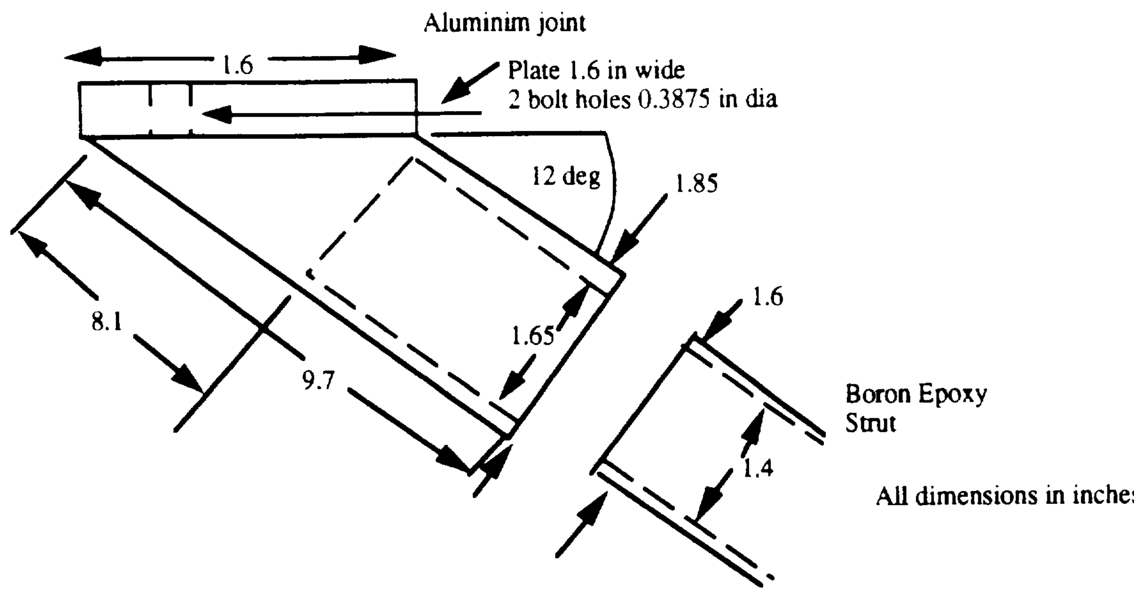


Figure 16. Connection of struts from points 1, 2, 5 and 6 to the base of the seat.

FINAL LAYOUT

The final design layout for the recumbent seating system is shown in Figure 17. This design meets the specifications except for the one concerning disassembly dimensions. The vertical leg support could be easily broken down into several components. The horizontal structure could be disassembled in the same manner. However, this adds to the assembly time. The natural frequency of the design is well above the required 30 Hz (see Appendix K). The final weight of the design is also well below 180 lbs (see Appendix K). Final design layouts of each component are shown in Appendix L.

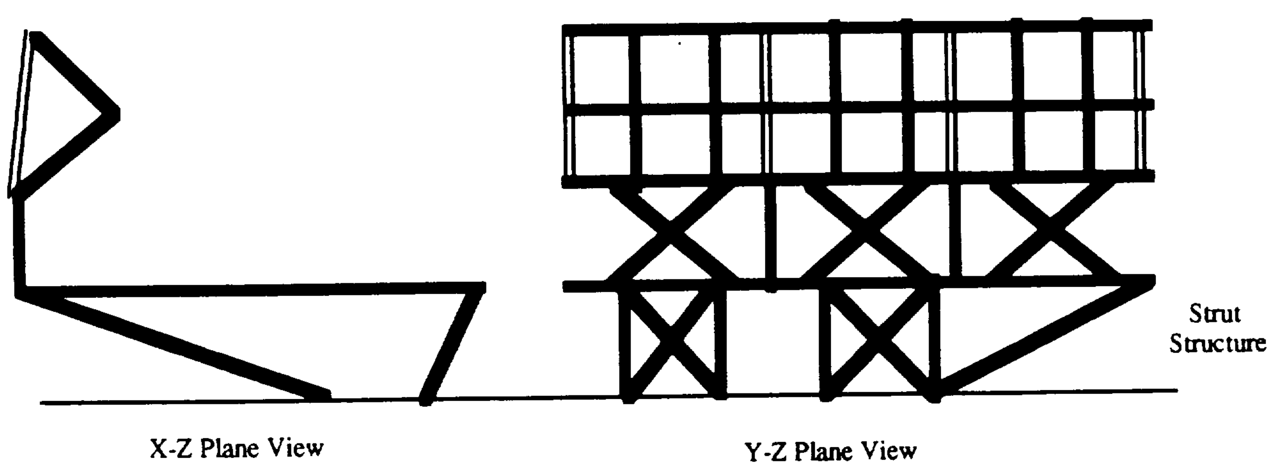


Figure 17. Final layout of the recumbent seating system.

CONCLUSION AND RECOMMENDATIONS

The final design of the RSS presented by the design team meets and many times exceeds all requirements set forth in the specification list. The strengths and weaknesses of this design are discussed in this section, as well as recommendations for future consideration.

This design has many advantages over other alternatives. The RSS fits easily within the operational space, and the weight is well below the specified maximum. The materials chosen are strong but lightweight, and they are commonly used in the aerospace industry. Also, the shock absorbers act as struts, thus sharing the functions of damping and support. Because the RSS does not require expensive materials or a large amount of machining, the cost of the structure should be low.

The major drawback of this design is that it is can not be completely disassembled. The use of the epoxy glue permanently combines some parts. This disadvantage could be rectified by increasing the weight restriction so that a material other than composite epoxy could be used for the panels and struts.

Although the design presented adequately performs the required functions, all of the floor points are not used. As a result, some of the floor points take a great deal more load than others causing an unequal force distribution in the structure. This problem could be minimized by creating a more complex structure that would distribute the forces more equally.

The final design reached by the team is based solely upon available data and a process of selection supported by quantitative decision making. Should more data regarding the functioning of the RSS or constraints on the RSS become available revisions would have to be made to the final design presented in this report.

Appendix A
Specification list for the RSS

		Recumbent Seating System (RSS)	4-15-93
F/C	D/C	Requirement	
		<u>Geometry</u>	
C	D	• Maximum storage dimensions are 2' x 1.5' x 3'	
F	D	• The center of gravity of the seat must be less than 16 inches from the floor in the deployed configuration with occupants	
F	D	• The center of gravity of the seat must be less than 6 inches from the floor in the stowed configuration	
C	D	• The seat must fit in area specified by Figure A-1	
C	D	• The seat may attach to the points shown in Figure A-1	
C	D	• The crew members head shall be positioned with the head aft and feat forward so that the head and thorax are in a plane bounded by the angle of 0° to 6° relative to the middeck floor	
C	D	• The legs may be bent	
C	D	• The floor at all points in Figure A-1 is aluminum and has a minimum thickness of 0.1"	
C	D	• The floor mounts must use a hole in the floor no greater than 2" diam.	
		<u>Kinematics</u>	
C	D	• The system natural frequency must be above 30 Hz	
		<u>Forces</u>	
F	D	• The seat must accommodate astronauts wearing pressure suits. The suit weighs no more than 100 pounds each	
C	D	• The seat must weigh less than 180 pounds	
C	D	• In Figure A-1, attachment points 1-8 take all tension and compression loads; points 9-12 take less than 80 pounds in tension or compression; all points have a maximum shear rating of 5000 pounds each	
		<u>Energy</u>	
C	D	• Device must operate on less than 5 amps (if necessary)	
C	D	• Device must operate on 28V DC (if necessary)	
		<u>Material</u>	
C	D	• Material must conform to restrictions specified in NASA-STD-3000. This standard specifies restrictions due to fire hazards and off-gassing	
		<u>Safety</u>	
C	D	• Device must not endanger the shuttle or astronauts	
F	W	• In the event of an accident, the occupants must be able to remove themselves in less than 30 seconds	
C	D	• No sharp edges or pinch points are allowed above the back cushion	

		Recumbent Seating System (RSS)	4-15-93
		<u>Ergonomics</u>	
F	D	• The seat must accommodate the weight and dimensions of three fully suited 95th percentile American males for reentry and landing. 5th percentile oriental females must also be accommodated. The suits add a maximum of 6.5 inches to the seated height. The parachutes have maximum dimensions of 15" x 20" x 4". Also, the crew will experience growth of 3% in their seated height	
F	D	• Device must be comfortable to astronaut as specified in NASA-STD-3000	
		<u>Assembly</u>	
C	D	• Assembly of the RSS should take two trained astronauts no more than 5 minutes	
C	D	• Assembly of seat must be accomplished without tools	
		<u>Transport</u>	
C	D	• The seat in the stowed configuration must withstand typical liftoff vibration of 5 to 100 Hz for a period of 30 minutes	
C	D	• The seat in the stowed configuration during liftoff must withstand 3.3 G's for 30 minutes continuously	
		<u>Operation</u>	
F	D	• The seat must restrain and protect occupants during normal and emergency loads shown in Table A-1 and A-2. These loads are applied individually, not simultaneously	
F	D	• The seat must dampen acceleration felt by astronaut to the levels shown in Figure A-3	
F	D	• The seat must operate in the area shown in Figure A-1. The seat must not come within 1 inch of existing shuttle fixtures.	
C	D	• Seat must operate in an environment consisting of 50% relative humidity	
C	D	• Seat must operate in a temperate range: $65 < T < 85°F$	
		<u>Maintenance</u>	
C	D	• Free of maintenance for at least one shuttle trip (Maximum of twelve days)	
F	D	• Astronauts setting up the seat must be able to visually inspect the RSS	
C	D	• The seat must have a life of at least 100 uses	
		<u>Production</u>	
C	D	• RSS must be completed by December 1994	
C	D	• Minimum of one unit	

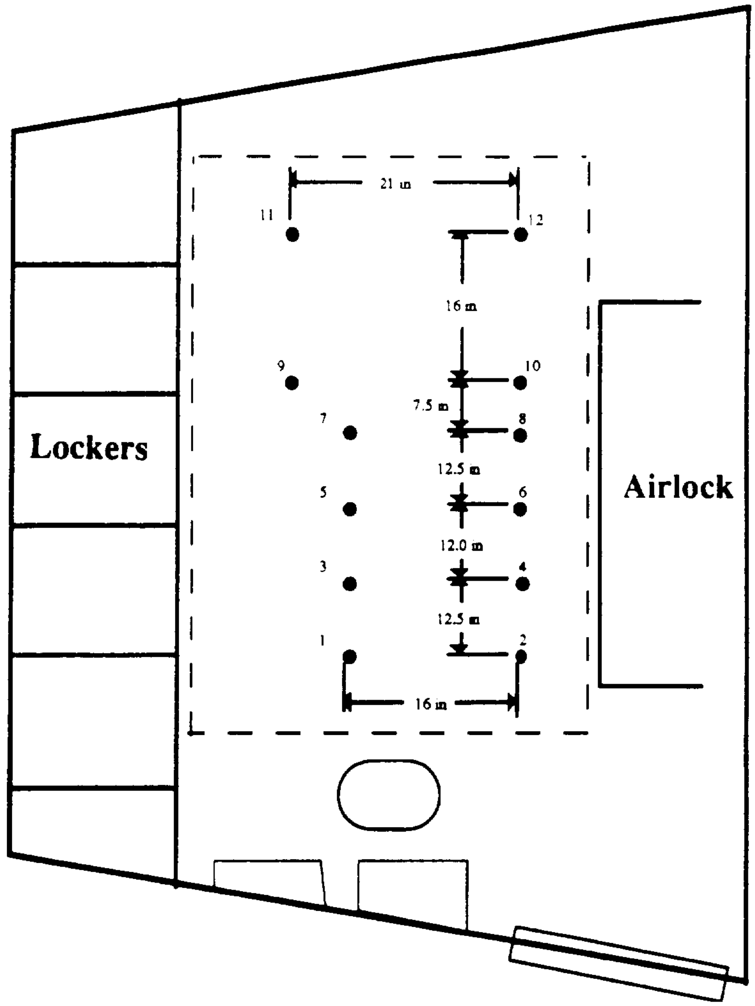


Figure A-1. Depiction of middeck floor. The middeck floor defines the operational space in the shuttle [Mongan, 1993].

Table A-1. Transient Response Load Factors (g's)

Load Factor	Limit load factors		
	X	Y	Z
Lift-off	±9	±3.2	±7.4
Landing	±6.25	±2.5	±12.5

Table A-2. Emergency Landing Load Factors

Ultimate Inertia Load Factors		
X	Y	Z
+20.0	+3.3	+10.0
-3.3	-3.3	-4.4

Note: For the emergency landing load factors: the longitudinal load factor (X) shall be directed in all directions within a 20° of the longitudinal axis.

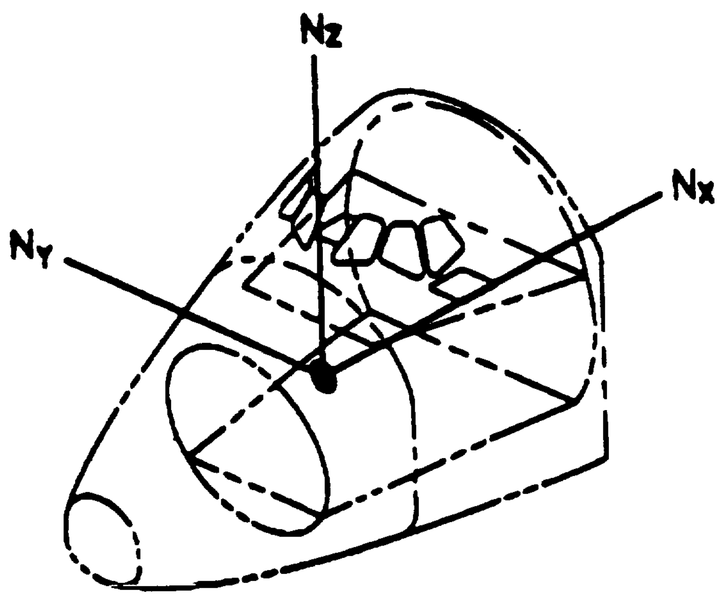


Figure A-2. Direction of load factors.

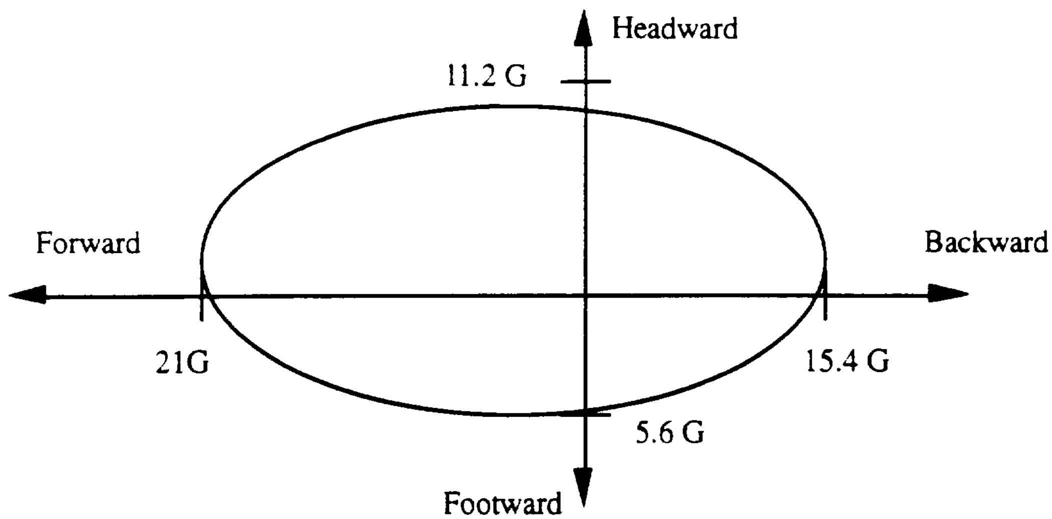
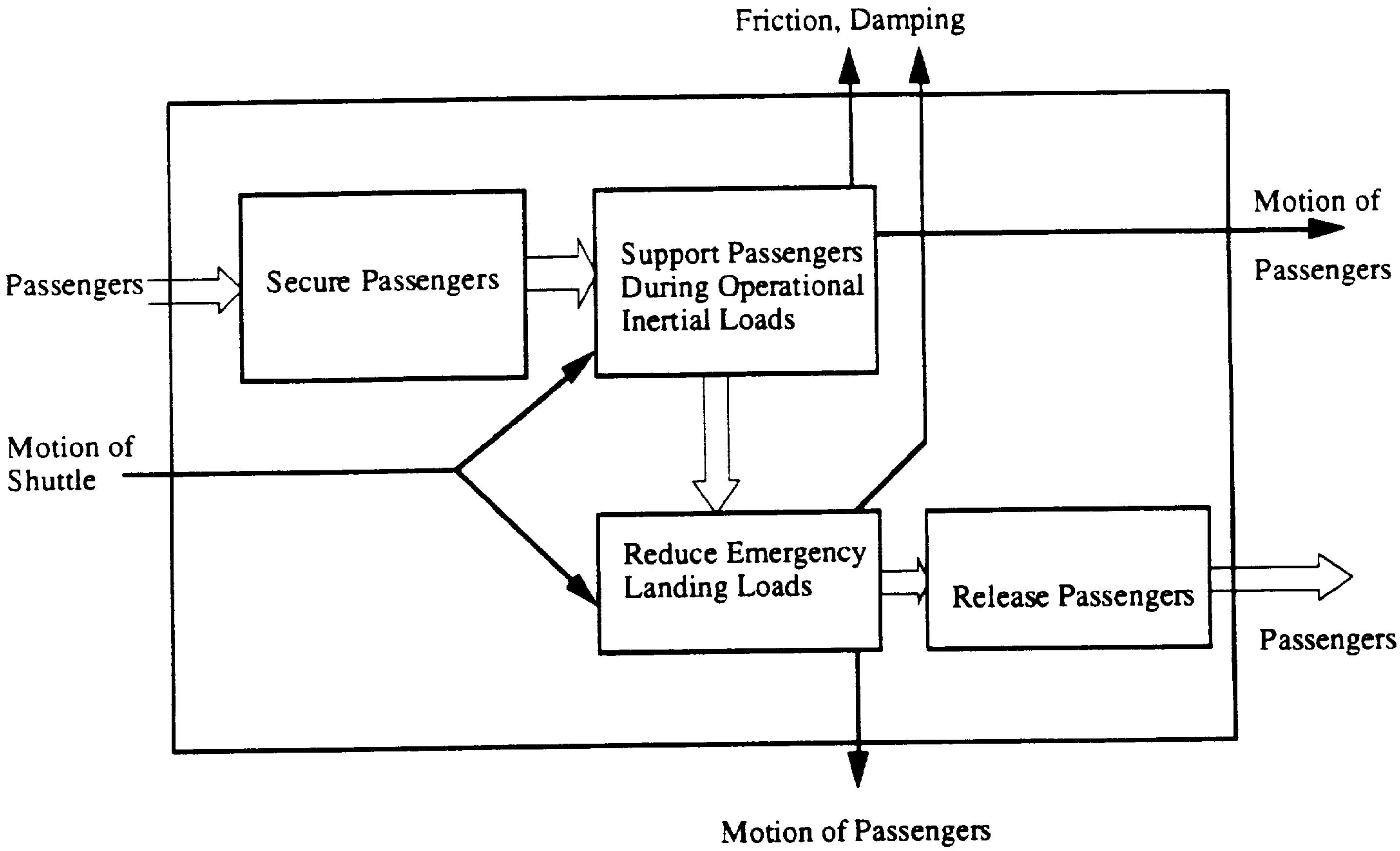


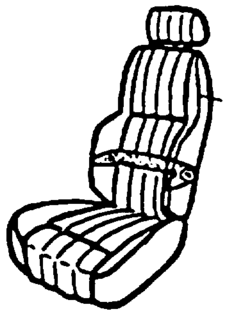
Figure A-3. Acceptable acceleration for an impulse of 1.2 seconds [Sanders, 1987].

Appendix B
Function Structure for the RSS

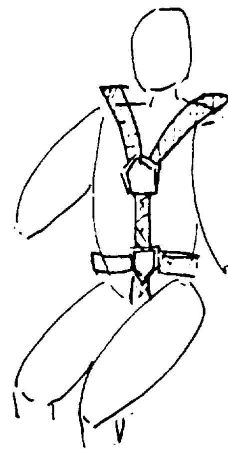


Appendix C
Solution Variants

Secure and Release passenger solution variants:

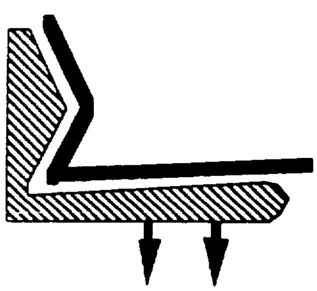


Cushioned seat

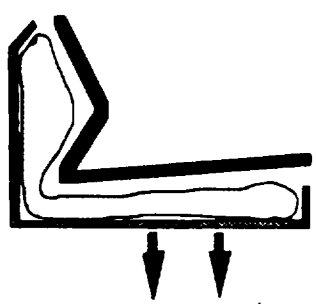


5-point safety belt

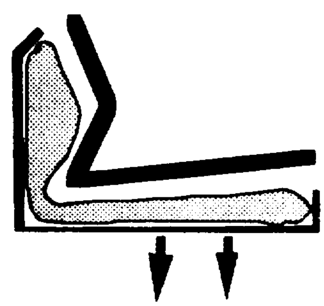
Supporting structure solution variants:



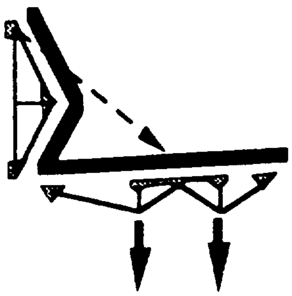
1. Solid medium



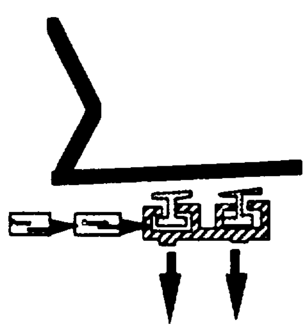
2. Gaseous medium



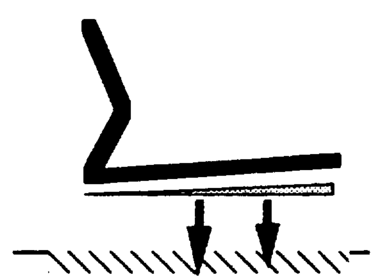
3. Liquid medium



4. Struts

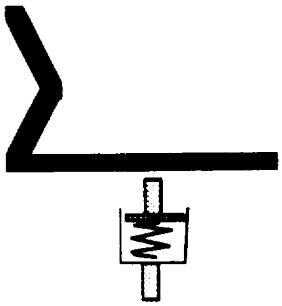


5. Magnetic field

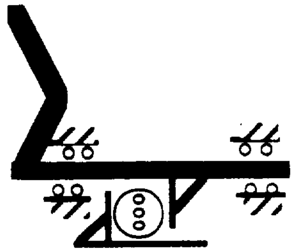


6. Direct floor connection

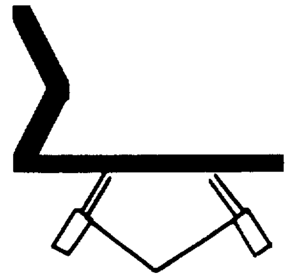
Damping method solution variants:



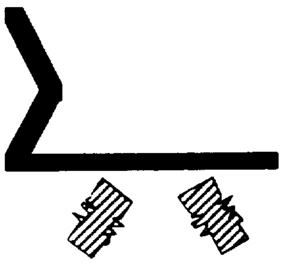
1. Mechanical spring/damper



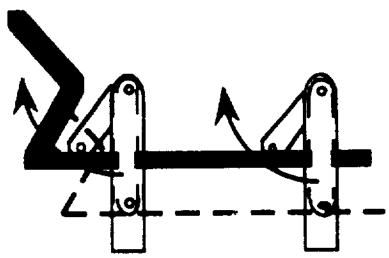
2. Airbag/extrusion



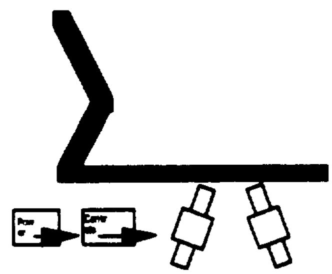
3. Pressfits



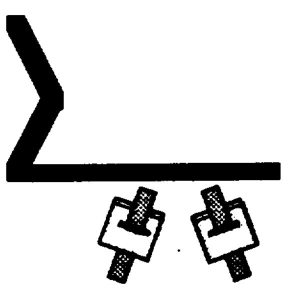
4. Crushable material



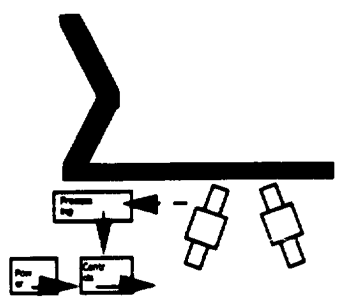
5. Motion of passengers



6. Magnetic damping



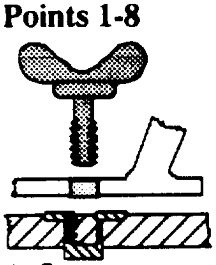
7. Particle damping



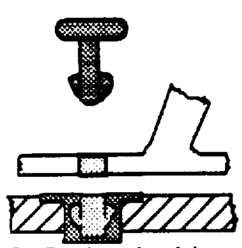
8. Active damping

Means of connection solution variants:

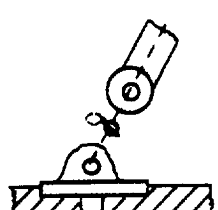
Points 1-8



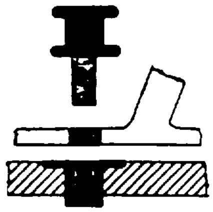
1. Screws



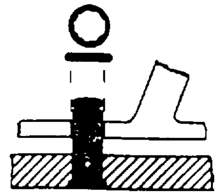
2. Spring locking



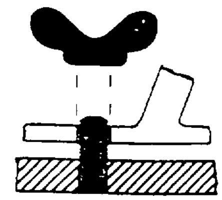
3. Clevis joint



4. cross-pin slot

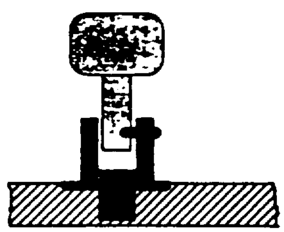


5. self locking retaining rings



6. prefixed bolts with wingnuts

points 9-12



shear pins

Table C-1. Dominance matrix for Latching mechanisms.

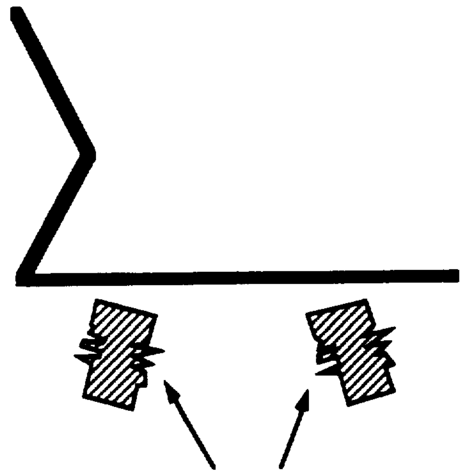
	1	2	3	4	5	6
1	*	1	1	1	0	0
2	0	*	1	1	0	0
3	0	0	*	0	0	0
4	0	0	1	*	0	0
5	1	1	1	1.	*	1
6	1	1	1	1	0	*
Total	2	3	5	4	0	1
Rank	4th	3rd	1st	2nd	6th	5th

Decision criteria for dominance matrix

- Stress concentrations
- Removable fasteners
- Use of tools
- Withstand all forces applied

Appendix D
Feasibility of the Four Remaining Concepts

Crushable Material:



Proof of feasibility:

S_a = Median crushing stress = 16 kips for Al honeycomb

G = Acceration factor = 11g's

W=Weight of object = 1089 lb Where did this come from?

F_a=Average allowable force on mass

v_2, v_1=Velocity

V = Volume

m = mass

A=Area

ε=Allowable Strain

$$F_a = W(G_m + 1) = AS_a$$

$$A = \frac{W}{S_a}(G+1)$$

$$A = \frac{1089}{16000}(11+1) = .8168\ \text{ft}^2 = 117.6\ \text{in}^2 \qquad A \approx 10''\text{x}12''$$

$$V = \frac{\text{Kinetic Energy}}{\text{Material Energy}} = \frac{\frac{1}{2}m(v_2^2 - v_1^2)}{NS_a\varepsilon} = \frac{\frac{1}{2}(1089/32.2)(323^2 - 315^2)}{4(17000)(.7)} = 1.81\ \text{ft}$$

$$h = V / A \approx 2.2\ \text{ft}$$

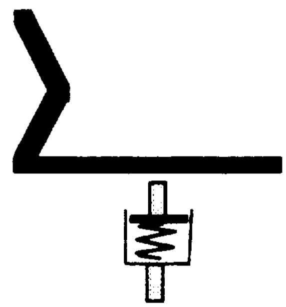
Advantages of crushable material:

- Provides adequate acceleration reduction
- Dissipates energy quickly
- Inexpensive

Disadvantages of crushable material:

- Non-reusable ← That's OK after a crash.
- Bulky
- Possible storage problems
- Only dampens compressive loads

Spring/Damper:



Proof of feasibility:

Given:

ω_n=natural frequency = 30 Hz = 188 r/s = 200r/s
mass = 500 kg

Find spring constant necessary

$\omega_n = (k/m)^{1/2}$

$k = 20 \times 10^6$ Nm

Given:

amount of accelerations to be damped = 10G
mass = 500 kg

Find energy dissipation necessary

(10) (9.81) (500) = 49 kN
motion over a maximum of 0.2 m
E = Force x distance = F*d
E = (49 kN) (0.2 m) = 9.96×10^5 Nm

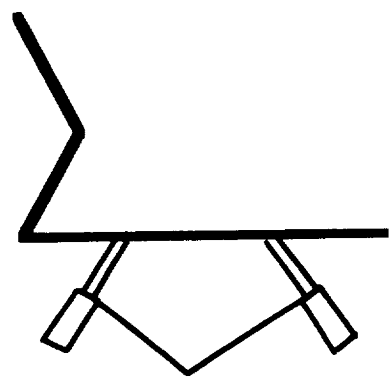
Advantages of spring/damper:

- can use existing springs and shock absorbers
- versatile set up
- reusable
- reliable
- damps during normal flight

Disadvantages of spring/damper:

- heavy
- more moving parts

Pressfits:



Proof of feasibility:

SAE 1020 steel properties:

$E = 30 \times 10^6$ psi

$\sigma_y = 65000$ psi [Eshbach's, 1990]

interference (max)	$\delta = .003$ inches
diameter of shaft (max)	$D = 2$ inches
inner diameter of hub	$d_i = 2 - .003 = 1.997$ inches
set thickness of hub	t .500 inches
outer diameter of hub	$d_o = 1.997 + .500 = 2.497$

Contact pressure

$$P = \frac{E\delta}{2d_i}\left[1 - \left(\frac{d_i}{d_o}\right)^2\right] = 8120.8$$ [*Machine*, 1968]

The: .ᶜore, material is feasible.

height of strut $h = 8$ inches (Assumption)
force to be damped $F_1 = 21780$ lbf
force from astronauts $F_2 = 900$ lbf
coefficient of friction $\mu = .1$

Slippage threshold:

$$S = \pi \frac{fPd_i^2}{2} = 5087 \text{lbf}$$ [Black, 1987]

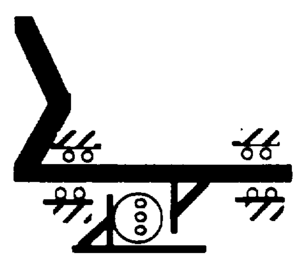
Advantages of pressfits:

- Simple
- Withstands high forces
- Adjustable threshold of reaction

Disadvantages of pressfits:

- No damping during normal flight
- Only dampens compressive forces
- Not reusable

Airbag/Extrusion:



Advantages of airbag/extrusion:

- Low weight
- Speed sensitive

Disadvantages of airbag/extrusion:

- Lower reliability
- Dampens in one direction only (compressive motion)
- Higher set up time

Appendix E
Decision Matrix

Table E-1. Meaning of scores in decision matrix.

Score	Acceleration of Passengers	Reliability (confidence)	Set up Time	Weight	C.O.G location
100%	3G	absolutely	1 minute	100 lb	6 in
90%	5G	extensively	2 minutes	120 lb	8 in
80%	7G	considerably	3 minutes	140 lb	10 in
70%	9G	moderately	4 minutes	160 lb	13 in
40%	11.2G	marginally	5 minutes	180 lb	16 in

Design I. Spring/Damper

Design II. Press Fits

Design III. Crushable Material

Design IV. Airbag/Extrusion

Design	Accel. Felt	Reliability	Set Up Time	Weight	COG	Total
	0.31	0.21	0.16	0.16	0.16	
I	70 21.7	90 18.9	80 12.8	80 12.8	80 12.8	79
II	40 12.4	80 16.8	80 12.8	80 12.8	80 12.8	67.6
III	40 12.4	80 16.8	40 6.4	70 11.2	90 14.4	61.2
IV	40 12.4	70 14.7	40 6.4	90 14.4	40 6.4	54.3

Appendix F
Anthropometric Data and Center of Mass

<u>Body Dimensions</u>.

In determining the required equipment dimensions, the following steps were followed:

1 Collection of anthropometric data for 95th % military male and 5th % oriental female. Similar body dimensions from 5th % civilian or military female were used when this data was not available (see Figure F-1) [Woodson, 1992].

2. Addition of increments to nude-body dimensions by the partial pressure suit worn and micro-gravity effects.

3. Selection of the most relevant body dimensions for the design of the proper seat configuration.

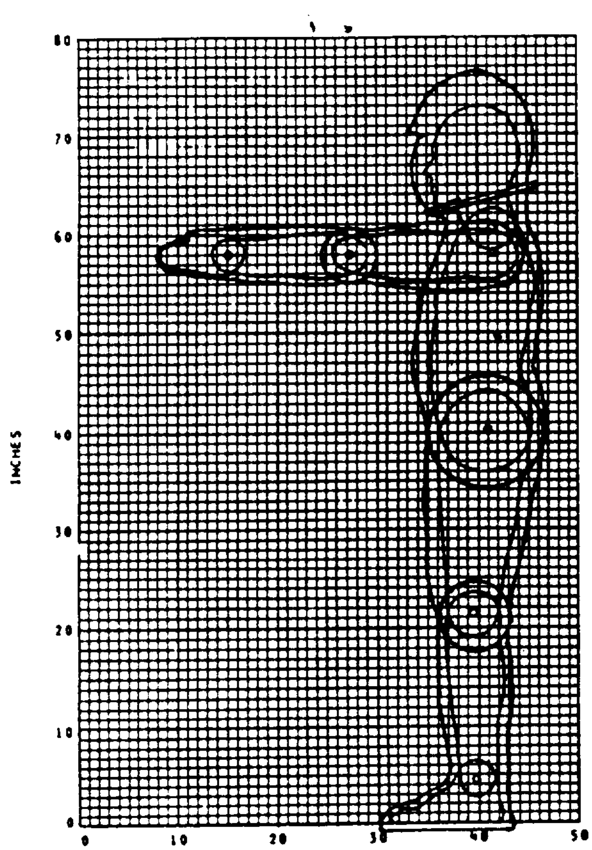


Figure F-1a. Scaled Mannequin -- 95th percentile civilian male

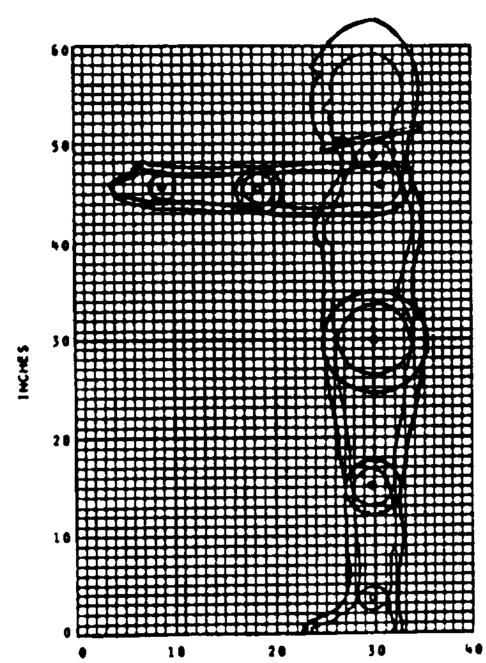


Figure F-1b. Scaled Mannequin -- 5 th percentile civilian female

Table F-1. Critical Dimensions of Fully Suited Astronauts

Important body dimensions	Size (in.)	Increment (in.)	Working dimension (in)
95th % male			
1. Seated height	37.8	+ 6.5 + 1.15	45.45
2. Buttock-to-popliteal length	21.6	+ 0.125	21.73
3. Popliteal height	19.3	- 1.075	18.23
4. Max. strap position	12.43	0	12.43
5. Shoulder-to -shoulder breadth	19.9	+ 6.0	25.9
5th % female			
1. Seated height	30.9	+ 6.5 + 0.927	38.33
2. Buttock-to-popliteal length	17.0	+ 0.125	17.125
3. Popliteal height	13.8	- 1.075	12.375
4. Max. strap position	7.73	0	7.53
5. Shoulder-to-shoulder breadth	14.5	+ 6.0	20.5

In fitting the seat to the maximum and minimum dimensions of the astronauts, an allowable tolerance of ±0.5 inches is selected. Thus, Table F-2 contains the critical dimensions that determine the final layout of the seat configuration.

Table F-2. Critical Dimensions Used in Sizing Seat

Critical dimension	Size (in.)
1. Min. Seated height	46.05±0.050
2. Max. buttock-to-popliteal length	16.63±0.050
3. Min. popliteal height	18.73±0.050
4a. Max. strap position (95th% male)	11.93±0.050
4b. Max.strap position (5th% female)	7.53±0.050
5. Min. Shoulder-to -shoulder breadth	26.4±0.5

<u>Weight of Fully Suited Astronauts</u>

The maximum total weight of the astronauts to be support was determined by adding the weight of three 95th % male astronauts and the weight increments of the partial pressure suit & parachute.

Table F-3. Weight Parameters of the Three Astornauts.

Weight component	Weight (lbs)
1. 95th % male nude-body weight (x3)	203 lbs.x3 = 609 lbs.
2. partial pressure suit and parachute (x3)	100 lbs x3 = 300 lbs.
Total weight	909 lbs

Center of Mass of the 3 Fully Suited Astronauts

The center of mass of the three astronauts is determined by using the anthropometric data for the 95th % male, the weight and dimensions of the pressure suit, and the dimensions of the parachute. The reference point for the location of each component is the top surface of the back support (z = 0) and the top of the astronaut's helmet (x = 0). The weight of the astronaut and the garments worn is shown in Table F-4.

Table F-4. Weight of Single Astronaut and Garments

1. 95th % nude-body weight	203 lbs.
2. Partial-pressure suit	60 lbs. (est.)
3. Parachute	27 lbs. (est.)
4. Helmet	8 lbs. (est.)
5. Boots	5 lbs. (est.)
Total weight of suited astronaut	303 lbs. (est.)

The weight of each body part is determined by distributing the cumulative weight of the suited astronaut by the percentage of the nude-body weight. Then, the weight of each component of the astronauts garments is added it corresponding body part. The weight distribution of the main parts is shown in Table F-5.

Table F-5. Weight Distribution of Main Parts of the Suited Astronauts

main part	% of nude-body weight	weight of part (lbs.)	increment (lbs.)	weight of main part (lbs.)
1. Lower legs	12.37	32.53	+ 5.0 [1]	37.5
2. Upper legs	19.81	52.10	0	52.1
3. Forearms & Hands	4.52	11.89	0	11.9
4. Upper Arms	5.40	14.20	0	14.2
5. Torso & head	57.9	152.28	+ 5.0	157.3
6. Parachute	– – –	– – –	+ 30	30.0

The center of mass of the fully suited astronaut with the parachute is determined by using Equation F-1.

$$m * \bar{z}_{body} = \sum_{i=1}^{6} m_i * \bar{z}_i \qquad \text{(F-1)}$$

The center of mass of the fully suited astronaut from the top of the back support is calculated to be 10.9 in.

Table F-6. Weight Distribution from Back Support

main part	weight (lbs.)	z (in.)	m z (lb-in)
1. Lower legs	37.5	27.0	1012.5
2. Upper legs	52.1	13	677.3
3. Forearms & Hands	11.9	10.5	125.0
4. Upper Arms	14.2	6	85.2
5. Torso & head	157.3	8.5	1337.1
6. Parachute	30.0	1.7	51.0

$\Sigma mz = 3288.1$ lb-in

$$\bar{z}_{body} = \sum (m_i * \bar{z}_i) + m_{total}$$

$$\bar{z}_{body} = 3288.1 + 303$$

$$\bar{z}_{body} = 10.9 \text{ in.}$$

Appendix G
Miscellaneous Seat Calculations

X-Z Plane View

Initial Design (Figure 3a):

Proof of feasibility begins by assuming the angle between the back support and upper leg support is 50°. The length of the back support is approximately 46 inches (from Table F-2) which leaves only 3 inches unused in the operational space (see Figure 1). The back to popliteal length must be 16 inches and the popliteal height is 19 inches. Using these dimensions, the Figure G-1 is created from simple geometry.

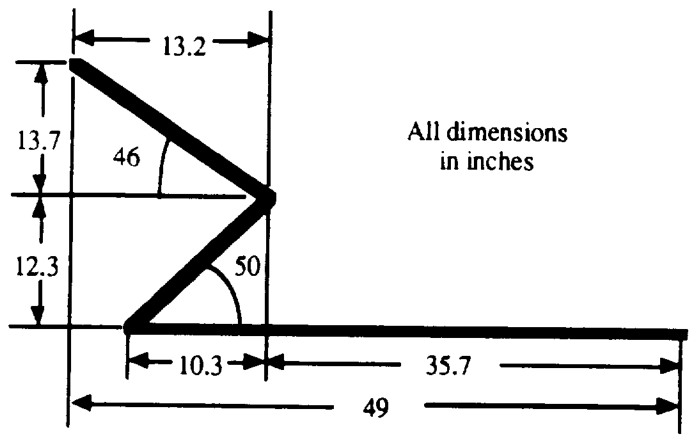


Figure G-1. Feasible Initial Design.

Final Design (Figure 3b):

By placing a flat plate at the end of the back support, full use of the operational space is achieved. Scale models are constructed from the anthropometric data in Appendix F of both a 95th percentile male and a 5th percentile female in pressure suits. These models are used to derive the geometry and dimensions shown in Figure 3b.

Initial Calculation of Seat Material Volume

Section	Number	Length (in)	Total Length(in)
Back Structure:			
Bars that form crosses	6	49	294
Bars at center of gravity	3	22	66
Restraint bars	6	12	72
End bars	2	80	160
Vertical Structure:			
Bars that form crosses	6	21	126
Straight bars on hip section	2	11	22
Straight bars on upper leg	6	13	78
Straight bars on [illegible] leg	6	13.5	81
Straight support [illegible]	4	18	72
End bars	3	80	240
Total			1211

Assuming all the bars to be used are one inch hollow square cross-section with a thickness of one eighth of an inch, the cross-sectional area is 0.4375 square inches. Using the length of 1211 inches, the total volume of material is 530 cubic inches.

Appendix H
Two-Dimensional Model for pal2

Structure file:

```
TITLE TWO DIM STRUT
NODAL POINT LOCATIONS 1
1 40,0,0
2 46,0,6.5
3 7,0,6.5
4 24,0,0

MATERIAL PROPERTIES 10400E3,0,0.1013,0.33,70E3
BEAM TYPE 3,1.612,1.4
CONNECT 1 TO 2
CONNECT 2 TO 3
CONNECT 3 TO 4
ZERO 1
TA 1,4
TY ALL
RX ALL
RZ ALL
RY 3

END DEFINITION
```

Load file:

```
FORCES AND MOMENTS APPLIED 0
FX 3,21000

SOLVE
QUIT
```

The structure file for the two dimensional model is relatively simple. The section entitled NODAL POINT LOCATIONS defines the geometry of the struts. The materials and the beam type are defined next.

The load file for the two dimensinal model contains only one force. This test defines the angle of the strut.

Appendix I
Simulated Forces in the Members
MSC/pal2 Simulation

Three dimensional load file:

```
TITLE THREE DIM
C FIRST SET OF STRUTS
NODAL POINT LOCATIONS 1
1 40,0,0
2 46,0,6.5
3 0,0,6.5
4 24,0,0
C SECOND SET OF STRUTS
NODAL POINT LOCATIONS 1
5 40,12.5,0
6 46,12.5,6.5
7 0,12.5,6.5
8 24,12.5,0
C THIRD SET OF STRUTS
NODAL POINT LOCATIONS 1
9 40,24.5,0
10 46,24.5,6.5
11 0,24.5,6.5
12 24,24.5,0
C FOURTH SET OF STRUTS
NODAL POINT LOCATIONS 1
13 40,37,0
14 46,37,6.5
15 0,37,6.5
16 24,37,0
C NODES FOR X'S
NODAL POINT LOCATIONS 1
17 0,-11.5,6.5
18 0,11.5,6.5
19 0,15.5,6.5
20 0,38.5,6.5
21 46,-11.5,6.5
22 46,11.5,6.5
23 46,15.5,6.5
24 46,38.5,6.5
C ADD STRUTS FOR 3RD PERSON
NODAL POINT LOCATIONS 1
26 46,44.5,6.5
27 0,44.5,6.5
30 46,60.5,6.5
31 0,60.5,6.5
C ADD PTS FOR X'S IN XY PLANE
NODAL POINT LOCATIONS 1
33 0,42.5,6.5
34 0,65.5,6.5
35 0,68.5,6.5
36 46,42.5,6.5
37 46,65.5,6.5
38 46,68.5,6.5
C ADD POINT FOR CANTELEVER
NODAL POINT LOCATIONS 1
39 32,68.5,6.5
40 40,68.5,6.5
```

```
MATERIAL PROPERTIES 10400E3,0,0.1013,0.33,70E3
BEAM TYPE 3,1.612,1.4
CONNECT 1 TO 2
CONNECT 5 TO 6
CONNECT 9 TO 10
CONNECT 13 TO 14
C HORIZONTAL BAR BY FEET
CONNECT 17 TO 3
CONNECT 3 TO 18
CONNECT 18 TO 7
CONNECT 7 TO 19
CONNECT 19 TO 11
CONNECT 11 TO 15
CONNECT 15 TO 20
C  HORIZONTAL BAR BY HELMET
CONNECT 21 TO 2
CONNECT 2 TO 22
CONNECT 22 TO 6
CONNECT 6 TO 23
CONNECT 23 TO 10
CONNECT 10 TO 14
CONNECT 14 TO 24
C X'S FOR SUPPORT
CONNECT 17 TO 22
CONNECT 21 TO 18
CONNECT 19 TO 24
CONNECT 23 TO 20
C EXTEND HORIZONTAL BARS
CONNECT 20 TO 33
CONNECT 33 TO 27
CONNECT 27 TO 31
CONNECT 31 TO 34
CONNECT 34 TO 35
CONNECT 35 TO 39
CONNECT 39 TO 40
CONNECT 40 TO 38
CONNECT 38 TO 37
CONNECT 37 TO 30
CONNECT 30 TO 36
CONNECT 36 TO 26
CONNECT 26 TO 24
C CONNECT 3RD X
CONNECT 33 TO 37
CONNECT 36 TO 34
C CONNECT CANTELEVER
CONNECT 13 TO 40
CONNECT 16 TO 35
C CREATE LARGER BEAMS FOR STRUTS THAT FAIL

MATERIAL PROPERTIES 31000E3,0,0.075,0.21,220E3
BEAM TYPE 3,1.612,1.4
CONNECT 3 TO 4
CONNECT 7 TO 8
CONNECT 11 TO 12
CONNECT 15 TO 16
C CROSS BARS OF BORON EPOXY
CONNECT 4 TO 7
CONNECT 3 TO 8
CONNECT 11 TO 16
CONNECT 12 TO 15

ZERO 1
TA 1,4,5,8,9,12,13,16,25,28,29,32
RY 3,7,11,15,27,31

END DEFINITION
```

Load file for the 20g forward load:

```
FORCES AND MOMENTS APPLIED 0
FX 17,-2150
FX 18,-2150
FX 19,-2150
FX 20,-2150
FX 33,-2150
FX 34,-2150
FX 21,-1433
FX 22,-1433
FX 23,-1433
FX 24,-1433
FX 36,-1433
FX 37,-1433

SOLVE
QUIT
```

Load file for the 12.5g downward load:

```
FORCES AND MOMENTS APPLIED 0
FZ 17,-1700
FZ 18,-1700
FZ 19,-1700
FZ 20,-1700
FZ 33,-1700
FZ 34,-1700
FZ 21,-567
FZ 22,-567
FZ 23,-567
FZ 24,-567
FZ 36,-567
FZ 37,-567

SOLVE
QUIT
```

Load file for the 3.3g side load:

```
FORCES AND MOMENTS APPLIED 0
FY 17,450
FY 18,450
FY 19,450
FY 20,450
FY 33,450
FY 34,450
FY 21,150
FY 22,150
FY 23,150
FY 24,150
FY 36,150
FY 37,150

SOLVE
QUIT
```

Results of pal 2 simulations:

The strut number corresponds to the floor point to which it is attached. For example, strut 1 connects point 1 on the middeck floor to the couch.

Table I-1. Forces in Struts for 20g Foward Load

Strut	Axial Load (Lb)	Percent yield
1	-1133	31%
2	-7685	53%
3	617	20%
4	-1014	54%
5	-2171	53%
6	-8406	84%
7	757	43%
8	-1061	15%

Table I-2. Forces in Struts for 12.5g Downward Load

Strut	Axial Load (Lb)	Percent yield
1	-722	86%
2	-304	65%
3	392	72%
4	-1506	80%
5	1323	69%
6	-1350	70%
7	948	85%
8	-4527	20%

Table I-3. Forces in Struts for 3.3g Side Load

Strut	Axial Load (Lb)	Percent yield
1	1213	2%
2	-1312	2%
3	165	10%
4	3.3	14%
5	1262	2%
6	-1420	3%
7	73	14%
8	-260	10%

Table I-4. Shear in Floor Points 1-8

Floor Point	Foward 20g	Downward 12.5g	Side 3.3g
1	56	256	1825
2	3422	1235	1883
3	596	378	159
4	978	1454	3
5	1100	2895	2051
6	3112	1850	1970
7	730	915	70
8	990	4369	177

Appendix J
Clevis Joint Calculations

The clevis joint is shown in Figure J-1

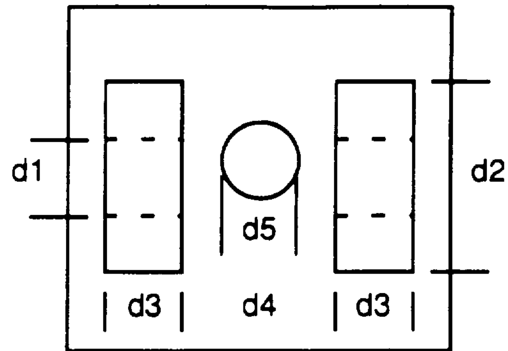


Figure J-1

To be determined:

diameter of pin d1
length of clevis attachment d2
width of clevis attachment d3
width between clevis attachments d4
attaching bolt diameter d5

Known:

strut diameter = 1.6 in
worst case load = 5000 LB

The pin will take all of the load so the design team chose a strong material.
4340 steel

$\sigma = 132{,}000$ psi $\qquad \sigma_y = \sigma/2 = 66{,}000$ psi

Assume loading on the pin is approximated by Figure J-2

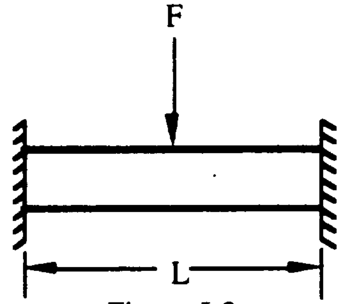


Figure J-2

$V_{max} = F/2 = 2500$ LB $\qquad M_{max} = FL/8 = 333.3$ LB-in

check for bending failure:

$\sigma = 32M/\Pi d1^3$ which gives d1 = 0.295 in

check for shear failure:

$\tau = F/A$ which gives d1 = 0.31 in apply safety factor = 1.25

diameter of pin = 0.3875 in

The clevis attachment should be wide enough to have one diameter on each side of the pin hole.

$$d2=1.1625 \text{ in}$$

Assign d3 and d4 so that they undergo equal stress:

$$2d3 + d4 = 1.6 \text{ in (width of strut)}$$

$\sigma d3 = \sigma d4$ which gives $2d3 = d4$

$d3 = 0.4$ in $d4 = 0.8$ in

Attachment bolt should withstand 5000 Lb shear

$$\sigma_{sy} = 5000/A = 66000$$

$d5 = 0.31$ apply safety factor = 1.25

$$d5 = 0.3875 \text{ in}$$

Appendix K
Final Layout Calculations

Natural Frequency:

Given:

natural frequency $\geq$ 30 Hz = 30 s^{-1}

mass = 1089 lbm

Find:

spring constant necessary:

$w_n = (k/m)^{1/2}$

k=980100 lbm/s^2

Any constant higher than this will have a higher natural frequency than 30 Hz.

spring constant of design:

Back epoxy struts:

k = E A/L

E (epoxy) = 210 GPa = 30 x 10^6 psi [Lee, 1991]

A = $(\pi/4)(1.6^2 - 1.4^2)$= .47 in^2

L = 26.8 in

k = 526119 lbm/s^2

Since there are 4 back bars: k_{tot} = 2104477 lbm/s^2 which is already much higher than the required spring constant and the shock absorbers have not been included yet.

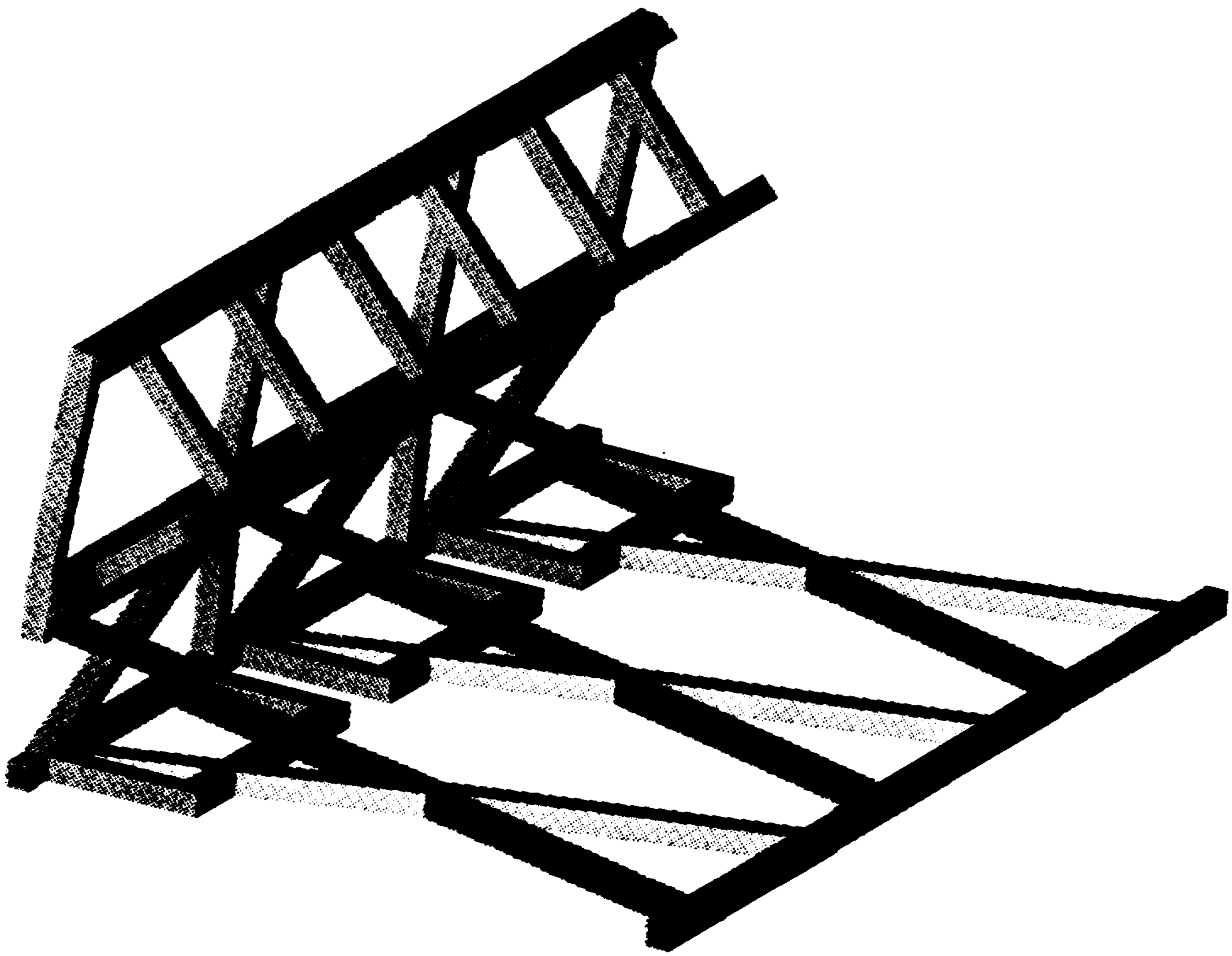
Total Weight:

The total weight of the RSS unit is calculated by adding the weight of all the component parts, as shown below.

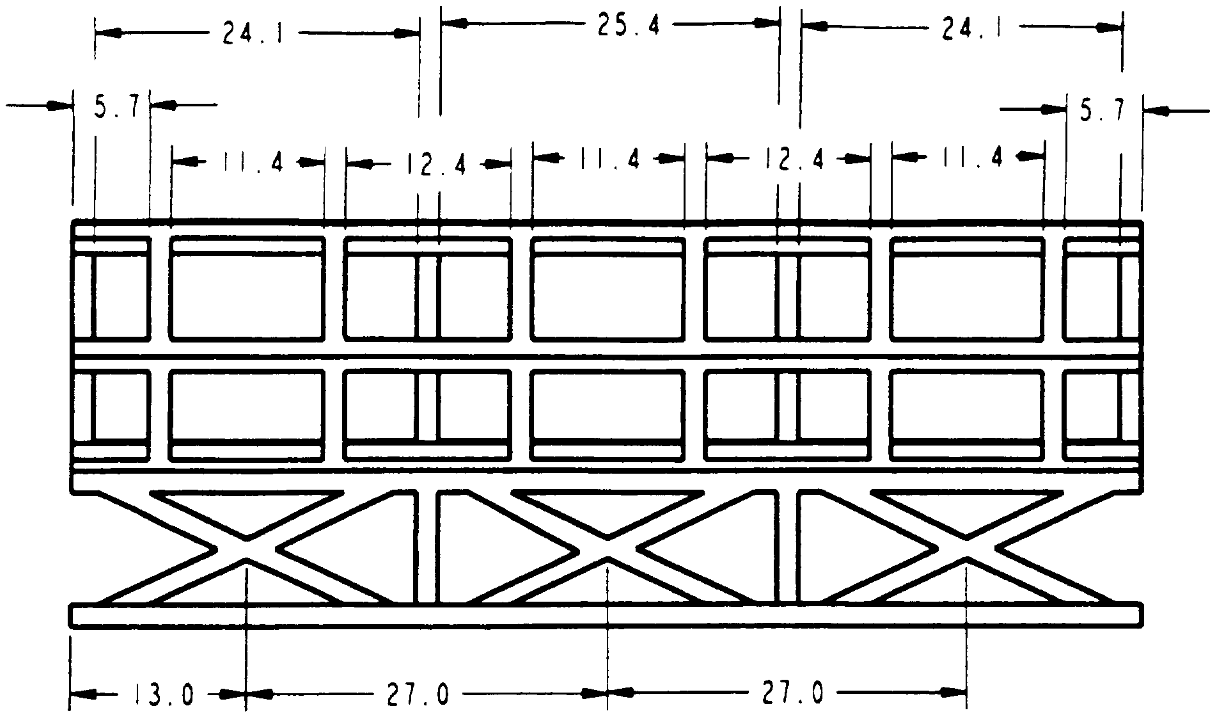
Seat Cushioning- Soft Neo prene Foam Layer	8 lbs	
Seat Cushioning- Balsa Wood Layer	17.5 lbs	
Safety Belts and Buckles	3 lbs	
Seat Panels , (P1, P2, P3, P4)	37 lbs	
Seat Frame , Back Support	26.2 lbs	
Seat Frame , Vertical Leg Support	27.4 lbs	
2 Aluminum Supporting Struts	4.2 lbs	
3 Low Capacity Shock Absorbers	7.5 lbs	(est.)
1 High Capacity Shock Absorber	4.5 lbs	(est.)
8 Boron/epoxy Supporting Struts	8.6 lbs	
10 Aluminum Floor Connecting Joints	4 lbs	
10 Aluminum Frame Connecting Joints	6 lbs	
20 Steel Clevis Joint Pins	2.5 lbs	
20 Steel Locking Pins	0.5 lbs	
Total RSS Weight =	156.9 lbs	

Appendix L

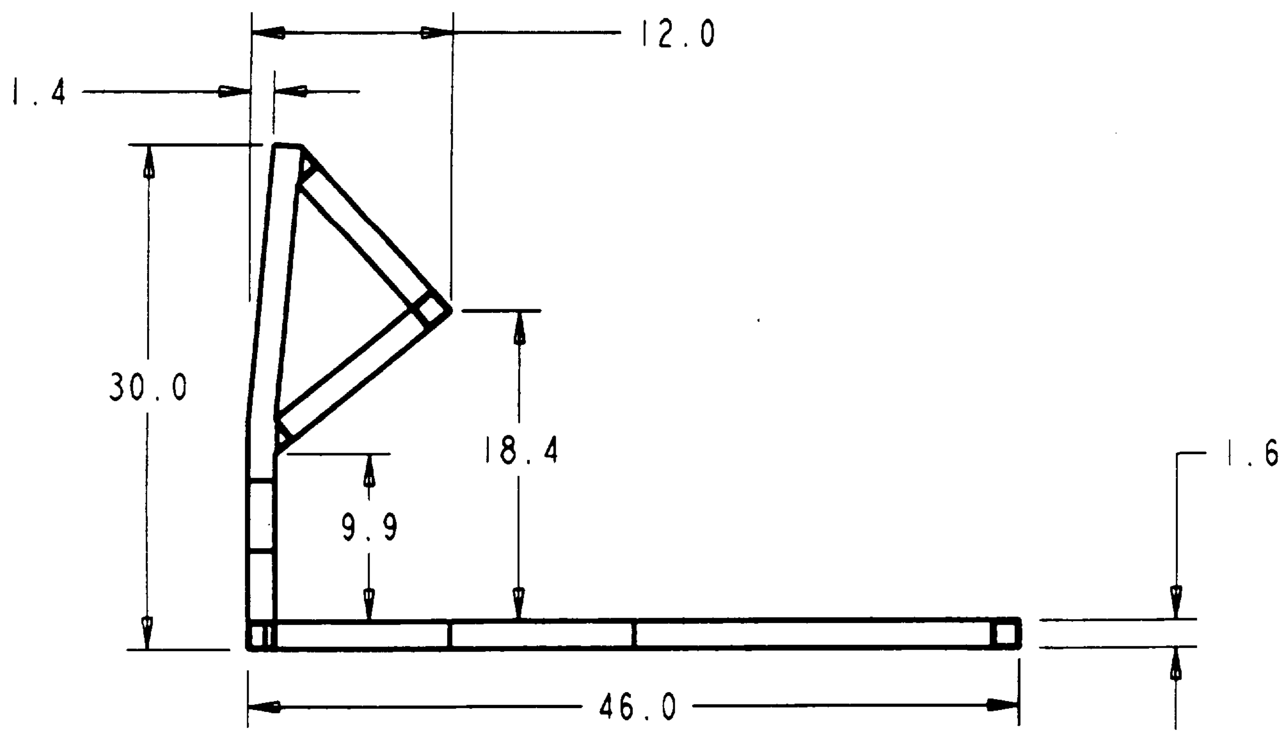
Final Layout Drawing



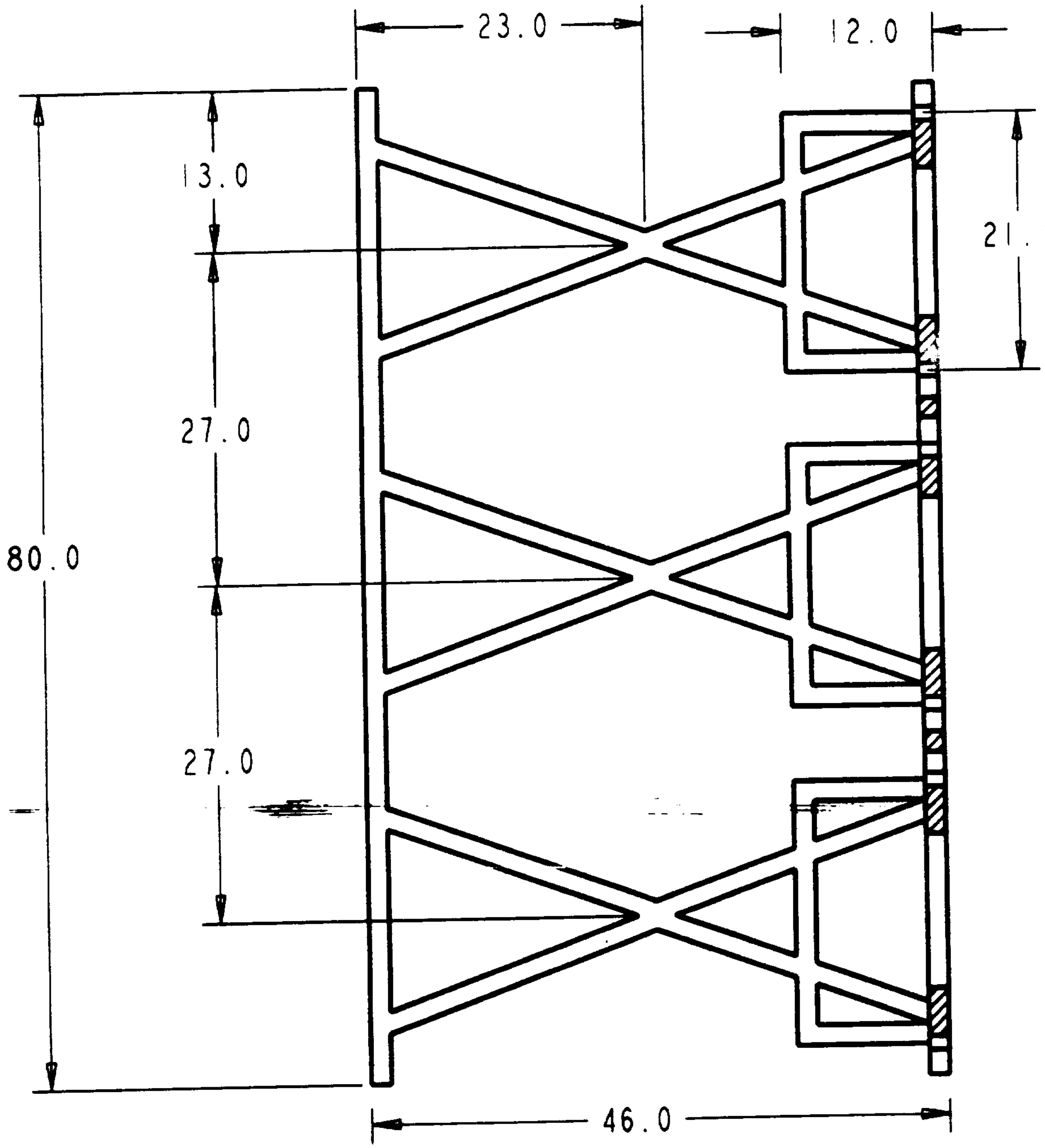
Seat Support



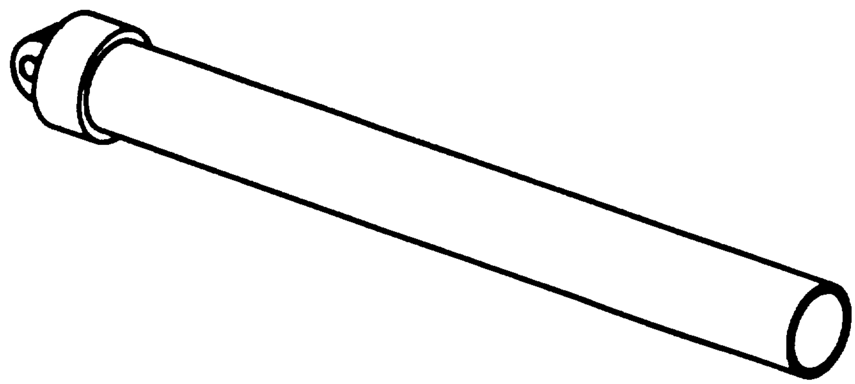
Front View -- Looking into vertical support

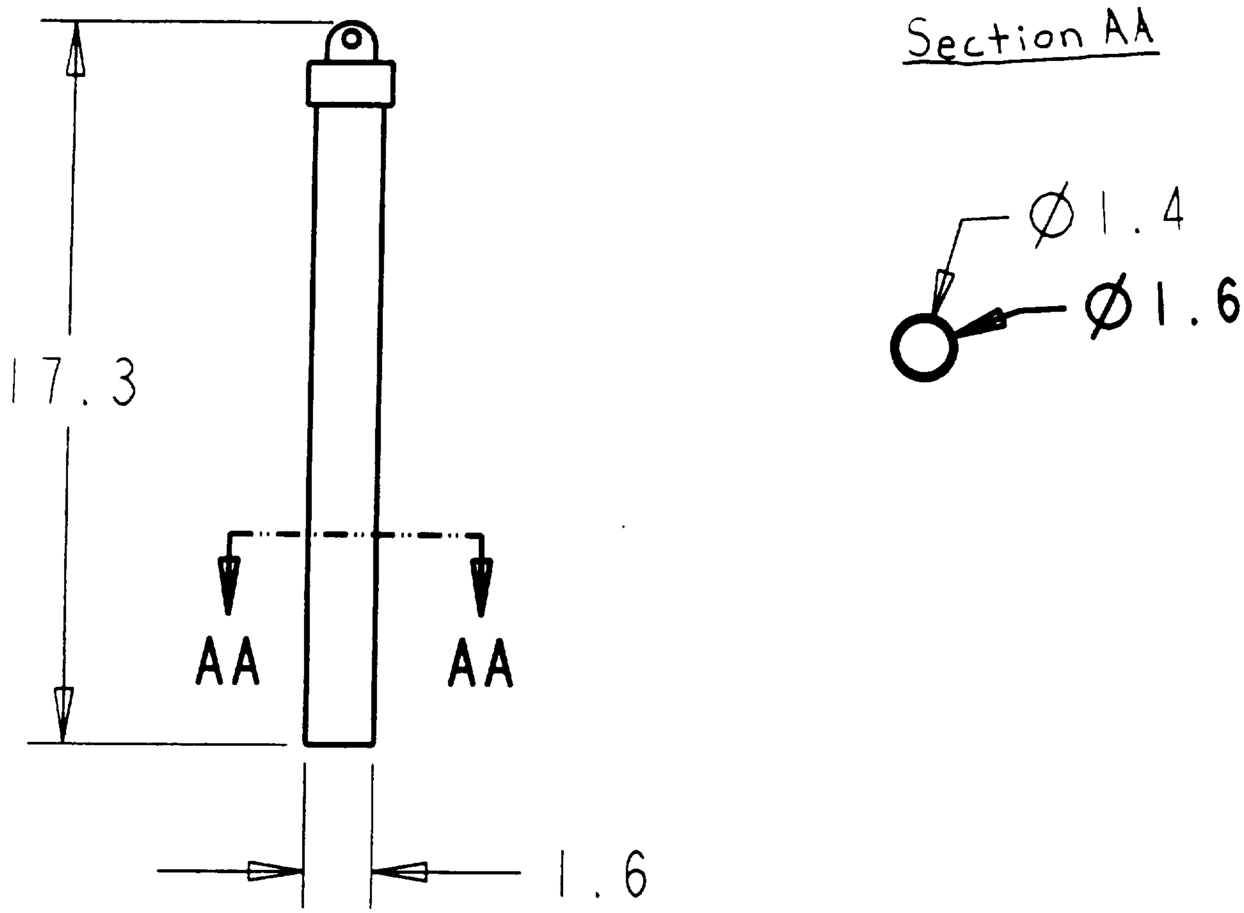


Side View



Top View





Struts 1, 2, 5, and 6 - Boron Epoxy

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