Design of composite sandwich panels for lightweight applications in air cargo containers

Mariana M. William

Follow this and additional works at: https://researchrepository.wvu.edu/etd

Recommended Citation
William, Mariana M., "Design of composite sandwich panels for lightweight applications in air cargo containers" (2016). Graduate Theses, Dissertations, and Problem Reports. 3980.
https://researchrepository.wvu.edu/etd/3980

This Problem/Project Report is protected by copyright and/or related rights. It has been brought to you by the The Research Repository @ WVU with permission from the rights-holder(s). You are free to use this Problem/Project Report in any way that is permitted by the copyright and related rights legislation that applies to your use. For other uses you must obtain permission from the rights-holder(s) directly, unless additional rights are indicated by a Creative Commons license in the record and/or on the work itself. This Problem/Project Report has been accepted for inclusion in WVU Graduate Theses, Dissertations, and Problem Reports collection by an authorized administrator of The Research Repository @ WVU. For more information, please contact researchrepository@mail.wvu.edu.
Design of Composite Sandwich Panels for Lightweight Applications in Air Cargo Containers

Mariana M. William

A Problem Report submitted to
The Benjamin Statler College of Engineering and Mineral Resources
West Virginia University
in partial fulfillment of the requirements
for the degree of

Masters of Sciences
in
Mechanical Engineering

Samir N. Shoukry, Ph.D., Chair
Jacky C. Prucz, Ph.D.
Kenneth H. Means, Ph.D.

Department of Mechanical and Aerospace Engineering
Morgantown, WV

November 2016

Keywords: Design of Composite Sandwich Panels, Air Cargo Containers, Lightweight Technologies, Sandwich Panels, Composite Materials.
ABSTRACT

Design of Composite Sandwich Panels for Lightweight Applications in Air Cargo Containers

Mariana M. William

Air cargo containers are used to load freight on various types of aircrafts to expedite their handling. The current containers are closed containers made of aluminum or combination of aluminum (frame) and Lexan (walls). The objective of this study is to develop innovative, lightweight design and joining concepts for air cargo containers that would allow for weight reduction in the air cargo transportation industry. For this purpose, lightweight carbon fiber woven composite design configuration of a typical air cargo container was developed and manufactured. The new design was devised to meet the FAA-approved certification requirements of the Technical Standard Order TSO-C90, Cargo Pallets, Nets, and Containers. The manufactured model was used to evaluate the technical feasibility and economic viability of creating such a container from fiber-reinforced polymer (FRP) composite materials. The model was also used to assess the need for the development of suitable and innovative joining techniques that could be used in building such containers and estimate the expected weight reduction.

The new design is expected to lower the structural weight of the LD-3 cargo containers from 76 kg for a typical aluminum container to about 20 kg, which represents a weight reduction of 75 percent. This weight reduction would achieve significant savings in fuel cost that would recover the increase in the cost of building such containers.
ACKNOWLEDGEMENTS

It is almost impossible to complete this long journey without the support from my family and friends. First, I would like to thank my parents for their encouragement and support over the years and my husband for his understanding and love during the hardest period in my life.

I would like to express my deepest gratitude and respect for my advisor, Professor Samir Shoukry, for supporting my research and helping me find an area of intellectual pursuit that fit my abilities and interests well. His endless contribution of ideas and insights resulted in bettering my efforts to design the air cargo containers. Having learned under his tutelage helped me to grow both intellectually and as a person.

I would like to thank Dr. Jacky Prucz for all his help in my study. Many thanks are owed to Dr. Ken Means for serving on my examining committee. Special thanks also extended to Ms. Susan Dess, Manager of Engineering Support Center at American Airlines, for her assistance and advice during the composite manufacturing and prototyping of the air cargo container.
# TABLE OF CONTENTS

<table>
<thead>
<tr>
<th>Section</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>ABSTRACT</td>
<td>II</td>
</tr>
<tr>
<td>ACKNOWLEDGEMENTS</td>
<td>III</td>
</tr>
<tr>
<td>TABLE OF CONTENTS</td>
<td>IV</td>
</tr>
<tr>
<td>LIST OF TABLES</td>
<td>V</td>
</tr>
<tr>
<td>LIST OF FIGURES</td>
<td>VI</td>
</tr>
<tr>
<td>CHAPTER 1 INTRODUCTION</td>
<td>1</td>
</tr>
<tr>
<td>1.1 BACKGROUND</td>
<td>1</td>
</tr>
<tr>
<td>1.2 UNIT LOAD DEVICES</td>
<td>2</td>
</tr>
<tr>
<td>1.3 COMPOSITE UNIT LOAD DEVICES</td>
<td>5</td>
</tr>
<tr>
<td>1.4 RESEARCH OBJECTIVES</td>
<td>5</td>
</tr>
<tr>
<td>CHAPTER 2 LITERATURE REVIEW</td>
<td>7</td>
</tr>
<tr>
<td>2.1 INTRODUCTION</td>
<td>7</td>
</tr>
<tr>
<td>2.2 SANDWICH COMPOSITES</td>
<td>7</td>
</tr>
<tr>
<td>2.2.1 Core</td>
<td>8</td>
</tr>
<tr>
<td>2.2.1.1 Honeycomb Core</td>
<td>11</td>
</tr>
<tr>
<td>2.3 FACE MATERIALS</td>
<td>12</td>
</tr>
<tr>
<td>CHAPTER 3 UNIT LOAD DEVICE DESIGN</td>
<td>16</td>
</tr>
<tr>
<td>3.1 INTRODUCTION</td>
<td>16</td>
</tr>
<tr>
<td>3.2 DESIGN REQUIREMENTS</td>
<td>16</td>
</tr>
<tr>
<td>3.2.1 Base Performance</td>
<td>17</td>
</tr>
<tr>
<td>3.3 THEORETICAL MODEL</td>
<td>18</td>
</tr>
<tr>
<td>3.4 FAILURE MODES FOR HONEYCOMB SANDWICH STRUCTURES</td>
<td>25</td>
</tr>
<tr>
<td>3.4.1 Failure Modes in the Skin</td>
<td>25</td>
</tr>
<tr>
<td>3.4.2 Core Failure</td>
<td>27</td>
</tr>
<tr>
<td>3.5 LIGHTWEIGHT DESIGN OF LD-3 BASE</td>
<td>28</td>
</tr>
<tr>
<td>CHAPTER 4 PROTOTYPING OF LIGHTWEIGHT ULD</td>
<td>34</td>
</tr>
<tr>
<td>4.1 INTRODUCTION</td>
<td>34</td>
</tr>
<tr>
<td>4.1.1 Hand Lay-up</td>
<td>34</td>
</tr>
<tr>
<td>4.1.2 Bag Molding</td>
<td>35</td>
</tr>
<tr>
<td>4.2 PROTOTYPING</td>
<td>36</td>
</tr>
<tr>
<td>CHAPTER 5 SUMMARY AND CONCLUSIONS</td>
<td>40</td>
</tr>
<tr>
<td>REFERENCES</td>
<td>41</td>
</tr>
<tr>
<td>TABLE 1.1</td>
<td>Weights and Volumes of Current LD-3 Containers</td>
</tr>
<tr>
<td>TABLE 2.1</td>
<td>Honeycomb Sandwich Efficiency</td>
</tr>
<tr>
<td>TABLE 2.2</td>
<td>Comparison between Honeycomb and Foam Cores (Bitzer 1997)</td>
</tr>
<tr>
<td>TABLE 3.1</td>
<td>Ultimate Load Criteria for LD-3</td>
</tr>
<tr>
<td>TABLE 3.2</td>
<td>Numerical Factors $\alpha$, $\beta$, $\gamma$, $\delta$, $n$ for Uniformly Loaded and Simply Supported Rectangular Plates ($\nu = 0.3$)</td>
</tr>
<tr>
<td>TABLE 3.3</td>
<td>Properties of Nomex Honeycomb Core</td>
</tr>
</tbody>
</table>
LIST OF FIGURES

Figure 1.1  ULD Cargo Containers in Airbus A300 (Wikipedia, 2014) ..................3
Figure 1.2  Unit Load Device LD-3..................................................................3
Figure 2.1  Types of Sandwich Panel Cores (Petras 1998).................................9
Figure 2.2  Cross Section of Monococque and Sandwich Construction. ............10
Figure 2.3  Specific Stiffness and Specific Strength for Various Materials .........14
Figure 3.1  Internal Forces on the Plate............................................................18
Figure 3.2  Coordinate System of the plate. ....................................................21
Figure 3.3  Face Yielding ..............................................................................25
Figure 3.4  Intra-Cell Dimpling ....................................................................26
Figure 3.5  Face Wrinkling ..........................................................................26
Figure 3.6  Core Shear Failure .......................................................................27
Figure 3.7  Cure Crushing Failure ...................................................................28
Figure 3.8  Nomex Honeycomb Core HD-1/8-1.8 .........................................29
Figure 4.1  Nomex Honeycomb Core Used in Prototyping ...............................36
Figure 4.2  Finished Composite Sandwich Panel for ULD Base......................37
Figure 4.3  Side Walls of the Air Cargo Container ..........................................38
Figure 4.4  Full Scaled Model of Air Cargo Container .....................................38
Figure 4.5  Clip Joint for Attaching Side Walls...............................................39
Chapter 1

Introduction

1.1 Background

The role of air transport in providing rapid and intercontinental connections has made it an essential economic and social conduit throughout the world. In 2010, the air transport industry transported approximately 43.3 million tons of freight worldwide, up from 30.4 million in 2000, which account for nearly 40 percent of all goods by value. Many developing countries today depend heavily on air cargo for their exports as other modes of transportation are unreliable or non-existent (The World Bank, 2012). This demonstrates clearly that the air transport sector is undergoing an optimistic growth rate while at the same time eliciting growing concern, due to its environmental impact and its vulnerability with respect to energy security. These issues have put the sector at the forefront of the tide in achieving energy efficiency. Efforts have been made on every front to improve efficiency through better technology, optimized operation, as well as energy-saving infrastructure.

According to the International Energy Agency (IEA), aviation used 246 million tons of oil equivalents (Mtoe) of energy in 2006, which represented 11 percent of all transport energy used. Aviation’s energy usage is expected to triple to about 750 Mtoe by 2050, according to the IEA’s baseline scenario; as a result, aviation would account for 19 percent of all energy used (IEA 2009). The growing demand in energy along with rising fuel costs is endangering the air transport’s optimistic growth. Traditionally, fuel costs were less than 15 percent of airline operational costs; however, they have risen substantially since 2003. Fuel costs rose to around 33 percent in 2008 and exceeded 40 percent for carriers with lower labor costs (IATA 2009). However, some studies suggest that aviation overall warming impact is much higher given its emissions of the greenhouse gases such as NO\textsubscript{x}, CH\textsubscript{4}, and H\textsubscript{2}O among others, as well as differential effects of emissions at different altitudes.

With a growing sense of urgency for sustainability actions among consumers and governments around the world, air transport industry is under pressure to operate in
sustainable manners (Brown 2009). Freight transportation is a large and fast growing contributor of GHG emissions, especially carbon dioxide (CO₂) that accounts for more than 90 percent of GHGs (Varma and Clayton 2010). Aviation causes about 2 percent of total man-made carbon emissions according to the Intergovernmental Panel on Climate Change (IPCC). Aviation emitted about 810 million tons of CO₂ in 2006, which represents about 12 percent of all transport CO₂ emissions (IATA 2009). The industry is growing by around 5% a year in the longer term but efficiencies already in place mean aviation CO₂ emissions are growing by just 2 to 3 percent. Therefore, it is vital to develop new technologies in to reduce the overall emissions. This could be best achieved by lowering fuel consumption through enhancing aviation efficiencies.

Lightweighting aircraft and freight transport hardware by using new materials and composites was envisioned as a means that can significantly improve fuel efficiency and reduce greenhouse gas emission. This significant weight reduction will also result in an improved payload, and reduce the freight cost.

1.2 Unit Load Devices

A unit load device (ULD) is a pallet or container used to load luggage, freight, and mail on an aircraft. It allows a large quantity of cargo to be bundled into a single unit, thus leads to fewer units to load. As a result, it saves ground crews time and effort and helps prevent delayed flights. Because of regulatory requirements as well as practical considerations, the shape, size and maximum weight of a ULD for each type of aircraft have been standardized.

Typically, ULDS are shaped as boxes, which can include sloped surfaces, which conform the ULD to the aircraft’s fuselage when the ULD is placed in the cargo compartment as shown in Figure 1.1. The container is made of several panels that are joined together to form the ULD and define an enclose storage volume. The ULD is often constructed from a metal such as aluminum or one of its alloys that are able to tolerate the tough handling conditions the container experiences through transfer and transport situations.
ULDs are built in several shapes and types to be compatible with different aircrafts. This study will focus on LD-3, shown in Figure 1.2, as a case study keeping in mind that any new lightweight design that will be developed could be extended and applied to any other ULD types.

![Image of ULD Cargo Containers in Airbus A300](Wikipedia, 2014)

**Figure 1.1** ULD Cargo Containers in Airbus A300 (Wikipedia, 2014)

**Figure 1.2** Unit Load Device LD-3
<table>
<thead>
<tr>
<th>Company</th>
<th>Internal Volume</th>
<th>Maximum Gross Weight</th>
<th>Tare Weight (Approx.) with Net</th>
<th>Country</th>
</tr>
</thead>
<tbody>
<tr>
<td>FedEx (AVE/LD-3)</td>
<td>153 ft³, 4.3 m³</td>
<td>3,500 lb, 1,588 kg</td>
<td>215 lb, 98 kg</td>
<td></td>
</tr>
<tr>
<td>Profreight (LD-3)</td>
<td>150 ft³, 4 m³</td>
<td>3,500 lb, 1,588 kg</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Nippon Cargo Airlines</td>
<td>153 ft³, 4.3 m³</td>
<td>3,500 lb, 1,588 kg</td>
<td>198-231 lb, 90-105 kg</td>
<td></td>
</tr>
<tr>
<td>Grange Aerospace AKE</td>
<td>150 ft³, 4 m³</td>
<td>3,500 lb, 1,588 kg</td>
<td>187 lb, 85 kg</td>
<td></td>
</tr>
<tr>
<td>Shapiro AKE</td>
<td>152 ft³, 4.3 m³</td>
<td>3,500 lb, 1,588 kg</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Cathay Pacific Cargo AKE</td>
<td>152 ft³, 4.3 m³</td>
<td>3,500 lb, 1,588 kg</td>
<td>73-100 kg</td>
<td></td>
</tr>
<tr>
<td>Boeing</td>
<td>159 ft³, 4.5 m³</td>
<td>3,500 lb, 1,588 kg</td>
<td>181 lb, 82 kg</td>
<td></td>
</tr>
<tr>
<td>DSV Global Transport and Logistics</td>
<td>3,500 lb, 1,587 kg</td>
<td>181 lb, 82 kg</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Emirates Sky Cargo</td>
<td>152 ft³, 4.3 m³</td>
<td>3,500 lb, 1,587 kg</td>
<td>66 kg, Emirates</td>
<td></td>
</tr>
<tr>
<td>Air New Zealand</td>
<td>153 ft³, 4.3 m³</td>
<td>3,500 lb, 1,587 kg</td>
<td>187 lb, 85 kg, New Zealand</td>
<td></td>
</tr>
<tr>
<td>Royal Jordanian Cargo</td>
<td>3.8 ft³</td>
<td>3,500 lb, 1,588 kg</td>
<td>85 kg</td>
<td></td>
</tr>
<tr>
<td>Quantum Transportation LTD.</td>
<td>150 ft³, 4.2 m³</td>
<td>3,493 lb, 1,588 kg</td>
<td>158 lb, 72 kg</td>
<td></td>
</tr>
<tr>
<td>Atlas Logistics</td>
<td>152 ft³, 4.3 m³</td>
<td>3,500 lb, 1,588 kg</td>
<td>100 kg, India</td>
<td></td>
</tr>
<tr>
<td>Sea Rates</td>
<td>159 ft³, 4.5 m³</td>
<td>3,500 lb, 1,588 kg</td>
<td>181 lb, 82 kg, England</td>
<td></td>
</tr>
<tr>
<td>Air China Cargo</td>
<td>152 ft³, 4.3 m³</td>
<td>3,500 lb, 1,588 kg</td>
<td>73-100 kg, China</td>
<td></td>
</tr>
<tr>
<td>ANA Cargo</td>
<td>156 ft³, 4.4 m³</td>
<td>3,500 lb, 1,587 kg</td>
<td>157 lb, 71 kg, China</td>
<td></td>
</tr>
<tr>
<td>VRR Aviation</td>
<td></td>
<td>3,500 lb, 1,588 kg</td>
<td>&gt;152 lb, &gt;69 kg</td>
<td></td>
</tr>
<tr>
<td>Dragonair Cargo</td>
<td>152 ft³, 4.3 m³</td>
<td>3,500 lb, 1,588 kg</td>
<td>100 kg</td>
<td></td>
</tr>
<tr>
<td>Turkish International Forwarding &amp; Logistic Services</td>
<td>150 ft³, 4.2 m³</td>
<td>3,493 lb, 1,588 kg</td>
<td>158 lb, 72 kg, Turkish</td>
<td></td>
</tr>
<tr>
<td>TKM Global</td>
<td>150 ft³, 4.2 m³</td>
<td>3,493 lb, 1,588 kg</td>
<td>158 lb, 72 kg, Germany</td>
<td></td>
</tr>
<tr>
<td>Tetra Logistics</td>
<td>152 ft³, 4.3 m³</td>
<td>3,500 lb, 1,588 kg</td>
<td>100 kg</td>
<td></td>
</tr>
<tr>
<td>Air Fast Freight System</td>
<td></td>
<td>3,500 lb, 1,587 kg</td>
<td>90-105 kg, China</td>
<td></td>
</tr>
<tr>
<td>Chep</td>
<td></td>
<td>3,500 lb, 1,588 kg</td>
<td>82 kg</td>
<td></td>
</tr>
<tr>
<td>Egyptair</td>
<td>160 ft³, 4.53 m³</td>
<td>3,500 lb, 1,588 kg</td>
<td>154 lb, 70 kg, Egypt</td>
<td></td>
</tr>
</tbody>
</table>
The LD-3 provides a volumetric capacity of 4.50 m³ (159 cubic foot). Currently, there are multiple manufacturers for this LD-3. Although the outside dimensions for the units produced by different manufacturers are the same, the inside volume and tare weight of the unit differ among manufactures as shown in Table 1.1.

The performance requirements and test parameters for airworthiness approval of a unit load device determine the ultimate load capabilities under defined restraint conditions. The ULD ultimate load is defined as the maximum expected limit load multiplied by a factor of safety of 1.5. For a typical LD-3 container shown in Figure 1.2, the maximum gross weight is 15,567 N (3,500 lbs) and its ultimate load is 23,350 N (5,250 lbs). The current tare weight of this aluminum container is 805 N (181 lbs).

1.3 Composite Unit Load Devices

The new trend in the industry is to replace traditional all-aluminum or semi-composite ULDs with new lightweight, all-composite ULDs. Caro Composites produced new line of products named “AeroBox” whose upper structure is made of all-composite materials. The tare weight of this container is 58 kg (128 lb) that includes a 2.5 mm thick 7075-T6 aluminum base (Cargo Composites 2016).

The composite panels that form the body of the container are comprised of two tough, fiberglass/polypropylene composite skins thermo-fused to a resilient polypropylene honeycomb core. This combination of materials absorbs high impacts and evenly deflects forces across the panels, reducing damage and significantly increasing uptime. The edges of the panels are joined to the sides of the container using industry standard lock bolts. This eliminates the traditional damage-prone post and beam framework/superstructure that is used in both aluminum and semi-composite ULDs.

1.4 Research Objectives

The main objective of this study is to create design and joining methods that will allow the incorporation of lightweight composite plates, panels or structures into a ULD container design. Innovative joining methods will create solutions to
connect components in ULD container, in place of mechanical joints, which include bolting and riveting.
Chapter 2
Literature Review

2.1 Introduction

This chapter presents a review of the current and emerging technologies that could be adopted and integrated in order to reduce the structural weight of air cargo containers, hence improve fuel efficiency and reduce environmental impacts. Advanced composites have become an attractive design alternative for a wide range of industrial applications due to their excellent mechanical properties such as high strength-to-weight ratio, impact and fatigue properties. Such unique properties of composites promise a variety of applications ranging from lightweight construction, impact energy absorption and thermal insulation.

2.2 Sandwich Composites

The principle of the sandwich panel has been put to effective use for many years before it was defined by engineers and recognized as a separate type of construction dominated by certain mathematical principles (Vinson 1999). The sandwich concept was first recognized during the accelerated search for high-strength, lightweight materials for aircraft in World War II (Seidl 1956). Sandwich panels are comprised of two face sheets or top and bottom layers with a core material placed or “sandwiched” between. This type of arrangement creates a light and stiff structure, because the stiff faces are distanced from the neutral axis, similar to the flanges of an I-beam. The facings are made of high-strength material, such as steel, and composites such as graphite/epoxy while the core is made of thick and lightweight materials such as foam, cardboard or plywood (Kaw 2006). The faces carry the majority of the axial loading and transverse bending stress (Hexell Composites 2000). The core resists the shear loads, increases the stiffness of the structure by holding the facing skins apart, and improving on the I-beam, it gives continuous support to the flanges or facing skins to produce a uniformly stiffened panel. Thus, it serves to stabilize the faces against buckling and carries most of the shear forces (Nicholls 1976). The core-to-skin
adhesive rigidly joins the sandwich components and allows them to act as one unit with a high torsional and bending rigidity. When specific tailoring of a sandwich composite is required, the top and bottom face sheets may differ in material and thickness. A change of this nature would aide a sandwich composite that needs temperature resistance on one side more than the opposing side or perhaps one side will primarily carry an impact load or static deflection.

2.2.1 Core

The purpose of the core is to increase the flexural stiffness of the panel. The core should have a low density in order to add as little as possible to the total weight of the panel. However, it must have enough stiffness in shear and perpendicular to provide spacing between the face sheets. Additionally, the core must withstand compressive loads without failure (Mukundan 2003). In sandwich composite design, there are no limitations as to what material can be used as a core structure. Development of new core materials is a primary interest in sandwich composite design and has evolved tremendously over the years. Materials used for cores include polymers, aluminum, wood, papers, and composites. To minimize weight, these materials could be used in various structural forms, which could be classified into four main categories: (a) foam, (b) honeycomb, (c) corrugated, and (d) web as seen in Figure 2.1.

The sandwich structures shown in Figure 2.1 have variations and different attributes for each type of core material. Foam or solid cores, shown in Figure 2.1 (a), are relatively inexpensive and can consist of balsa wood, and infinite selection of foam or plastic materials with a wide range of densities and shear moduli. Honeycomb-core architecture, Figure 2.1 (b), have been widely used since 1940s. The two most common types are the hexagonally-shaped cell structure, also known as Hexcel, and the square cell, also known as egg-crate (Hex. Web core, Figure 2.1 (d) and corrugated core shown in Figure 2.1 (c) are analogous to set of I-beams or Z-sections with their flanges connected together. In both design, the space in the core could be utilized for liquid storage or as a heat exchanger.

In all cases, the primary loading, both in-plane and bending, are carried by the faces, while the core resists transverse shear loads. It is acceptable to assume that in foam and honeycomb core sandwich composites all the in-plane and bending loads
are carried by the faces only. In web-core and corrugated-core structures the core carries some of the in-plane and bending loads (Vinson 1999).

The increase in flexural stiffness from a monocoque construction to a sandwich composite can be shown mathematically. Figure 2.2 (a) shows a sandwich construction that employs two identical isotropic face plates of thickness $t_f$ and a core thickness of $h_c$. Figure 2.2 (b) shows a sheet monocoque construction of thickness 2 $t_f$. 

Figure 2.1 Types of Sandwich Panel Cores (Petras 1998).
The flexural stiffness per unit width, $D$, for a solid laminate panel is

$$D_{mon} = \frac{2E_{f}t_{f}^{3}}{3(1-v_{f}^{2})} \quad \text{............... (2.1)}$$

and the flexural stiffness of a sandwich panel with a foam or honeycomb core is

$$D_{sand} = \frac{E_{f}t_{f}h_{c}^{2}}{2(1-v_{f}^{2})} \quad \text{............... (2.2)}$$

The ratio of the flexural stiffness of the sandwich panel to that of the solid laminate plate is

$$\frac{D_{sand}}{D_{mon}} = \frac{3}{4} \left(\frac{h_{c}}{t_{f}}\right)^{2} \quad \text{............... (2.3)}$$

Analyzing the ratio shows that if the ratio of the face sheet thickness, $t_{f}$ to the core thickness, $h_{c}$ is 1/20 then the flexural stiffness of the sandwich panel is 300 times greater than that of the solid laminate plate. By comparison, the sandwich construction with the same material and total face sheet thickness identical to the laminate thickness, results in lower lateral deflections, higher overall buckling loads, and higher natural frequencies.

In the same way, for a bending moment $M$, the monocoque construction results in maximum stresses at the top and bottom surface of

$$\sigma_{mon} = \pm \frac{6M}{(2t_{f}^{2})} = \frac{3M}{2t_{f}^{2}} \quad \text{............... (2.4)}$$

Similarly, for the bending moment $M$, the maximum stresses in a sandwich face are:
\[
\sigma_{\text{sand}} = \pm \frac{M}{(h_c + t_f)t_f} \quad \text{.................. (2.5)}
\]

Therefore, the ratio of the bending stress in a sandwich face to the maximum stress in a monocoque structure of approximately the weight is:

\[
\frac{\sigma_{\text{sand}}}{\sigma_{\text{mon}}} = \frac{2t_f}{3(h_c + t_f)} \quad \text{.................. (2.6)}
\]

For the example of a sandwich in which \(t_f/h_c = 1/20\), the bending stress in the sandwich structure is 2/63 that of monocoque construction. This means the sandwich structure has a flexural capacity as 31.5 times as that of a monocoque construction of approximately same weight.

### 2.2.1.1 Honeycomb Core

The main reason for using honeycomb core is to save weight. However, besides the weight saving, honeycomb offers other advantages that supersede other types of cores including high-stiffness-to weight ratio, smooth skins and excellent fatigue resistance (Bitzer1997, Schwingshackl et al. 2006). If the web spacing is large in either web and corrugated cores, the skins can deform under applied loads causing a wavy surface. However, due to the small size of cells of honeycomb core, the skins retain smooth surface under load (Bitzer 1996).

| TABLE 2.1  Honeycomb Sandwich Efficiency |
|-----------------------------|-----------------------------|-----------------------------|
| Relative Stiffness         | 1                          | 7                           | 37                          |
| Deflection (in.)           | 1.000                      | 0.140                       | 0.027                       |
| Relative bending strength  | 1                          | 3.5                         | 11.5                        |
| Weight (psf)               | 0.910                      | 0.978                       | 0.994                       |

In order to demonstrate the potential weight saving, Table 2.1 compares the strength and stiffness values of different honeycomb structures made using a 0.064 in. (1.6 mm) thick piece of aluminum split in half as the top and bottom facing of the sandwich. Using Equations 2.1 through 2.6, the results in Table 2.1 illustrate that
while the weight of the sandwich panel increased by 9 percent more than the original solid plate, its flexural stiffness and strength increased by 37 and 11.5 times respectively.

Foam core is another foam material that competes with honeycomb. Table 2.2 compares the properties of these core materials as reported by Bitzer (1996). The honeycomb strengths and shear moduli are considerably higher. Therefore, honeycomb core is the most optimum lightweighting alternative when core mechanical properties govern the sandwich design. Foam could be better used in lightly loaded panels and in insulating panels. However, honeycomb could be also used in the later situation by filling the cell with foam or another insulating material, which provides a good structural panel with fair insulating properties.

<table>
<thead>
<tr>
<th>Material</th>
<th>Compression</th>
<th>Tensile</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Density (pcf)</td>
<td>Strength (psi)</td>
</tr>
<tr>
<td>Aluminum honeycomb</td>
<td>3.1</td>
<td>300</td>
</tr>
<tr>
<td>Nomex honeycomb</td>
<td>3.0</td>
<td>325</td>
</tr>
<tr>
<td>Fiberglass honeycomb</td>
<td>3.0</td>
<td>410</td>
</tr>
<tr>
<td>Rohacell foam</td>
<td>3.1</td>
<td>128</td>
</tr>
<tr>
<td>Klegecell foam</td>
<td>3.0</td>
<td>69</td>
</tr>
<tr>
<td>Rigicell foam</td>
<td>3.0</td>
<td>80</td>
</tr>
<tr>
<td>Divinycell foam</td>
<td>3.1</td>
<td>100</td>
</tr>
</tbody>
</table>

2.3 Face Materials

The faces of a sandwich panel can be comprised of almost any material that is available in a thin sheet. This sole requirement allows many material options for the designer to utilize in sandwich panel construction. As described by Zenkert (1997), the parameters that are of primary concern for developing a structurally sound sandwich panel are

- High stiffness resulting in flexural rigidity
• High compressive and tensile strength
• Impact resistance
• Aesthetics
• Chemical and environmental resistance
• Wear resistance

The properties listed can be met by two different categories of face materials, metallic and non-metallic. Metallic face materials are most commonly sheet metals because of their geometry and applicability to a sandwich composite design. The advantages to using a metallic face sheet are low cost, good strength and stiffness, and high impact properties.

Non-metallic face materials are defined by fiber reinforced polymers (FRP). FRP are composed of fibers and matrix that define the traditional composite material. Typical fibers are glass, aramid, and carbon. These fibers are combined with a matrix by one of the manufacturing methods previously discussed to form an FRP composite. Orienting the fibers in the direction of applied loads utilizes their high stiffness and strength properties and tailors the composite laminate to resist and sustain loads. Having the ability to directionally tailor the stiffness and strength of a composite allows for reduction of material in directions that do not experience loads, this ultimately reduces the material being used (cost) and weight.

Lightweight, high strength and stiffness composite materials have been envisioned as a key cross-cutting technology for reinventing energy efficient transportation, providing new mechanisms for storing and transporting reduced carbon fuels, and increasing renewable power production (TMS Energy 2012). Fiber reinforced polymer composites can be used in vehicles, industrial equipment, wind turbines, compressed gas storage, buildings and infrastructure, and many other applications. One industry analysis predicts the global carbon fiber polymer composite market alone to grow to $25.2 billion in 2020 (Industry Experts 2013) and glass fiber reinforcements to reach a value of $16.4 billion by 2016 (Industry Experts 2012).

Fiber reinforced polymer composite materials have traditionally been used in defense, aerospace and other high value, low volume applications where higher costs and longer production cycle times can be tolerated because of the high performance
design requirements and resulting high value add of composites in the end-use products (National Research Council 2005). Improvements to materials and manufacturing techniques have led to increased use of fiber reinforced polymer composites in other industries such as sports equipment. However, they have not yet surpassed the tipping point to meet production volumes and cost targets to support widespread adoption in various industrial applications, where the application of composite materials might have significant impact in energy sectors. The energy intensity of carbon fiber composites and the lack of recyclability for fiber reinforced polymer composites are further limitations to the use of these materials.

Figure 2.3  Specific Stiffness and Specific Strength for Various Materials (University of Cambridge 2002).
Fiber reinforced polymer composites (GFRP and CFRP) have superior strength and stiffness to density ratios relative to other materials as shown in Figure 2.3. Carbon fiber reinforced polymer (CFRP) composites offer the highest structural properties to density ratios (specific strength is axial tensile strength divided by density and specific stiffness is axial modulus divided by density), excellent corrosion resistance and other desirable properties but are costly relative to other materials on a weight basis. Glass fiber reinforced polymer composites (GFRP) have improved specific mechanical properties over metals and cost less than carbon fiber composites but have lower strength to weight ratio and are not as stiff as carbon fiber composites. Table 2.3 provides further data for GFRP, CFRP and common metals including estimated embodied energy and production costs.

The use of composite materials and structures can lead to significant life-cycle energy benefits by reducing oil consumption in transportation.
Chapter 3

Unit Load Device Design

3.1 Introduction

Unit Load Devices (ULDs) and other airplane cargo restraint devices are composed of two general categories, primary and supplemental. An air carrier should have procedures to control the airworthiness and subsequent operational serviceability of ULDs and other restraint devices whether used as a primary or a supplemental restraint.

In the United States, ULDs should meet the requirements of Technical Standard Order TSO-C90, Cargo Pallets, Nets, and Containers; or other FAA-approved certification requirements. ULDs that are designed and manufactured to meet the aforementioned requirements are called “Certified ULDs”. Containers that are designed to meet different design criteria are considered “Uncertified ULDs”. These design criteria may be industry standards such as Society of Automotive Engineers (SAE) Aerospace Standard (AS) 1677, General Requirements for Uncertified Cargo/Baggage Containers; International Standards Organization (ISO) Publication No. 4118, Non-certified Lower-deck Containers for Air Transport; International Air Transport Association (IATA) ULD Technical Manual (UTM) 50; or other FAA-accepted standard.

3.2 Design Requirements

SAE AIR36106A (SAE 2014) provides a process to determine the performance requirements and test parameters for airworthiness approval of a unit load device. This process determines the ultimate load capabilities under defined restraint conditions, for airworthiness approval under a ULD configuration of Technical Standard Order TSO C90. This process is independent from the aircraft type that will carry the unit load device. For example, The LD3 container is designed to meet TSO-C90c and NAS-3610 revision 10 (type 2K2C) load requirements and has a
maximum gross weight of 3,500 lbs. The actual gross weight limits for this container, in a given airplane, are determined in compliance with FAR 25 and listed in the Approved Weights and Balance Manual for that airplane. TABLE 3.1 shows the ultimate load criteria for the LD3 Container.

<table>
<thead>
<tr>
<th>Ultimate Load N (lb)</th>
<th>CG height mm (in)</th>
<th>CG eccentricity %</th>
</tr>
</thead>
<tbody>
<tr>
<td>Forward 23,350</td>
<td>43,600</td>
<td>±10</td>
</tr>
<tr>
<td>Aft 23,350</td>
<td>79,400</td>
<td>±10</td>
</tr>
<tr>
<td>Side 23,350</td>
<td>864</td>
<td>±10</td>
</tr>
<tr>
<td>Up 43,600</td>
<td>(9,800)</td>
<td>±10</td>
</tr>
<tr>
<td>Down 79,400</td>
<td>(17,850)</td>
<td>±10</td>
</tr>
</tbody>
</table>

The ULD ultimate load is defined as the maximum expected limit load multiplied by a factor of safety of 1.5. Conversely, since a ULD is tested only to ultimate load, the maximum limit load it is approved for in a given direction is the UC Table's ultimate load divided by 1.5.

The specified ultimate loads should be applied with the maximum specified Centre of Gravity (CG) height and horizontal eccentricities. The specified CG height was determined in accordance with the worst case for a given base size, i.e., 864 mm (34 in) for base sizes capable of lower deck carriage [1625 mm (64 in) contour height] only and 1218 mm (48 in) for base sizes capable of main deck carriage [2438 mm (96 in) or more height]. Where a container's contour or a net's size allows only a lower load height, the testing CG height may be reduced to 55% of the maximum height of the container or net contour.

### 3.2.1 Base Performance

Unit load devices are designed to have a minimum area load capacity of 10 kPa (209 lb/ft²). The base edges shall have a minimum vertical stiffness EI value of $5 \times 10^7$ N.cm² ($1.75 \times 10^6$ lb.in²).
3.3 Theoretical Model

The distribution of the internal forces in a plate subjected to lateral loading under different edge support conditions is dealt with in different textbooks (Timoshenko 1959; Ventsel and Krauthammer 2001).

For a rectangular plate subjected to a distributed load, p and having any boundary conditions, the internal forces and stresses in any direction can be obtained mathematically according to the following assumptions and basic equations.

\[
M_x = -D \left( \frac{\partial^2 w}{\partial x^2} + v \frac{\partial^2 w}{\partial y^2} \right) \quad \text{.................................. (3.1)}
\]

\[
M_y = -D \left( \frac{\partial^2 w}{\partial y^2} + v \frac{\partial^2 w}{\partial x^2} \right) \quad \text{.................................. (3.2)}
\]

\[
M_{xy} = M_{yx} = -D (1-v^2) \frac{\partial^2 w}{\partial x \partial y} \quad \text{.................................. (3.3)}
\]

Figure 3.1 Internal Forces on the Plate
The above equations give the values of the bending moments, torsional moments, and the shearing forces, shown in Figure 3.1, at any section of the plate in terms of the deflection, \( w \).

The load \( p \) may be assumed as a sum of distributed loads in the two directions \( x \) and \( y \) so that:

\[ p_x = \text{part of the load transmitted in the direction of the } x\text{-axis}. \]

\[ p_y = \text{part of the load transmitted in the direction of the } y\text{-axis}. \]

\[ p = p_x + p_y \tag{3.6} \]

Knowing that

\[ \frac{\partial Q_x}{\partial x} + \frac{\partial Q_y}{\partial y} + p = 0 \tag{3.7} \]

Equations 3.6 and 3.7 can be identical if:

\[ p_x = -\frac{\partial Q_x}{\partial x} \quad \text{and} \quad p_y = -\frac{\partial Q_y}{\partial y} \]

Substituting for \( Q_x \) and \( Q_y \) from Equations 3.4 and 3.5 yields that:

\[ p_x = D \left( \frac{\partial^4 w}{\partial x^4} + \frac{\partial^4 w}{\partial x^2 \partial y^2} \right) \tag{3.8} \]

and

\[ p_y = D \left( \frac{\partial^4 w}{\partial y^4} + \frac{\partial^4 w}{\partial x^2 \partial y^2} \right) \tag{3.9} \]

Adding these two equations:
\[ \frac{\partial^4 w}{\partial x^4} + 2 \frac{\partial^4 w}{\partial x^2 \partial y^2} + \frac{\partial^4 w}{\partial y^4} = \frac{p}{D} \]  \hspace{1cm} (3.10)

Which gives the differential equation of the elastic surface of a plate loaded perpendicular to its plane.

In applying this method to uniformly loaded and simply supported rectangular plates, a further simplification can be made by taking the solution of Equation 3.10 in the form

\[ w = w_1 + w_2 \]  \hspace{1cm} (3.11)

And assuming \( w_1 \) in the form of

\[ w_1 = \frac{p}{24D} \left( x^4 - 2ax + a^3 x \right) \]  \hspace{1cm} (3.12)

Where \( w_1 \) represents the deflection of a uniformly loaded strip parallel to the \( x \)-axis. The expression in Equation 3.12 satisfies Equation 3.10 as well as the boundary conditions at the edges \( x = 0 \) and \( x = a \).

The expression \( w_2 \) evidently has to satisfy the equation

\[ \frac{\partial^4 w_2}{\partial x^4} + 2 \frac{\partial^4 w_2}{\partial x^2 \partial y^2} + \frac{\partial^4 w_2}{\partial y^4} = 0 \]  \hspace{1cm} (3.13)

and must be chosen in such a manner as to make the sum of Equation 3.11 satisfy all boundary conditions of the plate. Taking

\[ w_2 = \sum_{m=1}^{\infty} Y_m \sin \frac{m\pi x}{a} \]  \hspace{1cm} (3.14)

in which, from symmetry, \( m = 1, 3, 5, \ldots \) and substituting it into Equation 3.13, we obtain

\[ \sum_{m=1}^{\infty} \left( Y_m^{IV} - \frac{2 m^2 \pi^2}{a^2} Y_m^{II} + \frac{m^4 \pi^4}{a^4} Y_m \right) \sin \frac{m\pi x}{a} = 0 \]  \hspace{1cm} (3.15)

This equation can be satisfied for all values of \( x \) only if the function \( Y_m \) satisfies the equation

\[ Y_m^{IV} - \frac{2 m^2 \pi^2}{a^2} Y_m^{II} + \frac{m^4 \pi^4}{a^4} Y_m = 0 \]  \hspace{1cm} (3.16)
The integral form of this equation can take the form:

\[
Y_m = \frac{pa^4}{D} \left\{ A_m \cosh \frac{m\pi y}{a} + B_m \sinh \frac{m\pi y}{a} \\
+ C_m \sinh \frac{m\pi y}{a} + D_m \cosh \frac{m\pi y}{a} \right\} 
\]

\[ \cdots \cdots \cdots (3.17) \]

Since the deflection surface of the plate is symmetrical with respect to the \( x \)-axis as depicted in Figure 3.2, the expression should be only even functions of \( y \). Thus, the integration constants \( C_m = D_m = 0 \).

The deflection surface is then represented by the following expression

\[
w = \frac{p}{24D} \left( x^4 - 2ax + a^3 x \right) + \frac{pa^4}{D} \sum \left( A_m \cosh \frac{m\pi y}{a} + B_m \sinh \frac{m\pi y}{a} \right) \sin \frac{m\pi x}{a}
\]

\[ \cdots \cdots \cdots (3.18) \]

Which satisfies Equation 3.10 and also the boundary conditions at the sides \( x = 0 \) and
x = a. The integration constants \( A_m \) and \( B_m \) can be obtained by satisfying the boundary conditions at the sides \( y = \pm b/2 \):

\[
\frac{\partial^2 w}{\partial y^2} = 0 \quad \text{and} \quad w = 0
\]

\[\text{............. (3.19)}\]

The deflection surface takes the form

\[
w = \frac{pa^4}{D} \sum_{m=1}^{\infty} \left( \frac{4}{\pi^5 m^5} + A_m \cosh \frac{m\pi y}{a} + B_m \frac{m\pi y}{a} \sinh \frac{m\pi y}{a} \right) \sin \frac{m\pi x}{a}
\]

\[\text{............. (3.20)}\]

Where \( m = 1, 3, 5, \ldots \) etc. This yields that

\[
A_m = -\frac{2(\alpha_m \tanh \alpha_m + 2)}{\pi^5 m^5 \cosh \alpha_m}
\]

\[\text{............. (3.21)}\]

\[
B_m = \frac{2}{\pi^5 m^5 \cosh \alpha_m}
\]

\[\text{............. (3.22)}\]

\[
\alpha_m = \frac{m\pi b}{2a}
\]

\[\text{............. (3.23)}\]

Therefore, the equation of the deflection surface is given by the equation

\[
w = \frac{4pa^4}{\pi^5 D} \sum_{m=1}^{\infty} \frac{1}{m^5} \left( 1 - \frac{\alpha_m \tanh \alpha_m + 2}{2 \cosh \alpha_m} \cos \frac{2\alpha_m y}{b} + \frac{\alpha_m}{2 \cosh \alpha_m} \frac{2y}{b} \sinh \frac{2\alpha_m y}{b} \right) \sin \frac{m\pi x}{a}
\]

\[\text{............. (3.24)}\]

The maximum deflection is obtained at the middle of the plate at \( x = a/2 \) and \( y = 0 \) by

\[
\frac{w_{\text{max}}}{w_{\text{max}}} = \frac{5pa^4}{384D} - \frac{4pa^4}{\pi^5 D} \sum_{m=1}^{\infty} \frac{(-1)^{(m-1)/2}}{m^5} \frac{\alpha_m \tanh \alpha_m + 2}{2 \cosh \alpha_m}
\]

\[\text{............. (3.25)}\]

The solution in the above equation is given in the form of

\[
w_{\text{max}} = \alpha \frac{pa^4}{D}
\]

\[\text{............. (3.26)}\]

Where \( \alpha \) is a numerical constant depending on the ratio \( b/a \) of the sides of the plate. Values of \( \alpha \) are given in Table 3.2. In a similar way, the bending moments \( M_x \) and \( M_y \) are calculated by substituting Equation 3.24 into Equations 3.1 and 3.2. The maximum
values of these moments are given by the expressions:

\[
(M_x)_{\text{max}} = \beta p a^2 \quad \text{…………. (3.27)}
\]

\[
(M_y)_{\text{max}} = \beta_1 p a^2 \quad \text{…………. (3.28)}
\]

The factors \(\beta\) and \(\beta_1\) are numerical factors depending on the ratio \(b/a\) and several of these values are given in TABLE 3.2.

Substituting Equation 3.24 into Equations 3.4 and 3.6, the general expressions for the shearing forces \(Q_x\) and \(Q_y\) are:

\[
Q_x = \frac{p x (a - 2x)}{2} - 2\pi^3 pa^3 \sum_{m=1}^{\infty} m^3 B_m \cos \frac{m\pi y}{a} \cos \frac{m\pi x}{a} \quad \text{…….. (3.29)}
\]

\[
Q_y = -2\pi^3 pa^3 \sum_{m=1}^{\infty} m^3 B_m \sin \frac{m\pi y}{a} \sin \frac{m\pi x}{a} \quad \text{…….. (3.30)}
\]

These shearing forces have their numerical maximum value at the middle of the side where

\[
(Q_x)_{x=0, y=0} = \frac{pa}{2} - \frac{4pa}{\pi^2} \sum_{m=1,3,5,...}^{\infty} \frac{1}{m^2 \cosh \alpha_m} = \gamma pa \quad \text{…………. (3.31)}
\]

\[
(Q_x)_{x=a/2, y=-b/2} = \frac{4pa}{\pi^2} \sum_{m=1,3,5,...}^{\infty} \frac{(-1)^{(m-1)/2}}{m^2 \tanh \alpha_m} = \gamma_1 pa \quad \text{…………. (3.32)}
\]

The numerical factors \(\gamma\) and \(\gamma_1\) are given in TABLE 3.2. The magnitude of the vertical reactions \(V_x\) and \(V_y\) along the plate boundaries is obtained by combining the shearing forces with the derivatives of the twisting moments. Along the sides \(x = 0\) and \(x = a\), these reactions can be represented in the form

\[
V_x = \left( Q_x - \frac{\partial M_{xy}}{\partial y} \right)_{x=0, x=a} = \pm \delta pa \quad \text{…………. (3.33)}
\]

And along the sides \(y = \pm b/2\), the forces \(V_y\) take the form

\[
V_y = \left( Q_y - \frac{\partial M_{xy}}{\partial x} \right)_{y=\pm b/2} = \pm \delta_1 pa \quad \text{…………. (3.34)}
\]

In which the factors \(\delta\) and \(\delta_1\) are numerical factors depending on the ratio \(b/a\) and on the coordinates of the points taken along the boundary. Numerical values for these
factors which correspond to the middle of the sides parallel to the $x$-axis are given in TABLE 3.2.

The pressure along the plate sides as well as the portion of the pressure produced by the twisting moment $M_{xy}$ along the sides are balanced by reactive forces concentrated at the plate corners. The magnitude of these forces is given by:

$$R = 2 \left( M_{xy} \right)_{x=a, y=b/2} = 2D (1 - \nu) \left( \frac{\partial^2 w}{\partial x \partial y} \right)_{x=a, y=b/2}$$

$$= \frac{4(1-\nu)pa^2}{\pi^3} \sum_{m=1,3,5,\ldots} \frac{1}{m^2 \cosh \alpha_m} [(1 + \alpha_m \tanh \alpha_m) \sinh \alpha_m - \alpha_m \cosh \alpha_m] = n pa^2 \quad \ldots (3.35)$$

These forces are directed downward and prevent the corner of the plate from rising up during bending. The values of the coefficient $n$ are given as a function of the values $b/a$ in TABLE 3.2.

**TABLE 3.2** Numerical Factors $\alpha$, $\beta$, $\gamma$, $\delta$, $n$ for Uniformly Loaded and Simply Supported Rectangular Plates ($\nu = 0.3$)

<table>
<thead>
<tr>
<th>$b/a$</th>
<th>$w_{\text{max}}$</th>
<th>$M_{x_{\text{max}}}$</th>
<th>$M_{y_{\text{max}}}$</th>
<th>$Q_{z_{\text{max}}}$</th>
<th>$Q_{y_{\text{max}}}$</th>
<th>$V_{z_{\text{max}}}$</th>
<th>$V_{y_{\text{max}}}$</th>
<th>$R$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.0</td>
<td>0.04046</td>
<td>0.0479</td>
<td>0.0479</td>
<td>0.338</td>
<td>0.338</td>
<td>0.420</td>
<td>0.420</td>
<td>0.065</td>
</tr>
<tr>
<td>1.1</td>
<td>0.0485</td>
<td>0.0554</td>
<td>0.0493</td>
<td>0.360</td>
<td>0.347</td>
<td>0.440</td>
<td>0.440</td>
<td>0.070</td>
</tr>
<tr>
<td>1.2</td>
<td>0.0564</td>
<td>0.0627</td>
<td>0.0501</td>
<td>0.380</td>
<td>0.353</td>
<td>0.455</td>
<td>0.453</td>
<td>0.074</td>
</tr>
<tr>
<td>1.3</td>
<td>0.0638</td>
<td>0.0694</td>
<td>0.0503</td>
<td>0.397</td>
<td>0.357</td>
<td>0.468</td>
<td>0.464</td>
<td>0.079</td>
</tr>
<tr>
<td>1.4</td>
<td>0.0705</td>
<td>0.0755</td>
<td>0.0502</td>
<td>0.411</td>
<td>0.361</td>
<td>0.478</td>
<td>0.471</td>
<td>0.083</td>
</tr>
<tr>
<td>1.5</td>
<td>0.0772</td>
<td>0.0812</td>
<td>0.0498</td>
<td>0.424</td>
<td>0.363</td>
<td>0.486</td>
<td>0.480</td>
<td>0.085</td>
</tr>
<tr>
<td>1.6</td>
<td>0.0830</td>
<td>0.0862</td>
<td>0.0492</td>
<td>0.435</td>
<td>0.365</td>
<td>0.491</td>
<td>0.485</td>
<td>0.086</td>
</tr>
<tr>
<td>1.7</td>
<td>0.0883</td>
<td>0.0908</td>
<td>0.0486</td>
<td>0.444</td>
<td>0.367</td>
<td>0.496</td>
<td>0.488</td>
<td>0.088</td>
</tr>
<tr>
<td>1.8</td>
<td>0.0931</td>
<td>0.0948</td>
<td>0.0479</td>
<td>0.452</td>
<td>0.368</td>
<td>0.499</td>
<td>0.491</td>
<td>0.090</td>
</tr>
<tr>
<td>1.9</td>
<td>0.0974</td>
<td>0.0985</td>
<td>0.0471</td>
<td>0.459</td>
<td>0.369</td>
<td>0.502</td>
<td>0.494</td>
<td>0.091</td>
</tr>
<tr>
<td>2.0</td>
<td>0.1013</td>
<td>0.1017</td>
<td>0.0464</td>
<td>0.465</td>
<td>0.370</td>
<td>0.503</td>
<td>0.496</td>
<td>0.092</td>
</tr>
<tr>
<td>3.0</td>
<td>0.1223</td>
<td>0.1189</td>
<td>0.0406</td>
<td>0.493</td>
<td>0.372</td>
<td>0.505</td>
<td>0.498</td>
<td>0.093</td>
</tr>
</tbody>
</table>
3.4 Failure Modes for Honeycomb Sandwich Structures

3.4.1 Failure Modes in the Skin

1. Face Yielding

Failure occurs in the top skin due to face yielding when the axial stress in either of the skins reaches the in-plane strength $\sigma_{fy}$ of the face material as illustrated in Figure 3.3.

![Figure 3.3 Face Yielding](image)

It is assumed that the skin behaves in a brittle manner. With a symmetrical sandwich panel, the stress is the same in the tension and compression faces. For composite face materials, the compressive face is generally the critical one (Petras 1998).

2. Intra-cell Dimpling

A sandwich with a honeycomb core may fail by buckling of the face where it is unsupported by the walls of the honeycomb as illustrated in Figure 3.4. Simple elastic plate buckling theory can be used to derive an expression for the in-plane stress $\sigma_\beta$ in the skins at which intra-cell buckling occurs as

$$\sigma_\beta = \frac{2E_{fx}}{1 - v_{fxy}^2} \left( \frac{2t_f}{\alpha} \right)^2$$

.................................................. (3.36)

Where: $\alpha$ is the cell size of the honeycomb (Petras 1998, Vinson 1999).

$E_{fx}$ and $v_{fxy}$ are the elastic modulus and Poisson’s ratio for the skin for loading in the axial direction. A similar expression, verified experimentally by Kuenzi (1951).
3. Face Wrinkling

Face wrinkling is a buckling mode of the skin with a wavelength greater than the cell width of the honeycomb. Buckling may occur either in towards the core or outwards, depending on the stiffness of the core in compression and the adhesive strength. Petras (1998) reported that inward wrinkling of the top skin occurs near the central load in three-point load tests. By modeling the skin as a plate on an elastic foundation, Allen (1969) expressed the critical compressive stress $\sigma_{fw}$ that results in wrinkling of the top skin as

$$\sigma_{fw} = \frac{3}{12(3-\nu_{czz})^2(1+\nu_{czz})^2} f_{cxz}^{1/3} E_{fx}^{2/3} \quad \text{.................. (3.37)}$$

Where: $\nu_{czz}$ is the out-of-plane Poisson’s ratio of the honeycomb core.

$E_3$ the out-of-plane Young’s modulus of the honeycomb core. This could
be estimated in terms of the density and elastic modulus of the honeycomb material by the rule of mixture expression

\[ E_s = \frac{\rho_s E_s}{\rho_s} \]

### 3.4.2 Core Failure

Honeycomb sandwich structures loaded in bending can fail due to core failure. There are two pertinent failure modes namely shear failure or indentation by local crushing in the vicinity of the loads.

1. **Core Shear**

   Similar to the I-beam, the shear stress varies through plate thickness in a parabolic way resulting in a large drop at the interface between the face and core. If the faces are much stiffer and thinner than the core, the shear stress can be taken as linear through the face and constant in the core. Neglecting the contribution from the skins, the mean shear stress in the core is given by

   \[ T_{ex} = \frac{V}{2d} \]  

   \[ \text{Figure 3.6 Core Shear Failure} \]

   Assuming brittle behavior, failure occurs when the applied shear stress \( T_{ex} \) equals the shear strength \( T_{cs} \) of the honeycomb core in this direction.
\[ T_{\text{ax}} = T_{\text{ex}} \]

This failure mode is very common in sandwich panels with low density Nomex core due to the anisotropy of the honeycomb.

2. Core Crushing

This failure mode occurs in sandwich panels subjected to concentrated loads at the point of load application due to core crushing. The bending stiffness of the skin and the core stiffness determine the degree to which the load is spread out at the point of application (Ciba 1995; Petras 1999).

![Core Crushing Failure](image)

Figure 3.7 Cure Crushing Failure

3.5 Lightweight Design of LD-3 Base

Lightweight, high strength composite sandwich panels were developed and utilized to build the scaled model air cargo prototype. The panel utilizes two epoxy-carbon fiber composite skin plates bonded to a Nomex aramid fiber reinforced honeycomb core. Nomex honeycomb core HD-1/8-3.0 is an extremely lightweight, high strength and nonmetallic product manufactured with aramid fiber paper impregnated with a heat resistant phenolic resin as shown in Figure 3.8. Aramid paper has been used in boat hulls, auto racing bodies and military shelters (The Gill Corporation 2015). This core was selected based on their high strength to weight ratio and good fatigue and impact resistance. The mechanical properties of the Nomex honeycomb core material selected for this study is summarized in TABLE 3.3.

The compressive strength 107 psi (15,408 lb/ft²) of the selected core far exceeds the
minimum area load capacity of 209 lb/ft² (10 kPa) specified for the air cargo containers.

TABLE 3.3 Properties of Nomex Honeycomb Core

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cell Size, in. (mm)</td>
<td>1/8 (3.2)</td>
</tr>
<tr>
<td>Density lb/ft³ (kg/m³)</td>
<td>3.0 (29)</td>
</tr>
<tr>
<td>Compressive Strength, psi (MPa)</td>
<td>309 (2.13) TYP</td>
</tr>
<tr>
<td></td>
<td>263 (1.81) MIN</td>
</tr>
<tr>
<td>L-Direction Shear Strength, psi (MPa)</td>
<td>224 (1.54) TYP</td>
</tr>
<tr>
<td></td>
<td>192 (1.32) MIN</td>
</tr>
<tr>
<td>L-Direction Shear Modulus, ksi (GPa)</td>
<td>7.26 (0.050)</td>
</tr>
<tr>
<td>W-Direction Shear Modulus, psi (GPa)</td>
<td>109 (0.75) TYP</td>
</tr>
<tr>
<td></td>
<td>193 (0.64) MIN</td>
</tr>
<tr>
<td>W-Direction Shear Modulus, ksi (GPa)</td>
<td>3.97 (0.027)</td>
</tr>
</tbody>
</table>

Each of the two facings comprised four 0.118 kg/m² (3.5 oz/sq yd) woven carbon fiber layers with a total thickness of 0.91 mm (0.036 in.). This ultralight carbon fabric is found suitable for applications which call for maximum strength and stiffness. The plain weave construction delivers uniform strength in both directions, and provides for excellent stability and easy handling. This fabric is also suitable for
aerospace, UAVs, competition auto and marine, and light industrial applications. The minimum reported tensile strength for this fabric is 3.5 GPa (510 ksi) and the elastic modulus is 227.5 GPa (33,000 ksi). A light amber laminating, medium viscosity epoxy resin is used for fabricating all the panels.

The properties of the carbon-epoxy laminate used for this design are taken as:

**Carbon / Graphite Fabric**

Carbon/Epoxy Fabric, fiber volume fraction 50%.

\[
\begin{align*}
F_{1t} &= 80 \text{ ksi} \\
F_{1c} &= 113.0 \text{ ksi} \\
F_{2t} &= 82.5 \text{ ksi} \\
F_{2c} &= 98.6 \text{ ksi} \\
E_{11} &= 7.9 \text{ Msi} \\
E_{22} &= 7.83 \text{ Msi} \\
G_{12} &= 0.59 \text{ Msi} \\
\nu_{12} &= 0.065
\end{align*}
\]

**Design Calculations**

The LD-3 base is 61.5 in × 60.4 in.

\[
b/a = \frac{61.5}{60.4} = 1.018
\]

From TABLE 3.2, for the b/a = 1.018

\[
\begin{align*}
\alpha &= 0.004355; \\
\beta &= 0.048289; \\
\beta_1 &= 0.048277 \\
\delta &= 0.424051; \\
\delta_1 &= 0.424218
\end{align*}
\]

(a) **Facing Bending Stress**

\[
M_{max} = \beta pa^2
\]
Check that stress is less than the critical stresses for intra-cell buckling

\[
\sigma_f = \frac{2E_{es}}{1 - \nu^2_{es}} \left( \frac{2f}{\alpha} \right)^2 = \frac{2(7.83 \times 10^3)}{1 - (0.065)^2} \left( \frac{2 \times 0.036}{0.25} \right)
\]

= 1,304 ksi

Check that stress is less than the critical stresses for face wrinkling

\[
\sigma_{fw} = \frac{3}{12(3 - \nu_{cxz})^2 (1 + \nu_{cxz})^2} E_{li}^{1/3} E_{3}^{2/3}
\]

For the Nomex material, \( \nu_{cxz} = \nu_s = 0.4 \)

And \( \rho_s = 45.18 \text{lb/ft}^3 \)

\[
E_{33} = \frac{3}{45.18} (130.534) = 8.668 \text{ksi}
\]

\[
= \frac{3}{12(3 - 0.4)^2 (1 + 0.4)^2} (7,900)^{1/3} (8.668)^{2/3}
\]

= 1,365 ksi

Therefore, yielding is the critical failure mode for the proposed face sheet. The factor of safety for bending is

\[
FOS = \frac{80,000}{9,056} = 8.83
\]

(b) **Core Shear Stress**

\[
V_s^\text{max} = V_L = \delta L a
\]

\[
= (0.4242)(0.9676)(60.4) = 24.78 \text{ lb/in.}
\]

\[
\tau_c^L = \frac{V_L}{bh} = \frac{24.78}{(1)(0.572)} = 46.24 \text{ psi}
\]
\[ FOS = \frac{224}{46.24} = 4.84 \]

\[ V_y^{\text{max}} = V_y = \delta p \]

\[ = (0.4240)(0.9676)(60.4) = 24.79 \text{ lb/in.} \]

\[ \tau_e^W = \frac{V_{WL}}{bh} = \frac{24.78}{(1)(0.572)} = 46.25 \text{ psi} \]

\[ FOS = \frac{109}{46.25} = 2.36 \]

The shear controls the design of this proposed panel.

(c) Panel Deflection

\[ D_{\text{panel}} = \frac{E_f t_f h^2}{2(1-\nu_f^2)} \]

\[ D_{\text{panel}} = \frac{(10.20 \times 10^6)(0.036)(0.536)^2}{2(1-(0.33)^2)} = 59,194 \]

\[ w_{\text{max}} = \frac{\alpha p a^4}{D} \]

\[ w_{\text{max}} = \frac{(0.0043545)(0.9676)(60.4)^4}{(59,194)} = 0.94 \text{ inch} \]

The weight of the eight epoxy carbon fiber laminates = 5.02 lb

The weight of the 0.5 inch thick honeycomb core = 3.22 lb

The overall weight of the base panel = 8.22 lb.

The current design configuration for LD-3 base utilizes either an aluminum plate aluminum plate or glass fiber composite plate. The amount of information about the structural design details of the existing air cargo containers is very limited. The lightest tare weight reported for a classic aluminum LD-3 container is 76 kg (186 lb). Such a container utilizes a 0.1 inch (2.5 mm) thick aluminum base plate whose weight
is 36.2 lb (16.5 kg), which represents approximately 22 percent of the container weight (Nordisk 2016). This 2.5 mm thick aluminum plate needs to be stiffened in order to meet the strength and stiffness requirements of the ULDs, but such details are not available. The thickness of the glass fiber composite plate could not be found in the literature.

The results above reveal that the current weight of a baseline LD-3 air cargo container can be reduced by as much as 77 percent when the aluminum plates are replaced, through an integrated design approach, by a composite sandwich panel. The concept could be also extended to redesign the whole assembly of the container. The joining concept developed during prototyping by flanging the sidewalls and bonding or clamping them to the base would allow for fastener-free joints and reduce weight associated with joining hardware.
Chapter 4
Prototyping of Lightweight ULD

4.1 Introduction

This Chapter describes the construction of a scaled prototype of a ULD-3 air cargo container. The main purpose of building a solid model is to explore the potential benefits and drawbacks of various joining configurations and sandwich composite implementation. The construction of such a model provides reliable, extensive data for comparative assessments of alternative manufacturing and joining methods as well as material selection. The manufacturing and close examination of such a scaled model is necessary in order to reduce the cost of tooling and materials that to be used at a later stage, for producing full-scale prototypes.

The primary design criteria guiding the fabrication of a scaled trailer prototype are the achieving of optimal tradeoffs between structural weight and performance, based on extensive use of lightweight, strong and durable components, connected by fastener-free joints that allow easy assembly and maintenance. This approach has been proved to be cost effective and provide the means to implement high performance advanced sandwich structures into the model design after the initial fabrication process has been completed and studied (Prucz et al. 2004, 2006, 2009).

This section presents a brief description of the manufacturing techniques that are available at West Virginia University and have been used in constructing the scaled model.

4.1.1 Hand Lay-up

The hand lay-up process, also referred to as a wet lay-up, combines the reinforcement fibers with a liquid resin in a mold. Layers of fibers are placed into the mold and saturated with the resin. The part is hand rolled to create a uniform resin coat and extract any voids or air pockets within the combination. Layers are added until the thickness or desired orientation of fibers is reached. The curing process is
the final stage of the hand lay-up manufacturing; it involves the chemical process of the resin changing state from a liquid to a solid (Barbero 1998).

The lay-up process begins with the development of a proper mold to accommodate the desired part geometry and requirements of the curing process. The material used for a mold depends on the number of times the mold will be used, temperature and pressure of the curing process, and the manufacturing of the mold itself. To avoid the resin curing to the mold and damaging the finished part by forced removal, a release agent is applied to the areas where the mold and resin come into contact. Common release agents are wax, poly vinyl alcohol, silicones, and release fabric.

The fibers are then placed on the mold to be saturated with resin. The proper measurements of mixing ratio of the resin and catalyst must be carefully followed and mixed thoroughly before application. After the different layers of fabric have been applied to the mold and saturated with the resin, hand rollers are used to compress the layers together and against the mold. Hand rolling of the lay-up ensures removal of any air pockets that will become voids during the curing process if not removed. The curing process is usually done at room temperature. However, elevated pressures are sometimes applied to the part during the curing process to remove excess resin and air via bag molding (Barbero 1998).

### 4.1.2 Bag Molding

Pressure can be applied to a laminate during the curing process by using bag molding techniques. Vacuum bagging uses a flexible plastic or bag that is placed over the laminate and sealed. A vacuum pump is connected so the air is pumped out from the inside of the bag which ultimately applies a uniform pressure onto the top surface of the laminate. The pressure forces the laminate against the mold creating an accurate resemblance to the mold geometry while removing excess resin and air [3].

The three main methods of applying a pressure to a laminate are by pressure bag, vacuum bag, and autoclave manufacturing. Vacuum bagging is a popular manufacturing process because it is relatively inexpensive, allows large size parts to be manufactured, and the quality of the resulting part is mainly dependent on the manufacturer’s skill and not a machining process.
4.2 Prototyping

Lightweight, high strength composite sandwich panels were developed and utilized to build the scaled model air cargo prototype. The panel utilizes two epoxy-carbon fiber composite skin plates bonded to a Nomex aramid fiber reinforced honeycomb core. Nomex honeycomb core HD-1/8-3.0 is an extremely lightweight, high strength and nonmetallic product manufactured with aramid fiber paper impregnated with a heat resistant phenolic resin as shown in Figure 4.1. Aramid paper has been used in boat hulls, auto racing bodies and military shelters. This core was selected based on their high strength to weight ratio and good fatigue and impact resistance.

![Nomex Honeycomb Core Used in Prototyping](image)

Each of the two facings comprised four 0.118 kg/m² (3.5 oz/sq yd) woven carbon fiber layers with a total thickness of 0.91 mm (0.036 in.). This ultralight carbon fabric is found suitable for applications which call for maximum strength and stiffness. The plain weave construction delivers uniform strength in both directions, and provides for excellent stability and easy handling. This fabric is also suitable for aerospace, UAVs, competition auto and marine, and light industrial applications. Figure 4.2 illustrates the finished sandwich composite panel.

Each laminate was oversaturated with epoxy resin using a brush and they were wet-laid up. The process starts with laying up the bottom face laminates and placing them over a waxed smooth aluminum plate. The honeycomb core is placed on top.
and finally the top facing. A release peel ply was placed on top of the laminate followed by vacuum bagging film. The release peel ply provides an easy release barrier between the laminate surface and the breather and bleeder layer that traps and holds the excess resin from the laminate. Vacuum connector is placed at a corner to connect the bag to vacuum tubing to the pump. For this purpose, an 1/8 HP vacuum pump was used.

Figure 4.2  Finished Composite Sandwich Panel for ULD Base
A prototype of an air cargo was constructed at a 1 to 6 scale. The side walls of the prototype were made as a thin epoxy-carbon fiber composite laminate, however they could be also made of sandwich composite similar to the base as illustrated in Figure 4.3. The two sides were made flanged as shown in order to be clamped to the main part for the ease of assembly. The assembled sides of the scaled model are shown in Figure 4.4.

Figure 4.3 Side Walls of the Air Cargo Container

Figure 4.4 Full Scaled Model of Air Cargo Container
The building of the prototype model was performed in distinctive phases in order to allow continual assessment of the feasibility, potential advantages and disadvantages of different design configurations. Phasing of the fabrication process allowed for incremental improvements in the design and fabrication concepts. The first phase was the construction of the base of the container using the lightweight sandwich panel by utilizing vacuum bagging process. The process of fabricating this section progressed into the side walls and provided an effective method to culminate the full model design.

The side panels have been bonded to the base through the flanged edges in order to secure the integrity of such joints. Furthermore, this approach would allow structural flexibility and effectively absorb typical static, thermal, and dynamic forces associated with typical loading scenarios. However, other mechanical joining options by clipping were investigated in order to make the container collapsible as needed for more flexibility.

A clip joint was made to join the sides of the container to the base and roof assembly shown in Figure 4.4. The proposed joint is shown in Figure 4.5. The side panels can be attached permanently through the joint by the use of adhesives or the joint could be used as it is a mechanical joint to allow for a collapsible container.

Figure 4.5  Clip Joint for Attaching Side Walls
Chapter 5

Summary and Conclusions

This study aims at developing innovative, lightweight design concepts for air cargo containers that would allow for weight reduction in the air cargo transportation industry. For this purpose, innovative design and assembly concepts of lightweight design configurations of air cargo containers have been developed through the applications of lightweight composites. A scaled model prototype of a typical air cargo container was built to assess the technical feasibility and economic viability of creating such a container from fiber-reinforced polymer (FRP) composite materials. Based on the results of this study, the following conclusions could be drawn:

- The current weight of a baseline LD-3 air cargo container can be reduced by as much as 75 percent when the aluminum plates are replaced, through an integrated design approach, by a composite sandwich panel. The concept could be also extended to redesign the whole assembly of the container.

- According to CargoComposites (2015), assuming the fuel price cost of $2.86/gallon, the expected cost saving for 30 10-hour round-trips is $685,600. This would achieve significant savings in fuel cost that would recover any additional cost in the original container price.

- The joining concept developed during prototyping by flanging the sidewalls and bonding or clamping them to the base would allow for fastener-free joints and reduce weight associated with joining hardware.
REFERENCES


2. Air China Cargo


4. Air New Zealand
http://www.airnewzealand.com/international-cargo-containers

5. ANA Cargo http://www.anacargo.jp/ja/int/service/uld/container.html


9. Boeing


11. Cathay Pacific Cargo AKE


14. Dragonair Cargo
http://www.dragonaircargo.com/usrapps/content/shippingSpec/unitLoadDevices/uld.aspx?type=container&category=AKE

15. DSV Global Transport and Logistics
http://www.us.dsv.com/air-freight/unit-load-devices/LD3-AKE-AVE-container
17. Emirates Sky Cargo  
29. Nippon Cargo Airlines  
   [http://www.nca.aero/e/service/freighter/container.html#05](http://www.nca.aero/e/service/freighter/container.html#05)
32. Profreight
   http://www.profreight.co.nz/Useful+Info/Airline+Pallet+Dimensions.html


36. Quantum Transportation LTD

37. Royal Jordanian Cargo


41. Shapiro AKE


44. Society of Automotive Engineers (2012). Air Cargo Unit Load Devices –


49. TKM Global [http://www.tkmglobal.net/air_freight.htm](http://www.tkmglobal.net/air_freight.htm)

50. Turkish International Forwarding & Logistic Services [http://www.turkishtransporters.com/container-pallet.htm](http://www.turkishtransporters.com/container-pallet.htm)


