

ISTANBUL TECHNICAL UNIVERSITY★GRADUATE SCHOOL OF
SCIENCE ENGINEERING AND TECHNOLOGY

**PATTERN AND RIGIDITY EFFECTS OF ADHESIVE ON THE STRESS
DISTRIBUTION OF THE ADHESIVELY BONDED COMPOSITE JOINTS**



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VE RİJİTLİĞİNİN GERİLME DAĞILIMI ÜZERİNDEKİ ETKİSİ**

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To my family,



FOREWORD

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ABBREVIATIONS

FE	: Finite Element
FEM	: Finite Element Method
SLJ	: Single Lap Joint
DLJ	: Double Lap Joint
TBJ	: Tensile Butt Joint
OMCs	: Organic Matrix Composites
MMCs	: Metal Matrix Composites
CMCs	: Ceramic Matrix Composites
FRCs	: Fiber Reinforced Composites
PMCs	: Polymer Matrix Composites
PEEK	: Polyether-ether-ketone
PPS	: Polyphenylene sulfide



SYMBOLS

k	: Bending moment factor
M_0	: Bending moment
T	: The load applied to the adherends
t	: Adherends thickness
E_1	: Longitudinal elastic modulus
E_2	: Transverse elastic modulus
E_f	: Elastic modulus of fiber
E_m	: Elastic modulus of matrix
V_f	: Volume fraction of fiber
V_m	: Volume fraction of matrix
A	: Cross section area
A_f	: Cross section area of fiber
A_m	: Cross section area of matrix
ν_{12}	: Major Poisson's ratio
G_{12}	: Shear modulus
γ	: Shear strain
γ_f	: Shear strain of fiber
γ_m	: Shear strain of matrix
Δ	: Total shearing deformation
Δ_f	: Shearing deformation of fiber
Δ_m	: Shear deformation of matrix
ϵ_1	: Longitudinal strain
ϵ_2	: Transverse strain
ϵ_f	: Longitudinal strain of fiber
ϵ_m	: Transverse strain of matrix
σ_1	: Longitudinal stress
σ_2	: Transverse stress
τ	: Longitudinal stress
P	: Force
W	: Transverse dimension of symbolic volume element
σ_{ij}	: Stress components
ϵ_{ij}	: Strain components
C_{ij}	: Stiffness matrix
S_{ij}	: Compliance matrix
Q_{ij}	: Reduced stiffness matrix
γ_{ij}	: Shear strain components
θ	: Fiber angle
T	: Transformation matrix
R	: Reuter's transformation matrix
$\bar{Q}_{ij}$	: Transformed reduced stiffness matrix
$(\sigma_1^T)_{ult}$	: Ultimate longitudinal tensile strength (in direction 1)
$(\sigma_1^C)_{ult}$	: Ultimate longitudinal compressive strength (in direction 1)
$(\sigma_2^T)_{ult}$	: Ultimate transverse tensile strength (in direction 2)

$(\sigma_2^C)_{ult}$	: Ultimate transverse compressive strength (in direction 2)
$(\tau_{12})_{ult}$	: Ultimate in-plane shear strength (in plane 1-2)
$(\epsilon_1^T)_{ult}$	: Ultimate longitudinal tensile strain (in direction 1)
$(\epsilon_1^C)_{ult}$	: Ultimate longitudinal compressive strain (in direction 1)
$(\epsilon_2^T)_{ult}$	: Ultimate transverse tensile strain (in direction 2)
$(\epsilon_2^C)_{ult}$	: Ultimate transverse compressive strain (in direction 2)
$(\gamma_{12})_{ult}$	: Ultimate in-plane shear strain (in plane 1-2)



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PATTERN AND RIGIDITY EFFECTS OF ADHESIVE ON THE STRESS DISTRIBUTION OF THE ADHESIVELY BONDED COMPOSITE JOINTS

SUMMARY

Adhesively bonded composite joints have attracted considerable attentions in industrial applications due to notable advantages such as, easy integrated manufacturing, low mounting weight, minimal stress concentration, smooth load transfer, and excellent fatigue and damage tolerance performances. However, the joints are potential weakness under load conditions due to discontinuities of structures. In adhesively bonded joints structures, stress concentrations are occurred both ends of the adhesive regarding to both literature and performed finite element model analysis stress results. The stress concentrations in all mechanical systems should be prevented and provided uniform stress distribution due to durability of the structures.

In this study the single lap joints and double lap joints are investigated at first. Stress distributions and deformations of both joints are discussed and presented in order to reveal difference behavior of the single lap joint and double lap joint. Unlike the double lap joints, the single lap joints have unbalanced structures. As a results of this, the higher stress levels are observed at critical regions in the single lap joints. Therefore, the research concentrates on the single lap joints and reductions of the stress concentrations. The study suggests several ways to minimize the peak stress levels in the critical regions and provides more uniform stress distributions on the adhesives.

The pattern and rigidity effects of adhesive on the stress distribution are examined in details. The different adhesives which are ductile (low elastic modulus according to the general adhesives), general (normal elastic modulus for adhesives) and brittle adhesives (high elastic modulus according to general adhesives) and different shape of adhesives patterns are generated and performed analyses via finite element method in ABAQUS.

The general adhesive is used for only in master model. On the other hand, both ductile and brittle adhesives are used together in the other all configurations. Except master adhesive model, five different configurations are generated in order to realize the pattern and rigidity effects of the adhesive on the stress distribution. Due to revealing importance of the ratio of the ductile and brittle adhesive usage, different ratios of the ductile-brittle adhesive are used in the configurations. Three models of five have same adhesive pattern shape but different ratio of ductile-brittle adhesive usages. The other two models of five have also same adhesive pattern shape but different ratios of ductile-brittle adhesive usages, too. The purpose of different ratios of ductile-brittle adhesive usage is to define the most suitable configuration for stress reduction of the stress concentration regions.

The ductile adhesive is used for overlap specific ends region due to minimizing stress concentration by the help of more flexible structure. On the other hand, the brittle adhesive is used for overlap specific center region for shifting stresses in center region of the overlap and enhancing durability of the structure. There is a balanced ratio of the ductile-brittle adhesive usage for strength of the adhesive bonding joints.

In order to examine the stress concentrations changes for different adhesive and shapes, the finite element models for each configuration are generated and analyzed. All models' peel and shear stresses are compared with the general master adhesive model and each other.

As it is expected the maximum stress for both peel and shear stress are observed both ends of the adhesive for master adhesive model. On the other hand for both peel and shear stress levels of center region of overlap are almost zero, unlike the stress concentration region where both peel and shear stress levels are around 50 MPa and 40 MPa, respectively.

The stress reduction of the critical regions for different adhesives and pattern shapes models are varied 25% to 50% according to master adhesive model. Both peel and shear stress levels are around 30 MPa and 20 MPa, respectively for adhesive shape models. On the other hand, center region of the overlap, especially the shear stress levels are around 25 MPa. The shear stress levels are around 5 MPa for the same region of master model. The stress concentration regions shift towards the center region of the overlap. Therefore, the more uniform stress distributions can be obtained.

The other way to decreasing stress levels is taper adherends model and spew fillet model. Both models are examined in details and compared each other and master single lap joint model. Especially, the low stress levels in critical regions on the spew fillet model are observed according to master single lap joint and taper adherends model.

Finally, the fiber angles effects on the stress distributions of the adhesives are investigated and compared each other and master single lap joint model. The stress analysis results and stress distributions on the adhesives are effected significantly by fibers angles changing.

According to the orthotropic elastic mechanics theory and plane strain assumption, composite plies in all models are generated via the equivalent engineering constants which are obtained only by the material properties of unidirectional composites.

YAPIŞKAN BAĞLANTILI KOMPOZİT YAPILARDA YAPIŞKAN DESEN VE RİJİTLİĞİNİN GERİLME DAĞILIMI ÜZERİNDEKİ ETKİSİ

ÖZET

Yapışkan bağlantılı kompozit yapılar kolay üretilebilirlik, hafif montaj ağırlığı, gerilim dağılımının diğer bağlantı tiplerine oranla daha düzgün olması ve yük aktarımının homojenliği, yüksek yorulma ve hasar performansları sebebi ile endüstriyel uygulamalarda sıkça kullanılmaktadır. Ancak, bağlantı bölgelerinin her daim yapı için bir süreksizlik meydana getirdiğinden, yükleme altında potansiyel hasar bölgeleridir. Yapışkan bağlantılı kompozit yapıların yapışkan üzerindeki gerilme yığılmasının, literatür araştırmalarına paralel olarak gerçekleştirilen analizlerde de yapışkanın uçlarında yığıldığı gözlemlenmektedir. Sistemin dayanımı açısından bu gerilme yığılmasının olabildiğince üniform olması, tüm bağlantı sistemleri için temel hedefi oluşturmaktadır.

Bu çalışmada, ilk olarak tek tesirli ve çift tesirli bindirme bağlantıları incelenmiştir. Tek tesirli ve çift tesirli bindirme bağlantılarının çekme yükü altında farklı davrandıklarını göstermek için, her iki bindirme bağlantısı için gerilme dağılımları ve deformasyonları incelenmiştir.

Çift tesirli bindirmeli bağlantıların aksine, tek tesirli bindirme bağlantıları yükleme açısından daha dengesiz yapıya sahiptir. Bunun sonucu olarak, kritik bölgeler için tek tesirli bindirme bağlantılarında daha yüksek gerilme değerleri ortaya çıkmaktadır. Bu nedenle, araştırmada tek tesirli bindirme bağlantılarında, gerilme yığılmalarının azaltılması üzerine yoğunlaşmıştır. Ayrıca, yapışkan üzerindeki kritik bölgelerde yüksek gerilmeleri azaltıcı ve yapışkan üzerinde daha üniform bir gerilme dağılımı sağlamak amacıyla yöntemler önerilmektedir.

Bu yöntemlerinden ilki ve tezin temel araştırma konusu olan, yapışkan desen ve rijitliğinin gerilme dağılımı üzerindeki etkileri ayrıntılı olarak ele alınmıştır. Farklı yapışkanlar, bunlar düşük elastiklik modülüne sahip yapışkanlar, genel endüstriyel uygulamalarda kullanılan orta elastiklik modülüne sahip yapışkanlar ve yüksek elastiklik modülüne sahip daha rijit yapışkanlardır, farklı desen ve geometriye sahip modeller sonlu elemanlar yöntemi ile ABAQUS programında oluşturulmuştur.

Referans alınan temel model ile beraber birbirinden farklı beş adet konfigürasyon incelenmek üzere oluşturulmuştur. Baz alınan temel model ile beraber tüm modeller için sınır şartları, yapıştırılan kompozit parçanın toplam kalınlıkları, her bir tabakaya ait kalınlık, oryantasyon ve tabaka yerleşimi, fiber açıları, malzeme özellikleri, yapışkan kalınlıkları ve uygulanan kuvvetler aynıdır. Ancak beş adet konfigürasyonda, yapışkanın malzemesi, yapışkan desenleri ile beraber farklı yumuşak-sert yapışkan oranları oluşturularak analizler gerçekleştirilmiştir. Bu farklı yumuşak-sert yapışkan oranlarına sahip modeller ile kritik bölgelerdeki gerilme dağılımları değişimin incelenmesi amaçlanmakla beraber en uygun yumuşak-sert yapışkan oranına sahip model belirlenmeye çalışılmıştır.

Farklı yapışkan ve desene sahip tüm konfigürasyonlarda, elastik modülü daha düşük olan yapışkan (yumuşak yapışkan) bindirme bölgesi uçlarına belli desen ile yerleştirilirken, elastik modülü daha yüksek olan yapışkan (sert yapışkan), bindirme bölgesinin merkezine konumlandırılmıştır. Bu şekilde yerleştirilmesinin başlıca sebebi; temel alınan model incelendiğinde, bindirme bölgesi uçlarında gerilmelerin yüksek olmasına paralel olarak deformasyonlarında bu bölgelerde daha fazla ortaya çıkmasıdır. Deformasyonların daha fazla olduğu bölgelerde elastik modülü daha düşük yapışkan kullanılarak yapıya belli ölçülerde esneklik kazandırılıp, gerilme yığılmalarının önüne geçilmeye çalışılmıştır. Bu uçlarda kullanılan yumuşak yapışkanların yapının mukavemetini azaltıcı yöndeki etkisini minimize etmek adına, bindirme bölgesi merkezinde ortalama yapışkan türlerine göre daha yüksek elastik modülüne sahip yapışkan kullanılmıştır.

Tüm modeller için gerçekleştirilen sonlu elemanlar analizleri sonucunda, temel alınan model için beklenildiği üzere maksimum soyulma ve kayma gerilmeleri yapışkan uç bölgelerinde ortaya çıkmaktadır. Gerilme yığılmalarının değişimini incelemek adına farklı desen ve farklı yapışkanlar bir araya getirilerek oluşturulan sonlu elemanlar model konfigürasyonlarında ise, bindirme bölgesi uçlarında hem soyulma gerilmeleri hem de kayma gerilmeleri için, farklı konfigürasyonlara göre değişmekle birlikte yaklaşık olarak ortalama %25 ile %50 aralığında gerilme yığılmalarında düşüş gözlemlenmiştir.

Daha kapsamlı incelemek adına tüm modeller için, yapışkanın merkezinden geçen bir çizgi boyunca hem soyulma hem de kayma gerilmelerinin değişimi, grafik olarak çizdirilmiştir. Bunun yanında sadece bir doğru üzerinde gerilmeleri incelemenin yeterli olmayacağı düşünülerek, MATLAB programı aracılığı ile analiz sonuçları işlenerek yapışkanın yüzeyindeki gerilme değişimi üç boyutlu olarak gösterilmiştir. Bu çizdirilen grafiklere göre, temel alınan modelde gerilme yığılmalarının meydana geldiği bindirme bölgesi uçlarında, soyulma gerilmeleri 50 MPa ve kayma gerilmeleri 40 MPa civarı ortaya çıkmaktadır. Aynı şekilde, farklı yapışkan ve desen konfigürasyonları için gerilme yığılmalarının gözlemlendiği bindirme bölgesi uçlarında, soyulma gerilmeleri 30 MPa ve kayma gerilmeleri 20 MPa civarı ortaya çıkmaktadır. Ayrıca, özellikle kayma gerilmesinde bindirme bölgesinin iç kısımlarında temel alınan modele göre daha yüksek gerilme değerleri ortaya çıktığı gözlemlenmiştir. Temel alınan modelde bindirme bölgesinin iç kısımlarında yaklaşık olarak 5 MPa civarı kayma gerilmesi gözlenmesine karşın, farklı yapışkan ve desen konfigürasyonlarında 25 MPa civarı gerilmeler ortaya çıkmaktadır. Böylece, yapışkan üzerinde daha uniform bir gerilme dağılımı elde edilmiştir. Tüm konfigürasyonların soyulma ve kayma gerilmeleri birbirleri ve baz alınan temel yapışkan model ile karşılaştırılmıştır.

Gerilme değerlerinin düşürülmesi adına önerilen bir başka yöntem konik yapıya sahip yapıştırılan plaka modeli ile bindirme bölgesinden taşırılmış yapışkan modelleridir. Her iki modelde tek tesirli bindirme bağlantılı olarak temel alınan model ile aynı sınır şartları, tüm modeller için aynı yapışkan ve yapıştırılan plakalar kullanılmıştır. Burada dikkat edilmesi gereken nokta, konik yapıya sahip yapıştırılan plaka, bindirme bölgesi ucunda 30° lik açı verilerek oluşturulmuştur. Yapının mukavemeti için, en uygun olan açının belirlenmesi adına literatürde birçok çalışma yapılmıştır. Diğer model olan taşırılmış yapışkan modelinde ise, bindirme bölgesi her iki ucunda yapışkan 60° lik açı ile yapıştırılan plakaların alınlarına tutturularak, yapıştırma yüzeyi alanı artırılmıştır. Yapının mukavemeti için, en uygun olan açının belirlenmesi adına literatürde birçok

alıřma yapılmıřtır. Bu bakımdan, her iki model iin de aıların gerilmeler zerinde nemli bir faktr olduėu sylenebilir.

Daha kapsamlı incelemek adına tm modeller iin, yapıřkanın merkezinden geen bir izgi boyunca hem soyulma hem de kayma gerilmelerinin deėiřimi, grafik olarak izdirilmiřtir. Bunun yanında sadece bir doėru zerinde gerilmeleri incelemenin yeterli olmayacaėı dřnlerek, MATLAB programı aracılıėı ile analiz sonuları iřlenerek yapıřkanın yzeyindeki gerilme deėiřimi  boyutlu olarak gsterilmiřtir. Bu izdirilen grafiklere gre, temel alınan tek tesirli baėlantılı modelde gerilme yıėılmalarının meydana geldiėi bindirme blgesi ularında, soyulma gerilmeleri 45 MPa ve kayma gerilmeleri 36 MPa civarı ortaya ıkmaktadır.

Konik yapıya sahip yapıřtırılan plaka modelinde yine gerilme yıėılmaları aynı řekilde bindirme blgesi ularında olmakla beraber, soyulma gerilmesi 38 MPa ve kayma gerilmesi 33 MPa civarı gzlenmektedir. Temel alınan modele gre ok az miktarda gerilme yıėılmalarında dřř gzlenmiřtir. Burada, yapıřtırılan plakanın koniklik aısının daha fazla olacak řekilde seilmesi, gerilme yıėılmasında daha ok dřře sebep olacaėı literatr arařtırmaları sonucunda varılmıřtır.

Tařırılmıř yapıřkan modelinde yine gerilme yıėılmaları aynı řekilde bindirme blgesi ularında olmakla beraber, soyulma gerilmesi 14 MPa ve kayma gerilmesi 23 MPa civarı gzlenmektedir. Temel alınan ve konik yzeeye sahip yapıřtırılan plaka modellerine gre soyulma gerilmesi iin %60, kayma gerilmesi iin ise %35 civarlarında gerilme yıėılması dřřleri olmaktadır. Burada deėinilmesi gereken nemli bir diėer nokta, her  model iin bindirme blgesi merkezi civarında birebir aynı olacak řekilde sıfıra yakın gerilme deėerleri gzlemlenmiřtir.

Son olarak, fiber aıları deėiřiminin gerilme daėılımı zerindeki etkisi incelenmiř ve tm modeller birbirleri temel alınan tek tesirli bindirme model ile beraber karřılařtırılmıřtır. Yapıřkan zerindeki gerilme daėılımı ve gerilme analizi sonularının, fiber aısı deėiřiminden ile nemli derecede etkilendiėi gzlemlenmektedir.

Tm konfigrasyonlar iin kompozit yapısındaki katmanlar ortotropik elastik malzeme yerine literatr arařtırmaları sonucu yapılan dzlem gerinim yaklařımı erevesinde, tek ynl tabakalar iin uygulanan enine izotropik malzeme modeli kullanılmıřtır.



1. INTRODUCTION

In the field of material technology, the science of composite material has become one of the most crucial and remarkable discipline in the recent years as a consequence of extraordinary development and theoretical research efforts. Adhesively bonded composite structures are frequently used in industry applications due to easy integrated manufacture, light assembly weight, low stress concentrations, smoothness of the load transfer and high fatigue and damage performances. However, due to discontinuity of the joint region, it is the potential weakness under load conditions for all mechanical structures. An overloaded adhesive joint can cause failure in varied different types. Generally, four failure modes cohesive failure, adhesive failure in bondline, delamination in adherents and yielding of the adherents, may occur in adhesively bonded joints. Especially, a cohesive failure is the most common type of adhesive bond failure in the adhesive [1].

The damage processes and mechanical relationships of the adhesively bonded joints are very complicated. Therefore, a solid understanding of the failure processes is extremely significant for the adhesively bonded joints. Generally in many cases a combination of the given failure modes may take place in a failure process of an adhesively bonded joints. In order to perform an accurate finite element (FE) analysis to exhibit the stress distributions in the composite adhesive and adherents is essential for further estimation of strength and failure. A number of studies have been carried out in order to get an accurate and convenient stress distributions of the composite plates and the adhesive via finite elements method (FEM) and analytical methods [2-5]. Moreover, these studies have been progressing and obtaining more satisfying stress and strain results for adhesively bonded joints. In particular, with the development of finite element techniques and computers, the FE method has become the most significant approach of stress analysis due to its revolutionary simulation capability for composite structures with arbitrary geometry and stacking sequences under complex loading conditions [6].

In order to understand the failure behavior of adhesive joints several tests have been carried out. The most common tests that exhibit ordinary joint designs used in structural engineering are the single lap joint (SLJ), the double lap joint (DLJ), the tensile butt joint (TBJ) and the tubular butt joint with torsional loading. Due to simplicity of the geometry, the single lap joint is commonly used in the industrial applications. In addition, stress analysis of the single lap joints has still been performed and developed for precise prediction of the failure loads by engineers and researchers. Due to revealing robust and efficient design, stress analysis should be done during the design stage for safety and strength of the joints. If the adhesively bonded joints are designed and calculated carefully in the working conditions, they could be more reliable than the conventional techniques such as, bolted joints, rivets, welding and soldering.

Table 1.1 : Comparisons of the basic types of joining mechanisms.

Evaluation Criteria	Bonded	Bolted	Welded
Surface preparation required	Extensive	Little	Little
Joining of dissimilar materials	Good	Limited	Poor
Added weight to structure	Low	High	Moderate
Fatigue Resistance	High	Poor	Moderate
Corrosion Resistance	High	Poor	Moderate
Inspection by NDT	Very Limited	Adequate	Adequate
Ease of Disassembly	Difficult	Easy	Difficult
Production Costs	Medium	Low	Low
Resistance to Environment effects	Poor	Poor	Moderate
Production Time	Slow	Fast	Moderate

In bonded joints, analysis presents assessing the local stresses and strains under given load conditions in the adhesive and adherends. In order to evaluate the joint strength, a proper failure criterion must be applied. Besides external loading, the joint is exposed to environmental effects such as heat, moisture, thermal cycling and mechanical fatigue loading, which can affect the robustness of the joint. Therefore, a joint life estimation should be made too. In brief, the design of bonded joints is a challenging process in which both the stress analysis, a convenient failure criterion, environmental effects and manufacturing process are taken into account the suitable joint configuration and materials.

1.1 Purpose of Thesis

The aim of this thesis is to investigate the stress distribution of adhesively single lap composite joints and double lap composite joints. For both joints types, shear stress and peel stress on the adhesives are investigated and stress distributions through the overlap region are examined in details. The stress analysis of both joint types are performed via ABAQUS software program. The 2D and 3D geometry models are generated and are used in this research. In ABAQUS, there are two types of 2D models, which are conventional shell model and continuum shell model. Continuum shell model is a full 3D geometry, and element thickness is defined by nodal geometry. Both conventional and continuum shell models are used in this research. Due to simple geometric design and efficient modelling process, adhesively single lap joint is examined in detail. In order to decrease stress concentration in the critical region of the single lap joint, mixed adhesives, different adhesive shape pattern configurations, tapered, and spew fillet models are investigated. Especially single lap joints unbalanced characteristics, stress concentration levels are severe according to other adhesively bonded joint types.

The other aim of this research is to reveal how effect fiber orientations changing on the stress distribution of the adhesive. In order to obtain too much stress results of the models, scripts have been written by means of Python programming language and MATLAB software program. Thus, all stress analyses are generated automatically and stress results are obtained and plotted automatically.

1.2 Literature Review

A great numbers of researches have been carried out to the study of adhesively bonded joints. Due to material non-linearity and geometric complexity of the adhesive, an accurate and exact solution for the stress of the joints is difficult to obtain. Therefore, in order to find a satisfying solution, wide range of developments have been still performed, and each with different simplifications, limitations and assumptions.

V. O. Volkersen [7] led the investigation into analyzing the stress distribution in an adhesive joint with shear lag analysis. Under differential shear loading, his work was to present non-uniform deformation of the adhesive layer. Moreover, it used to specify shear strength and shear modulus, respectively. Goland and Reissner [8] based on Volkersen's solution and took into account influence of the eccentric loading angle and hence presenting the significant influence of the edge moment by including the bending moment factor, k . This factor is associated with the bending moment M_0 , the load applied to the adherends T , and the adherend thickness t by equation (1.1).

$$k = \frac{2M_0}{Tt} \quad (1.1)$$

This bending moment factor is applicable for small loadings for which no rotational movement observed. The maximum stresses observed at the ends of the overlap region, which was to be expected. Though, due to negligence of the adherend shear strains, Goland and Reissner's solution is not proper to most joint especially different adherends materials according to Adams [9]. Therefore, he performed a geometrically non-linear finite element analysis to obtain more accurate and effective solution for wider range of lap joints. L. J. Hart-Smith [10] took Goland and Reissner's analysis a step forward by defining non-linear material property for the adhesive. Tsai and Morton [11] examined in detail and compared of the available analytical solutions, and presented Hart-Smith's model the most reliable and exact for the edge moment determining in short joints. In long joints, Oplinger's [12] proposed the most accurate model for the edge moment determining. Carpenter [13] discovered the influence of several mathematical assumptions on the adhesive stresses in single-lap joints. The maximum adhesive shear and peel stresses were predicted and especially neglecting the shear deformation of the adherends affects peel stress considerably.

Delale and Erdogan [14] were generated a single lap joint analysis by assuming the linearly viscoelastic adhesive and the linear elastic adherends. Adams and Peppiatt [15] with plane strain assumption, performed an adhesively bonded joint analysis using a two dimensional linear elastic finite element method. Silva and Adams [16] especially at low temperature, examined how to decrease the transverse stresses in the composite and thus to obtain more reliable, robust and strength joints. They used different adhesives in bonding region to react differently at the temperature. Ozel et al. [17] under tensile loading, examined mechanical properties of adhesively bonded joints with different material and geometry configuration of lower and upper adherends.

Kadioglu and Adams [18] investigated the potential use of ductile adhesives industrial applications and their advantages over conventional structural adhesives are examined. The performance of high strain capacity ductile adhesives to failure was investigated in the single lap joint configuration under tensile loading. Gultekin et al. [19] investigated numerically and experimentally strength of the different single lap joint configurations with different adherend width under the tensile loading. Pinto et al. [20] studied experimentally and numerically the effect of adherend recessing at the overlap edges under the tensile loading of single lap joint, bonded with a brittle adhesive. In order to maximize the joint's strength, the recess dimensions (length and depth) were optimized for different values of overlap length. Thus, reduction of the peak stresses at the overlap edges were observed. Akpınar [21] investigated the mechanical behavior of an adhesively bonded joint with spew fillet under bending moment, where lower and upper adherends' widths are not same, were examined experimentally and numerically. It is observed that the spew fillet with different widths had a remarkable effect on the stress distribution of the single lap joint. Furthermore, the spew fillet is caused the increasing the load carrying capacity and the decreasing the stress concentration of the joint.



2. COMPOSITE MATERIALS

Composite material is a structural material that consist of two or more constituents are combined at a macroscopic level and are not soluble in each other. One component is called the reinforcing phase and the one in which it is embedded is called the matrix. The reinforcing phase material may be in form of fibers, flakes, or particles. On the other hand, the matrix phase materials are generally continuous [22]. The matrix material acts as the bulk material and transfers load between reinforcement materials. Furthermore, the matrix has also another function, which is to preserve the reinforcement from the environment factors. The reinforcement ensures the stiffness and strength properties of the composite. Figure 2.1 displays the roles of the matrix and reinforcement in composites [23].

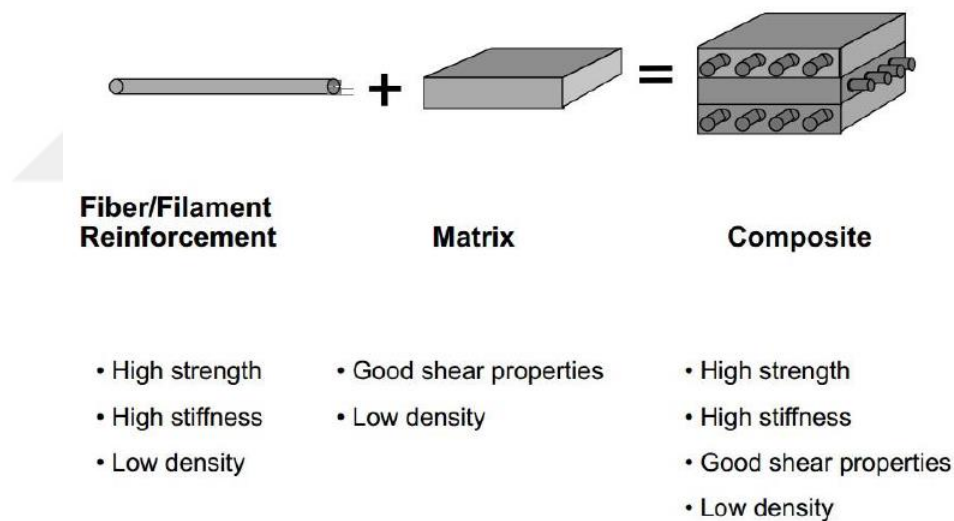


Figure 2.1 : Composition of composites.

Composite materials have been center of the interest in the variety of industries recently, especially aerospace and automotive industries. The fuel consumption is the major operating cost of an airliner. The fuel cost depends on mass of the aircraft, efficient of its engines, aerodynamic drag surface of the aircrafts, and flight trajectory of the aircrafts. The fuel cost can be decreased by using high efficient engines, lower mass of aircraft, reduction of the aerodynamic drag, optimization the flight trajectory. However, the reduction of the aircraft's mass is the direct factor of the fuel

consumption. In Figure 2.2, it can be realised that the portion of composite material in the total structural weight of Airbus aircrafts is increasing constantly [24].

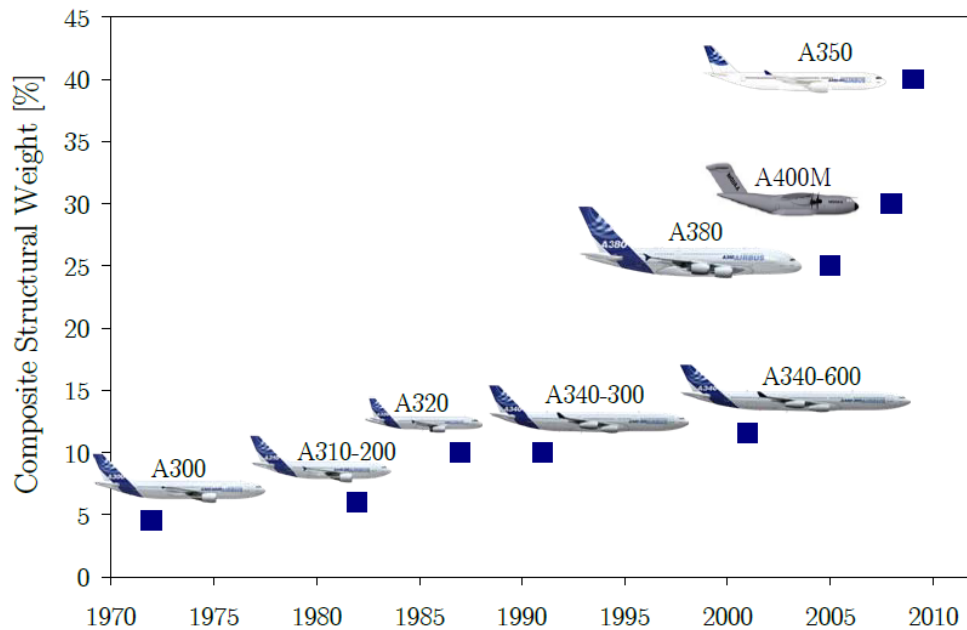


Figure 2.2 : Portion of composite materials in Airbus aircrafts.

In order to lighten aircrafts, some parts such as, wings, fairings, radome, spoilers have been developed during 1960s. New generation aircrafts are produced with all composite wing structures and fuselage. By applying composites on the aircraft, decreases maintenance cost, fatigue related inspections and corrosion related maintenance checks, according to conventional aluminum jetliners.

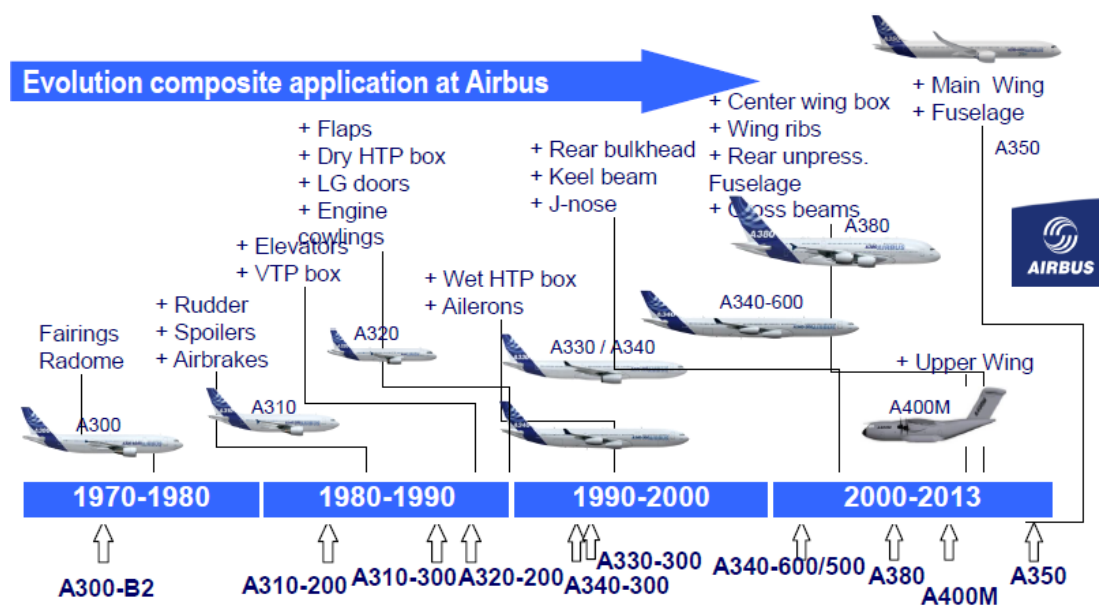


Figure 2.3 : Evolution of the use of composites in Airbus aircraft [25].

Composite materials are generally divided into two different levels according to constituent.

The first stage of the classification is related to matrix constituent which is shown in Figure 2.4. The primary composite classes are Organic Matrix Composites (OMCs), Metal Matrix Composites (MMCs) and Ceramic Matrix Composites (CMCs). The organic matrix composite contains two groups, polymer matrix composites and carbon matrix composites. Polymer matrix composites can be further divided into two main groups, thermoset and thermoplastic.

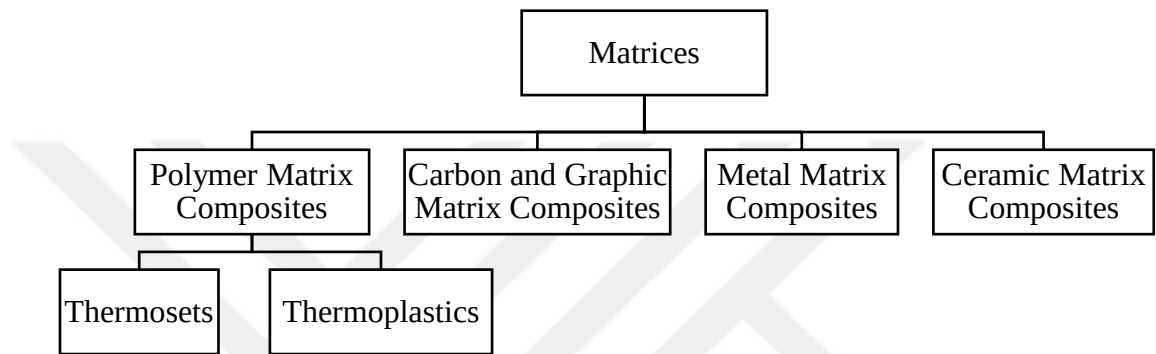


Figure 2.4 : Classification of the matrix materials.

The second stage classification is related to the reinforcement constituent, which is shown in Figure 2.5. These are fiber reinforced composites, which is divided into continuous and discontinuous fibers, laminar composites and particulate composites. Fiber reinforced composites (FRCs) consist of fibers embedded in the matrix material, resin.

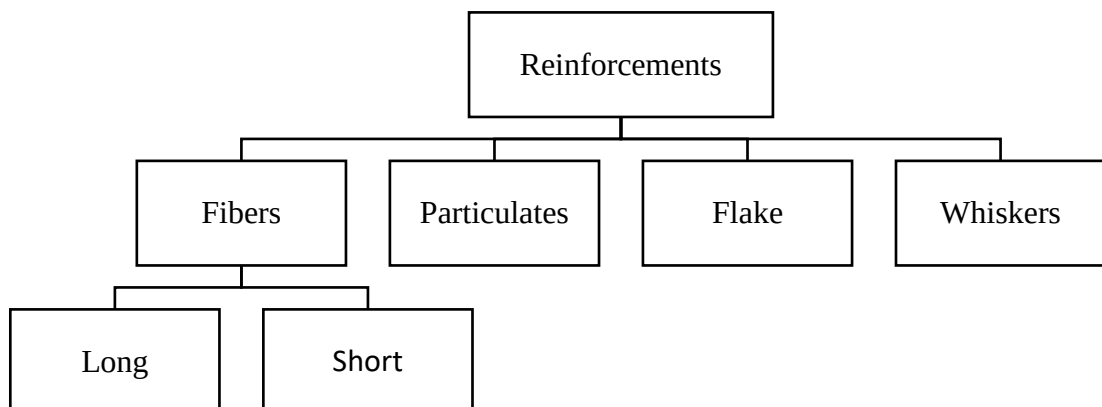


Figure 2.5 : Reinforcements

The properties of the composite varies with length of fiber, it is considered to be a discontinuous or short fiber composite. Mechanical properties of discontinuous fibers composites do not change along the direction of the force applied. Conversely, continuous fiber composites are all oriented to one direction. Thus, great strength can be obtained along this direction. Fibers are small in diameter and when tensile load is applied axially, they bend easily even though they have very good tensile properties.

A structural composite is generally consisted of both composite and homogeneous materials. The properties of a structural composite depend not only on the properties of the constituent materials but also on the geometrical design of the structural elements. The most common structural composites are laminar composites and sandwich panels. Laminar composites are combined two or more layers of materials held together by matrix. Laminated composites are generated by stacking the certain fiber direction plies. Orientation of the lamina with a lay-up of $[0^\circ/45^\circ/-45^\circ/90^\circ]_s$ is shown in Figure 2.6.

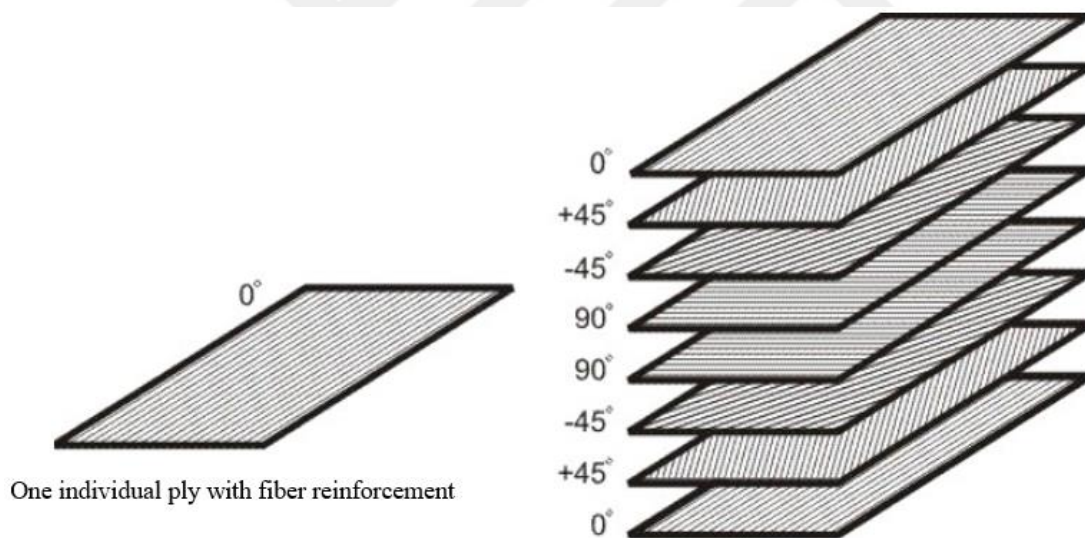


Figure 2.6 : Orientation of the lamina with a lay-up of $[0^\circ/45^\circ/-45^\circ/90^\circ]_s$.

A Sandwich structure composite is a special class of composite materials that is fabricated by attaching two thin but stiff facesheets to a lightweight but thick core. The core material is normally low strength material, but its higher thickness provides the sandwich composite with high bending stiffness with overall low density. Open- and closed-cell-structured foams like polyvinylchloride, polyurethane, polyethylene or polystyrene foams, balsa wood, syntactic foams, metal foams and honeycombs are commonly used as core materials. The selection of the optimal sandwich structure is

usually based on the different conditions associated with load-bearing capacity, production, and component design. Lightweight and stiff sandwich panels are a crucial element of many modern aircraft and spacecraft designs. Honeycomb sandwich structure is shown in Figure 2.7.

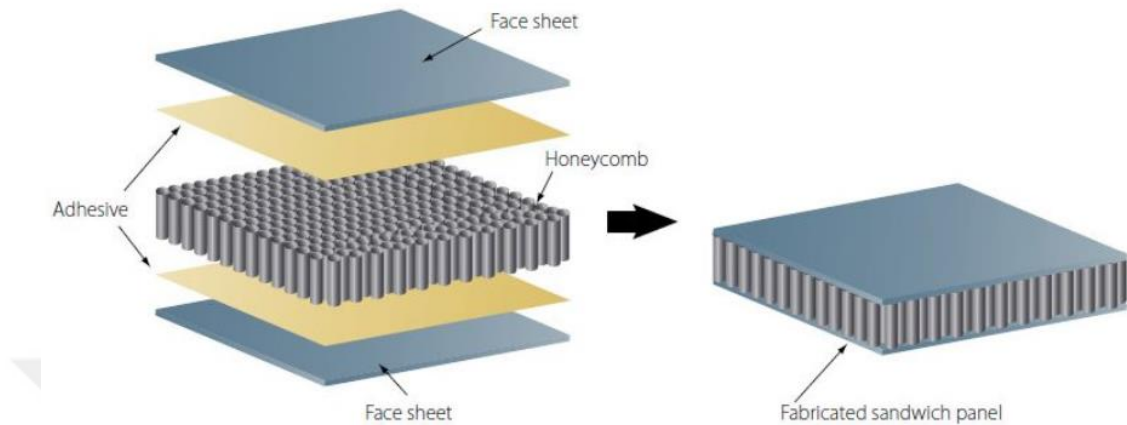


Figure 2.7 : Honeycomb sandwich structures.

Particulate composites are combined of distributed particles or embedded in a matrix. The particles can be flakes or in powder form. Due to adding randomly, they are generally defined isotropic properties. Particulate composites have benefits such as enhanced strength, enduring high temperature operating, and oxidation resistance.

2.1 Matrices

The strength of the composites is mainly provided by the reinforcement, but the significance of the matrix material cannot be neglected. It supports the reinforcement material, aids to fibers in carrying the load and prevents the environment effects such as rain, moist.

2.1.1 Organic matrix composites

2.1.1.1 Polymer matrix composites

Polymer matrix composites (PMCs) are the most broad enhanced composites, composing of a polymer such as epoxy, urethane, polyester reinforced by thin diameter fibers such as graphite, boron, aramids. Due to low cost, simple manufacturing principles, and high strength, they are the most widespread composites. The main disadvantages of PMCs are low elastic properties in particular direction, low operating temperature, and high coefficients of moisture and thermal expansions.

Two principal types of polymers are thermosets and thermoplastics. The most common fiber composite matrix is the thermoset especially in the aerospace industry. After curing, thermosets polymers are infusible and insoluble due to having strong covalent bonds. On the contrary, thermoplastics are formable at high temperatures and pressure as the bonds are weak and of the van der Waals type. Typical examples of thermoset include epoxies, polyesters, phenolics, and polyamide; typical example of thermoplastics include polyethylene, polystyrene, polyether-ether-ketone (PEEK), and polyphenylene sulfide (PPS). The differences between thermosets and thermoplastics are given in Table 2.1 [22]. Each polymer has its advantages and drawbacks in its use. Epoxies, the advantages are good adherence to metals and glasses, high mechanical strength and the disadvantages are complexity in processing and high cost. Polyester, the advantages are the skill of being made translucent, and low cost. The drawbacks are low service temperature (77°C), high shrinkage during curing and brittleness. Phenolics, high mechanical properties and low cost are the advantages, but the drawback is high void content.

Even though epoxy is costlier than other polymer matrices, it is the most widely used PMC matrix. Especially in aerospace applications more than two-thirds of the polymer matrices used are epoxy based.

Table 2.1 : Differences between thermoplastics and thermosets.

Thermoplastics	Thermoset
Soften on heating and pressure, and so easy to repair	Decompose on heating
High strains to failure	Low strains to failure
Indefinite shelf life	Definite shelf life
Not tacky and easy to handle	Tacky
Short cure cycles	Long cure cycles
Excellent solvent resistance	Fair solvent resistance
Higher fabrication temperature	Lower fabrication temperature

2.1.1.2 Carbon matrix composites

Carbon matrix composites generally refer to as carbon-carbon composites, due to using carbon fibers in carbon matrix. These composites distinguish resistance of the high-temperature environments of up to 3315°C and are 30% lighter and 20 times stronger than graphite fibers. Likewise ceramics, carbon is brittle and cracked sensitive. The other characteristics of the reinforcement of a carbon matrix are low density, good compressive and tensile strength, high thermal conductivity, high coefficient of friction, and high fatigue resistance. On the other hand, low shear strength, sensitiveness of oxidation at high temperatures, and high cost are the drawbacks.

2.1.2 Metal matrix composites

Metal matrix composites (MMCs), have a metal matrix such as aluminum, titanium and magnesium and potential fibers are silicon carbide and carbon. Metal matrices offer high fracture toughness, high stiffness, high strength, and insensitivity of humid and better fatigue, wear, and flaw resistance than those offered by their polymer matrix composites. Metals are used as decrease or increase their properties to encounter the design criteria. For instance, by the addition of the fibers such as silicon carbide, electric and thermal conductivity and large coefficients of thermal expansion can be decreased, the strength and elastic modulus of the metals can be raised.

2.1.3 Ceramic matrix composites

Ceramic matrix composites (CMCs) have a ceramic matrix and example of matrices in such composites contain aluminosilicate. Typical fibers include carbon or silicon carbide. High service temperature limits for ceramics, low density, chemical inertness, high strength, hardness are the advantages of CMCs. However, due to low fracture toughness, they can be failed easily under impact or tensile loading. In order to increase fracture toughness, silicon carbide or carbon is used as fibers reinforcing the ceramics. Therefore CMCs are more attractive for applications in which extreme service temperatures and high mechanical properties are required.

2.2 Reinforcements

The role of the reinforcement is basically one of enhancing the mechanical properties of the resin system in a composite. They also provides a number of extra purposes of

conduction, resistance to corrosion, heat resistance and rigidity. All of the different fibers used in composites have different properties and therefore influence the properties of the composite in different ways. A reinforcement is embedded the matrix must be stiffer and stronger than the matrix and ability of enhancing failure mechanism of the composite material.

2.2.1 Fiber reinforced composites

Fibers are used to enhance composite material properties. The direction of the fiber in the matrix is significant parameter of the composite strength and the strength is greatest along the longitudinal direction of fiber. There are a great numbers of fiber types such as glass fibers, aramid fibers, boron fibers, silicon carbide fibers, metal fibers. Fiber composites consist of matrices reinforced by long (continuous) or short (discontinuous) fibers. Carbon and aramids are an examples of the fibers which are usually anisotropic. Resins such as epoxy, metal such as aluminium, and ceramics such as calcium-alumino silicate are some examples of the matrices. Types of composites based on reinforcement shape are shown in Figure 2.6 [22].

2.2.2 Particulate reinforced composites

Particulate composites consist of particles embedded in matrices such as ceramics and alloys. They are generally isotropic due to adding the particles randomly. The advantages of the particulate composites are high strength, high operating temperature, and oxidation resistance. Aluminum particles in rubber, gravel, cement and sand to make concrete, silicon carbide particles in aluminum are typical examples of the particulate composites. Types of composites based on reinforcement shape are shown in Figure 2.6 [22].

2.2.3 Flake reinforced composites

Flake composites consist of flat reinforcements of matrices and glass, mica, silver, and aluminum are typical flake materials. The advantages of the flake composites are low cost, high strength and high out-of-plane flexural modulus. However, only a limited number of materials are available for use and flakes cannot be easily oriented. Types of composites based on reinforcement shape are shown in Figure 2.6 [22].

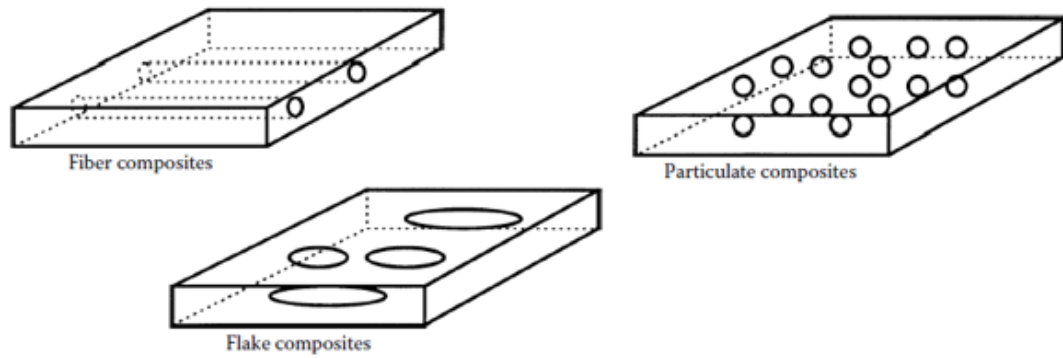


Figure 2.8 : Types of composites based on reinforcement shape.

2.2.4 Whiskers reinforced composites

Single crystals formed with zero defects are called whiskers. They are generally discontinuous and short fibers of different cross sections. They are made from various materials such as graphite, copper, silicon carbide, and iron. The differences from the particles of the whiskers is to have a definite length to width ratio greater than one. Whiskers strength changes inversely with effective diameter.

2.3 Micro-mechanics and Macro-mechanics Structures of the Composite

Composite materials can be analyzed and observed at different scales and levels, related to respective characteristics and behavior under consideration. Particle size, fiber diameter and spacing between fibers and matrix are observed at constituent level of the scale of observation. At this microscopic level, interactions of the constituents are defined and observed by micromechanics. Micromechanics investigates the constituents stress and deformations, local failures, such as matrix failure (tensile, shear and compressive), fiber failure (tensile, splitting, buckling). Micromechanics plays a key role in the study of mechanical properties such as strength and failure mechanisms, fatigue life, and fracture toughness. Contrary to micromechanics, lamina level is usually quicker and use average properties in the analysis, which is referred macromechanics and investigates the unidirectional lamina with its own average strength and stiffness properties. Macromechanical analysis is performed via lamination theory at the laminate level. As stacking sequence and a function of lamina properties, it examines overall behavior. Lastly, associated with lamination theory and finite element methods can calculate the overall behavior of the structure and the each

lamina stresses at the structure level. All levels and related types of analysis are shown in the Figure 2.9 [29].

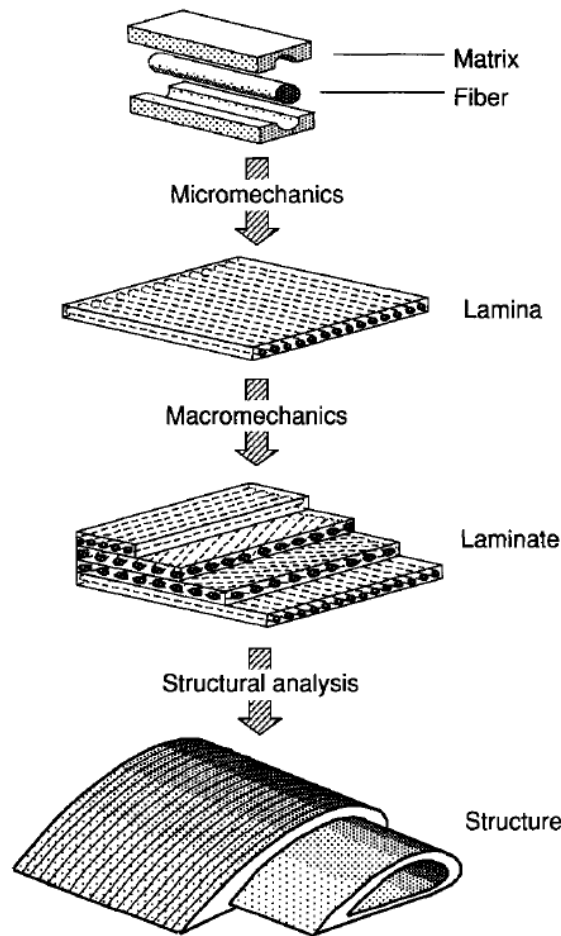


Figure 2.9 : Types of analysis and investigation levels for composite.

2.3.1 Micro-mechanical analysis of a lamina

Micromechanics investigates unknown properties of the heterogeneous composite material based on known properties of the fiber and matrix. Composite material behavior can be predicted by using constituents of the composite materials. Micromechanics of lamina can give a key information the designer to use proper constituents of a composite material in a laminated structure. In order to calculate properties of composite materials, several methods have been suggested. These approaches are mechanics of materials, theory of elasticity, and empirical methods. Theory of elasticity consists of three subgroups: self-consistent model, variation principles and exact solutions. In order to perform continuous fiber reinforced composite micromechanical analysis, some assumption should be done. Assumptions

for lamina are linearly elastic, macroscopically orthotropic and homogeneous, and non-prestress. Assumptions for fibers are isotropic, linearly elastic, homogeneous, regularly spaced, excellently bonded and aligned. Assumption for matrix are linearly elastic, homogeneous, isotropic and void-free. A symbolic volume element for a lamina with unidirectional fibers is given Figure 2.10[30].

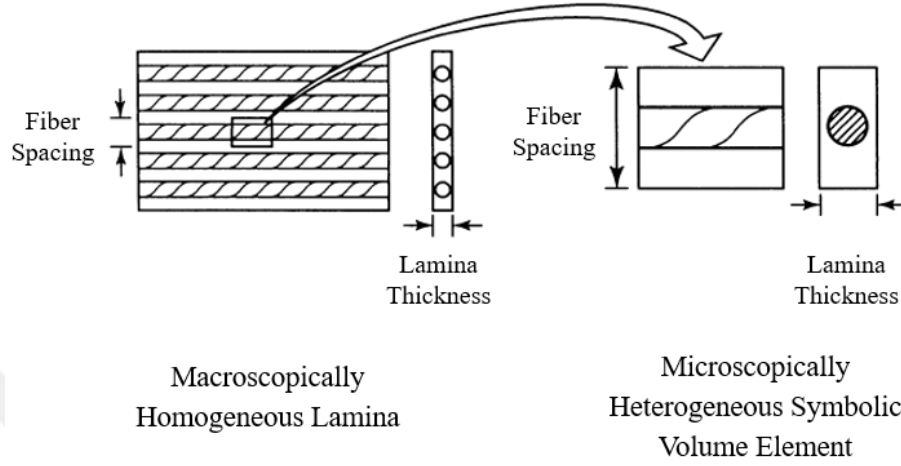


Figure 2.10 : A lamina with unidirectional fibers symbolic volume element.

In this study, mechanics of materials (strength of materials) approach is investigated. This approach simplifies assumptions of either uniform stress or strain in the constituents. Orthotropic moduli of a unidirectional fiber-reinforced composite material, that is E_1 , E_2 , ν_{12} , and G_{12} , can be derived via mechanics of materials approach.

2.3.1.1 Determination of E_1

When the load is applied fiber direction as σ_1 (Figure 2.11), according to the basic assumption of the mechanics of the material, both matrix and fiber are elongated ϵ_1 .

$$\epsilon_1 = \frac{\Delta_L}{L} \quad (2.1)$$

Due to both constituent material elastic behavior, the stresses in the direction are given in equation (2.2) as σ_f and σ_m .

$$\sigma_f = E_f \epsilon_1 \quad , \quad \sigma_m = E_m \epsilon_1 \quad (2.2)$$

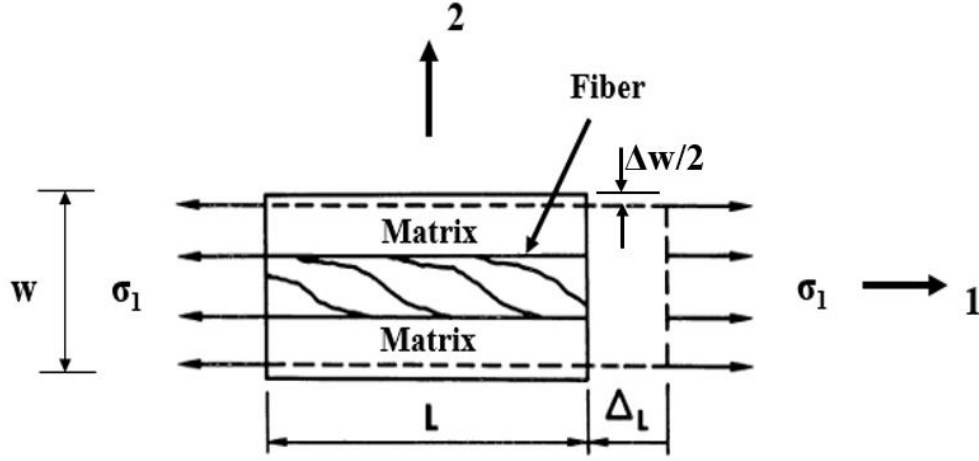


Figure 2.11 : State of both fiber and load in the 1-direction.

The average stress σ_1 acts on the symbolic volume element cross-sectional area A , which is given equation (2.3). Likewise, σ_f and σ_m act on the cross-sectional area of the fibers and matrix A_f , and A_m , respectively. Therefore the resultant force on the symbolic volume element is given equation (2.4).

$$\sigma_1 = \sigma_f + \sigma_m \quad , \quad A = A_f + A_m \quad (2.3)$$

$$P = \sigma_1 A = \sigma_f A_f = \sigma_m A_m \quad (2.4)$$

By substitution of equation (2.2) in equation (2.4) and from the general Hooke's Law stress-strain relation,

$$\sigma_1 = E_1 \varepsilon_1 \quad (2.5)$$

and hence;

$$E_1 = E_f \frac{A_f}{A} + E_m \frac{A_m}{A} \quad (2.6)$$

By writing the volume fraction of matrix and fiber as given in equation (2.7) and simplifying equation (2.6)

$$V_m = \frac{A_m}{A} \quad , \quad V_f = \frac{A_f}{A} \quad (2.7)$$

Thus,

$$E_1 = E_f V_f + E_m V_m \quad (2.8)$$

Equation (2.8) is defined as rule of mixtures for apparent elastic modulus of the composite material in direction of the fibers and depending on the volume fraction of the composite material, linearly changing of E_1 representation is given in Figure 2.12.

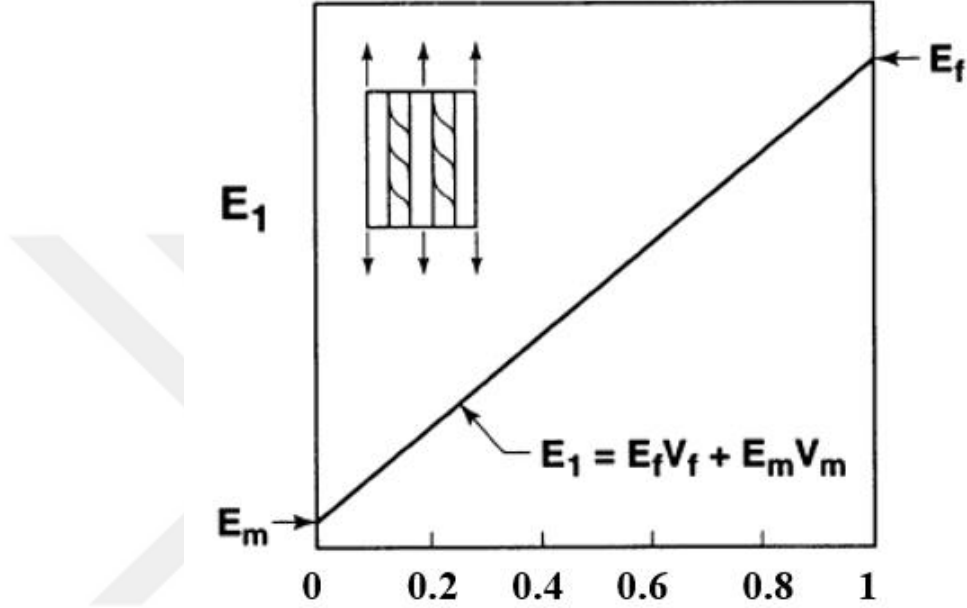


Figure 2.12 : Variation of E_1 depending on the fiber volume fraction.

2.3.1.2 Determination of E_2

Transvers to the fibers direction in the composite material is considered and determined E_2 elastic modulus. Likewise determination of E_1 , the same transverse stress, σ_2 , is assumed to be applied matrix and fibers which is shown in Figure 2.13. Due to equal stress subjected for matrix and fibers, the strains can be found from the stresses equation (2.9).

$$\varepsilon_f = \frac{\sigma_2}{E_f} \quad , \quad \varepsilon_m = \frac{\sigma_2}{E_m} \quad (2.9)$$

The transverse dimension over which, ε_f behaves is $V_f W$, while ε_m behaves $V_m W$ approximately. Therefore, the total transverse deformation can be defined as equation (2.10).

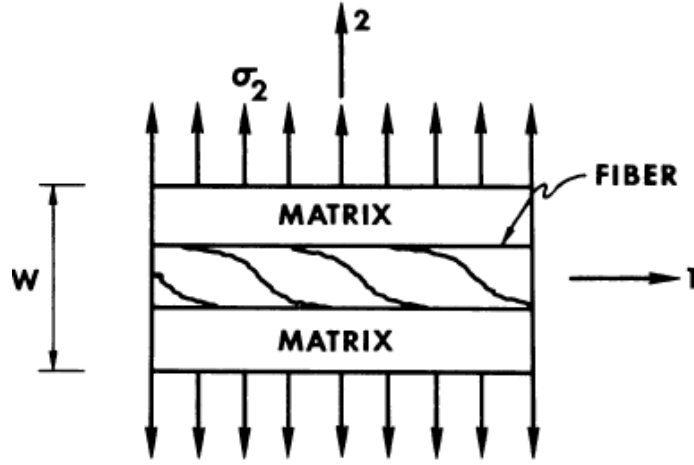


Figure 2.13 : Symbolic volume element loaded in the 2-direction.

$$\Delta W = \varepsilon_2 W = \varepsilon_f V_f W + \varepsilon_m V_m W \quad \text{if simplify} \quad \varepsilon_2 = \varepsilon_f V_f + \varepsilon_m V_m \quad (2.10)$$

Equation (2.9) strains are replaced in equation (2.11),

$$\varepsilon_2 = \frac{\sigma_2}{E_f} V_f + \frac{\sigma_2}{E_m} V_m \quad (2.11)$$

From the general Hooke's Law stress-strain relation,

$$\sigma_2 = E_2 \varepsilon_2 = E_2 \left[\frac{V_f \sigma_2}{E_f} + \frac{V_m \sigma_2}{E_m} \right] \quad (2.12)$$

Therefore, E_2 is

$$E_2 = \frac{E_f E_m}{V_m E_f + V_f E_m} \quad (2.13)$$

2.3.1.3 Determination of ν_{12}

It is also called major Poisson's ratio, ν_{12} is calculated by an approach similar to E_1 .

The major Poisson's ratio is

$$\nu_{12} = -\frac{\varepsilon_2}{\varepsilon_1} \quad (2.14)$$

All the stresses are zero except $\sigma_1 = \sigma$ and then, the deformations are shown in Figure 2.14. The fiber strains and matrix strains are identical that is the basic assumption, as in the approach to E_1 . The transverse deformation Δ_w is macroscopically

$$\Delta_w = -W\varepsilon_2 = Wv_{12}\varepsilon_1 \quad (2.15)$$

Δ_w is also microscopically defined as

$$\Delta_w = \Delta_{mW} + \Delta_{fW} \quad (2.16)$$

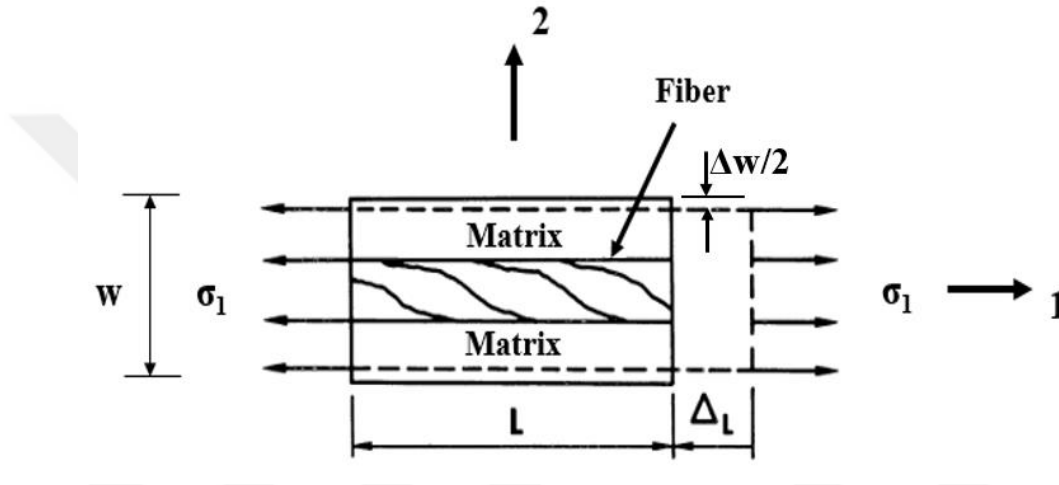


Figure 2.14 : Symbolic volume element loaded in the 1-direction.

The transverse deformations Δ_{mW} and Δ_{fW} are defined in equation (2.17) approximately as in the manner of the analysis for the transverse elastic modulus, E_2 .

$$\Delta_{mW} = WV_m v_m \varepsilon_1 \quad , \quad \Delta_{fW} = WV_f v_f \varepsilon_1 \quad (2.17)$$

Combine equations (2.15) – (2.17) and divide by $\varepsilon_1 W$ to obtain v_{12} .

$$v_{12} = v_m V_m + v_f V_f \quad (2.18)$$

2.3.1.4 Determination of G_{12}

The in-plane shear modulus, G_{12} , is determined in the mechanics of materials approach by assuming that the shearing stresses on the fiber and on the matrix are the same. It can be clearly realized that shear deformations cannot be the same (Figure 2.19).

According to the fundamental assumption,

$$\gamma_m = \frac{\tau}{G_m} \quad , \quad \gamma_f = \frac{\tau}{G_f} \quad (2.19)$$

Although fiber reinforced composite materials exhibit nonlinear shear stress- shear strain behavior, it is ignored and the behavior is considered as linear due to simplification.

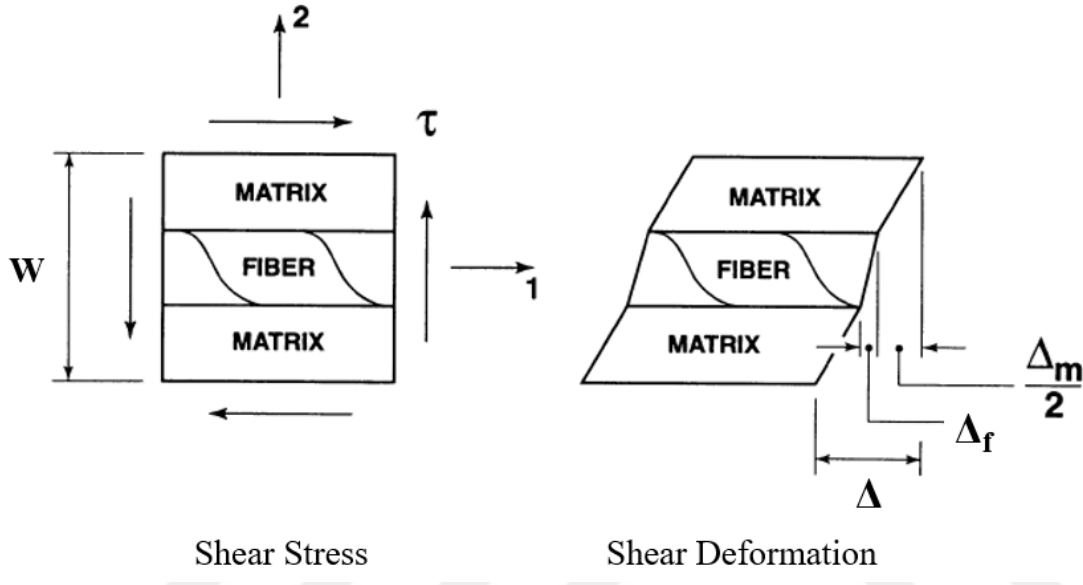


Figure 2.15 : Symbolic volume element loaded in shear.

It can be realised that due to lower shear modulus of matrix, matrix deforms more than fiber in shear and the total shearing deformation is

$$\Delta = \gamma W \quad (2.20)$$

According to microscopic deformations,

$$\Delta_m = V_m W \gamma_m \quad , \quad \Delta_f = V_f W \gamma_f \quad (2.21)$$

Then, because $\Delta = \Delta_m + \Delta_f$, and divided by W,

$$\gamma = V_m \gamma_m + V_f \gamma_f \quad (2.22)$$

By substitution of equation (2.19) and realization that macroscopically

$$\gamma = \frac{\tau}{G_{12}} \quad (2.23)$$

Equation (2.22) can be rewritten as

$$\frac{\tau}{G_{12}} = V_m \frac{\tau}{G_m} + V_f \frac{\tau}{G_f} \quad (2.24)$$

Finally,

$$G_{12} = \frac{G_m G_f}{V_m G_f + V_f G_m} \quad (2.25)$$

2.3.2 Macro-mechanical analysis of a lamina

A lamina is a flat structure of unidirectional or woven fibers in a matrix. A laminate is constructed by stacking several such laminae in the lamina thickness direction. Therefore, mechanical behavior of the lamina is significant to the understanding of laminated fiber-reinforced composite structures. Macromechanical behavior, that is, the behavior when only averaged mechanical properties are taken into account, rather than the interactions of the constituents of composite material which was mentioned in micro mechanical chapter. The main limitation of both chapters is to linear elastic behavior and considering the lamina is homogeneous.

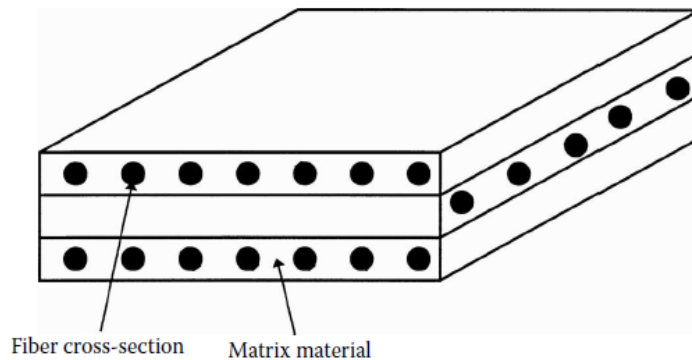


Figure 2.16 : A laminate made of three laminae [22].

Even if the homogenization of a lamina, the mechanical behavior is still different from a homogeneous isotropic material. Because the stiffness of the unidirectional lamina in the fibers direction is much larger than the stiffness in the perpendicular direction to the fibers. The materials are classified based on the behavior for a specified loading condition. These contain, anisotropic materials, monoclinic materials, orthotropic materials, transversely isotropic materials, and isotropic materials. In this study,

orthotropic materials and transversely isotropic materials are examined in details, because they are generally assigned as material properties of the composites. All materials classes can be stated below in briefly.

In anisotropic material, there are no symmetric planes of material properties, that is, it has different physical properties in different directions. There are 21 independent elastic constants in the stress-strain relationship. Normal stresses induce not only normal strains due to Poisson effect but also shear strains. Likewise, shear stresses induce not only shear strains but also normal strains.

In monoclinic material, there is a single symmetric plane of material property. There are 13 independent elastic constants in stiffness matrix.

In (specially) orthotropic material, there are three mutually orthogonal symmetric planes of material property. The intersections of these three symmetric planes are called the principal material directions. The normal stress are applied in the principal material directions. Otherwise, it is called as general orthotropic which behaves almost equivalent to anisotropic material. There are nine independent elastic constants in the stiffness matrix for specially orthotropic material.

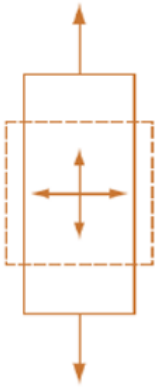
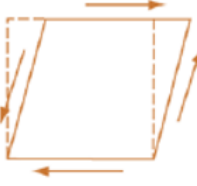
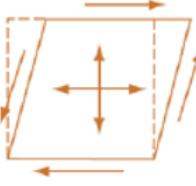
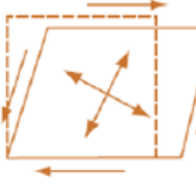
	Isotropic	Orthotropic	Anisotropic
Uniaxial Tension			
Pure Shear			

Figure 2.17 : Differences in the deformations on uniaxial tension and pure shear.

In transversely isotropic material, if a material has axes of symmetry in its longitudinal axis and all directions perpendicular to its longitudinal axis, that is, more than there mutually perpendicular axes of symmetry, then such material is transversely isotropic such as unidirectional composites.

In isotropic material, properties are the same all directions such as lateral, axial, and in between. Therefore, the material includes an infinite number of symmetric planes of material property. The material properties are independent directionally and hence, there are two independent elastic constants. Tensile normal stresses are applied in any direction on an isotropic material induce only elongation in that direction and contradiction in the two transverse directions. It will not induce any shear strain in any form in the material. Likewise, shear stresses induce only related shear strains.

Table 2.2 : Elastic constants of different materials for 2D and 3D.

Material	Three Dimensional Number of non-zero & independent constants	Two Dimensional Number of non-zero & independent constants
Anisotropic	36 & 21	9 & 6
Generally Orthotropic	36 & 9	9 & 4
Specially Orthotropic	12 & 9	5 & 4
Transversely Isotropic	12 & 5	5 & 4
Isotropic	12 & 2	5 & 2

2.3.2.1 Stress-strain relations for orthotropic laminates

The general Hooke's law stress-strain relationship is given equation (2.26)

$$\sigma_{ij} = C_{ijkl}\epsilon_{kl}, \quad i, j, k, l = 1, 2, 3 \quad (2.26)$$

Where σ_{ij} (9 components) are the stress components shown on a three-dimensional cube in x, y, and z coordinates in Figure 2.18, C_{ijkl} (81 components) is the stiffness matrix, and ϵ_{kl} (9 components) are the strain components. Due to symmetry of shear stresses and shear strains ($\sigma_{ij} = \sigma_{ji}$ and $\epsilon_{kl} = \epsilon_{lk}$), only 6 components of stress (σ_{ij}) are

independent, and likewise for strain (ϵ_{kl}). This leads to contracted notation, which is given Table 2.3.

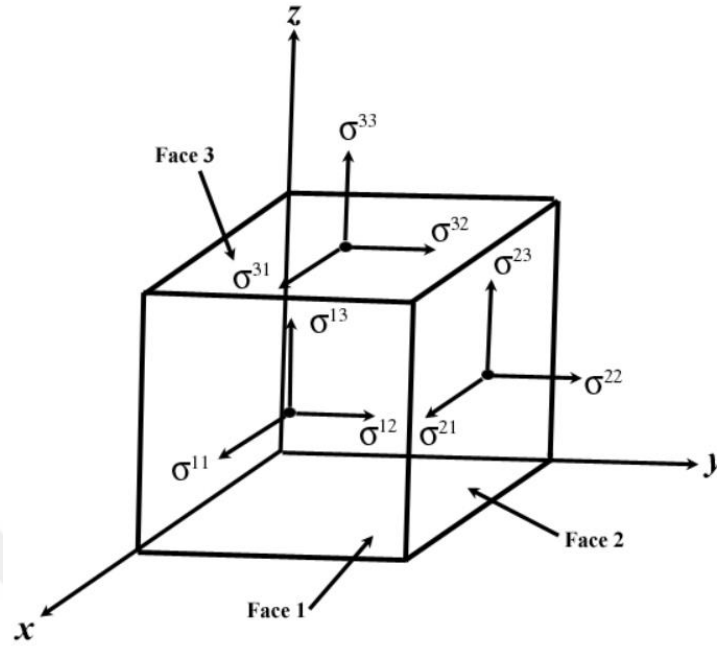


Figure 2.18 : 3D state of stress

Table 2.3 : Tensor versus contracted notation for stresses and strains

Stresses		Strains	
Tensor notation	Contracted notation	Tensor notation	Contracted notation
σ_{11} (σ_1)	σ_1	ϵ_{11} (ϵ_1)	ϵ_1
σ_{22} (σ_2)	σ_2	ϵ_{22} (ϵ_2)	ϵ_2
σ_{33} (σ_3)	σ_3	ϵ_{33} (ϵ_3)	ϵ_3
$\tau_{23} = (\sigma_{23})$	σ_4	$\gamma_{23} = 2\epsilon_{23}$	ϵ_4
$\tau_{31} = (\sigma_{31})$	σ_5	$\gamma_{23} = 2\epsilon_{23}$	ϵ_5
$\tau_{12} = (\sigma_{12})$	σ_6	$\gamma_{23} = 2\epsilon_{23}$	ϵ_6

It is realized that the engineering shear strain (γ_{ij}), is the total angle of shearing under simple shear stress condition. However, the tensor shear strain (ϵ_{ij}), is half of shearing under pure shear stress condition (Figure 2.19).

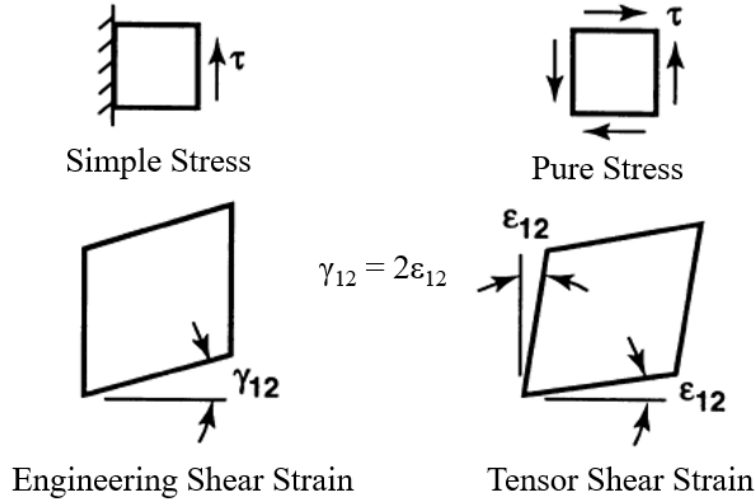


Figure 2.19 : Engineering shear strain and tensor shear strain.

Moreover, due to shear stress ($\sigma_{ij} = \sigma_{ji}$, then $C_{ijkl} = C_{jikl}$) and shear strain ($\epsilon_{kl} = \epsilon_{lk}$, then $C_{ijkl} = C_{ijlk}$) symmetry, and from thermodynamic first law considerations, $C_{ijkl} = C_{klij}$, and hence the stiffness matrix, C_{ijkl} , has 36 constants in equation (2.26). However, it can be shown that conservative materials possess a strain energy density function and hence, elastic materials for which strain energy density function exists have incremental work per unit volume of

$$dW = \sigma_i d\epsilon_i, \quad \sigma_i = C_{ij} \epsilon_j \quad \text{then} \quad dW = C_{ij} \epsilon_j d\epsilon_i \quad i, j = 1, 2, \dots, 6 \quad (2.27)$$

Upon integration for all strains, the work per unit of volume is

$$W = \frac{1}{2} C_{ij} \epsilon_i \epsilon_j \quad (2.28)$$

However, Hooke's law, can be derived from equation (2.28),

$$\frac{\partial W}{\partial \epsilon_i} = C_{ij} \epsilon_j \quad (2.29)$$

$$\frac{\partial^2 W}{\partial \epsilon_i \partial \epsilon_j} = C_{ij} \quad (2.30)$$

Similarly,

$$\frac{\partial^2 W}{\partial \epsilon_j \partial \epsilon_i} = C_{ji} \quad (2.31)$$

Since order of differentiation of W is immaterial, hence

$$C_{ij} = C_{ji} \quad (2.32)$$

Therefore, only 21 stiffness components are actually independent in the stiffness matrix. In similar way, compliance matrix is also a symmetric matrix ($S_{ij}=S_{ji}$). With the contradiction from 36 to 21 independent constants, the stress-strain relation are

$$\begin{bmatrix} \sigma_1 \\ \sigma_2 \\ \sigma_3 \\ \sigma_4 \\ \sigma_5 \\ \sigma_6 \end{bmatrix} = \begin{bmatrix} C_{11} & C_{12} & C_{13} & C_{14} & C_{15} & C_{16} \\ C_{12} & C_{22} & C_{23} & C_{24} & C_{25} & C_{26} \\ C_{13} & C_{23} & C_{33} & C_{34} & C_{35} & C_{36} \\ C_{14} & C_{24} & C_{34} & C_{44} & C_{45} & C_{46} \\ C_{15} & C_{25} & C_{35} & C_{45} & C_{55} & C_{56} \\ C_{16} & C_{26} & C_{36} & C_{46} & C_{56} & C_{66} \end{bmatrix} \begin{bmatrix} \varepsilon_1 \\ \varepsilon_2 \\ \varepsilon_3 \\ \varepsilon_4 \\ \varepsilon_5 \\ \varepsilon_6 \end{bmatrix} \quad (2.33)$$

Equation (2.33) is the most general expression of linear elasticity and is referred anisotropic material. Because there are no planes of symmetry for the material properties.

If there are three mutually perpendicular symmetry planes of material, then the stiffness matrix's independent constants numbers are decreased from 21 to 9, which is realized in general Hooke's law for orthotropic materials, equation (2.34). There are nine independent elastic constants are present. A single lamina of continuous fiber composite, arranged in a rectangular array, a wooden bar, and rolled steel can be given as examples of orthotropic materials.

$$\begin{bmatrix} \sigma_1 \\ \sigma_2 \\ \sigma_3 \\ \sigma_4 \\ \sigma_5 \\ \sigma_6 \end{bmatrix} = \begin{bmatrix} C_{11} & C_{12} & C_{13} & 0 & 0 & 0 \\ C_{12} & C_{22} & C_{23} & 0 & 0 & 0 \\ C_{13} & C_{23} & C_{33} & 0 & 0 & 0 \\ 0 & 0 & 0 & C_{44} & 0 & 0 \\ 0 & 0 & 0 & 0 & C_{55} & 0 \\ 0 & 0 & 0 & 0 & 0 & C_{66} \end{bmatrix} \begin{bmatrix} \varepsilon_1 \\ \varepsilon_2 \\ \varepsilon_3 \\ \varepsilon_4 \\ \varepsilon_5 \\ \varepsilon_6 \end{bmatrix} \quad (2.34)$$

It can be also given as the compliance matrix and $S = C^{-1}$.

$$\begin{bmatrix} \varepsilon_1 \\ \varepsilon_2 \\ \varepsilon_3 \\ \varepsilon_4 \\ \varepsilon_5 \\ \varepsilon_6 \end{bmatrix} = \begin{bmatrix} S_{11} & S_{12} & S_{13} & 0 & 0 & 0 \\ S_{12} & S_{22} & S_{23} & 0 & 0 & 0 \\ S_{13} & S_{23} & S_{33} & 0 & 0 & 0 \\ 0 & 0 & 0 & S_{44} & 0 & 0 \\ 0 & 0 & 0 & 0 & S_{55} & 0 \\ 0 & 0 & 0 & 0 & 0 & S_{66} \end{bmatrix} \begin{bmatrix} \sigma_1 \\ \sigma_2 \\ \sigma_3 \\ \sigma_4 \\ \sigma_5 \\ \sigma_6 \end{bmatrix} \quad (2.35)$$

If there is a plane of material isotropy in one of the planes of an orthotropic matter and direction 1 is normal to that plane (2-3) of isotropy, then the general Hooke's law for transversely isotropic material can be given in equation (2.36) and compliance form in equation (2.37). It can be realized that five independent elastic constants are present. A thin unidirectional lamina in which the fibers are arranged in a square or hexagonal array can be given as an example of transversely isotropic material.

$$\begin{bmatrix} \sigma_1 \\ \sigma_2 \\ \sigma_3 \\ \sigma_4 \\ \sigma_5 \\ \sigma_6 \end{bmatrix} = \begin{bmatrix} C_{11} & C_{12} & C_{12} & 0 & 0 & 0 \\ C_{12} & C_{22} & C_{23} & 0 & 0 & 0 \\ C_{12} & C_{23} & C_{22} & 0 & 0 & 0 \\ 0 & 0 & 0 & \frac{C_{22} - C_{23}}{2} & 0 & 0 \\ 0 & 0 & 0 & 0 & C_{55} & 0 \\ 0 & 0 & 0 & 0 & 0 & C_{55} \end{bmatrix} \begin{bmatrix} \varepsilon_1 \\ \varepsilon_2 \\ \varepsilon_3 \\ \varepsilon_4 \\ \varepsilon_5 \\ \varepsilon_6 \end{bmatrix} \quad (2.36)$$

$$\begin{bmatrix} \varepsilon_1 \\ \varepsilon_2 \\ \varepsilon_3 \\ \varepsilon_4 \\ \varepsilon_5 \\ \varepsilon_6 \end{bmatrix} = \begin{bmatrix} S_{11} & S_{12} & S_{12} & 0 & 0 & 0 \\ S_{12} & S_{22} & S_{23} & 0 & 0 & 0 \\ S_{12} & S_{23} & S_{22} & 0 & 0 & 0 \\ 0 & 0 & 0 & 2(S_{22} - S_{23}) & 0 & 0 \\ 0 & 0 & 0 & 0 & S_{55} & 0 \\ 0 & 0 & 0 & 0 & 0 & S_{55} \end{bmatrix} \begin{bmatrix} \sigma_1 \\ \sigma_2 \\ \sigma_3 \\ \sigma_4 \\ \sigma_5 \\ \sigma_6 \end{bmatrix} \quad (2.37)$$

Engineering constants such as Young's moduli, shear moduli, Poisson ratios, and other behavioral constants are measured in uniaxial tension or pure shear tests. These constants have more direct meaning than the components of relatively abstract stiffness and compliance matrices. Generally most material characterization tests are carried out with a known stress or load and hence, the displacement or strain results can be measured. The engineering constants are usually the slope of a stress-strain curve ($E = \sigma/\varepsilon$), or the slope of strain-strain curve ($\nu = -\varepsilon_y/\varepsilon_x$ for $\sigma_x = \sigma$ and all other stresses are zero). Therefore, instead of using the components of stiffness matrix (C_{ij}), the components of the compliance matrix (S_{ij}) are determined more useful.

In order to establish the relationship between engineering constants and the compliance coefficients, an orthotropic material in the principal material directions can be considered. If this orthotropic material is subjected to a 3D state of stress, the resulting strains can be expressed in terms of these stress components and engineering constants given equation (2.38).

$$\begin{aligned}
\varepsilon_{11} &= \frac{\sigma_{11}}{E_1} - \frac{\nu_{21}}{E_2} \sigma_{22} - \frac{\nu_{31}}{E_3} \sigma_{33} \\
\varepsilon_{22} &= -\frac{\nu_{12}}{E_1} \sigma_{11} + \frac{\sigma_{22}}{E_2} - \frac{\nu_{32}}{E_3} \sigma_{33} \\
\varepsilon_{33} &= -\frac{\nu_{13}}{E_1} \sigma_{11} - \frac{\nu_{23}}{E_2} \sigma_{22} + \frac{\sigma_{33}}{E_3}
\end{aligned} \tag{2.38}$$

Whereas the engineering shear strain components are given equation (2.39).

$$\gamma_{23} = \frac{\tau_{23}}{G_{23}} \quad , \quad \gamma_{13} = \frac{\tau_{13}}{G_{13}} \quad , \quad \gamma_{12} = \frac{\tau_{12}}{G_{12}} \tag{2.39}$$

Here, E_1 , E_2 , and E_3 are the Young's moduli in 1, 2, and 3 directions respectively (Figure 2.20). Therefore, E_1 represents the axial modulus and E_2 and E_3 represent in plane transverse and out-of-plane transverse moduli, respectively. It can be realized Figure 2.20, the axial direction is along the fiber direction. G_{ij} represents the shear moduli and G_{12} , G_{13} are the axial shear moduli in two orthogonal planes that include the fibers. G_{23} represents out-of-plane transverse shear modulus. Further, it should be noted that $G_{ij} = G_{ji}$. The Poisson ratio is defined as $\nu_{ij} = -\varepsilon_{jj}/\varepsilon_{ii}$ ($i \neq j$ no sum over i and j). ε_{ii} represents the strain in the direction of applied stress and ε_{jj} represents the strain the related lateral direction. It should be noted that $\nu_{ij} \neq \nu_{ji}$.

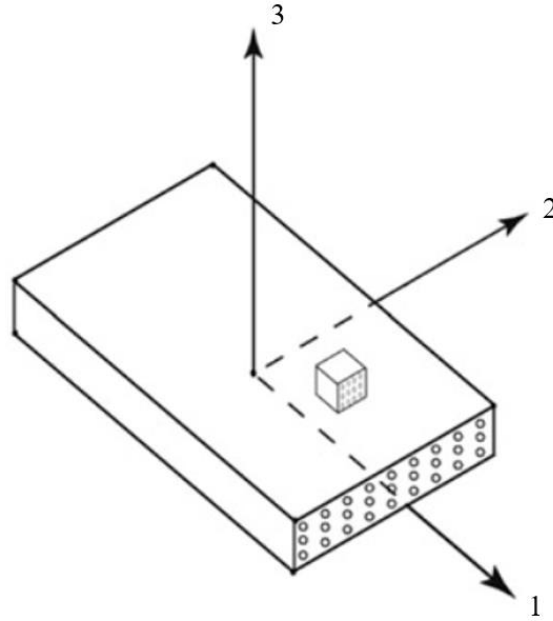


Figure 2.20 : Unidirectional lamina with principal material directions.

It can be assumed that all the engineering constants of an orthotropic material along principal directions have been measured. With these engineering constants it is known that the relation between the strain and stress components as given in Equation (2.38) and Equation (2.39). Thus, the strain components to stress components through compliance matrix can be related easily. Let reconsider the compliance matrix for orthotropic material equation (2.35). Now, compare equation (2.38) and equation (2.39) with equation (2.35). As a result of this, the compliance constants in terms of engineering constant can be obtain as in equation (2.40).

$$\begin{aligned}
S_{11} &= \frac{1}{E_1} \quad , \quad S_{12} = \frac{-\nu_{21}}{E_2} \quad , \quad S_{13} = \frac{-\nu_{31}}{E_3} \\
S_{21} &= \frac{-\nu_{12}}{E_1} \quad , \quad S_{22} = \frac{1}{E_2} \quad , \quad S_{23} = \frac{-\nu_{32}}{E_3} \\
S_{31} &= \frac{-\nu_{13}}{E_1} \quad , \quad S_{32} = \frac{-\nu_{23}}{E_2} \quad , \quad S_{33} = \frac{1}{E_3} \\
S_{44} &= \frac{1}{G_{23}} \quad , \quad S_{55} = \frac{1}{G_{13}} \quad , \quad S_{66} = \frac{1}{G_{12}}
\end{aligned} \tag{2.40}$$

It can be realized that both stiffness matrix and compliance matrix are symmetric. Now, for an orthotropic material, the compliance matrix components in terms of engineering constants can be written as equation (2.41).

$$[S_{ij}] = \begin{bmatrix} \frac{1}{E_1} & \frac{-\nu_{21}}{E_2} & \frac{-\nu_{31}}{E_3} & 0 & 0 & 0 \\ \frac{-\nu_{12}}{E_1} & \frac{1}{E_2} & \frac{-\nu_{32}}{E_3} & 0 & 0 & 0 \\ \frac{-\nu_{13}}{E_1} & \frac{-\nu_{23}}{E_2} & \frac{1}{E_3} & 0 & 0 & 0 \\ 0 & 0 & 0 & \frac{1}{G_{23}} & 0 & 0 \\ 0 & 0 & 0 & 0 & \frac{1}{G_{13}} & 0 \\ 0 & 0 & 0 & 0 & 0 & \frac{1}{G_{12}} \end{bmatrix} \tag{2.41}$$

Knowledge of linear algebra that inverse of a symmetric matrix is also a symmetric matrix. Since, the stiffness matrix, which is the inverse of compliance matrix, is symmetric; the compliance matrix has to be symmetric. More useful relations using

the symmetry of compliance matrix can be derived by comparing S_{12} and S_{21} . Thereby, it can be obtained that $\nu_{21}/E_2 = \nu_{12}/E_1$ and similarly, comparison of S_{13} and S_{31} and comparison of S_{23} and S_{32} give two more similar relations. All these relations are given in equation (2.42).

$$\frac{\nu_{21}}{E_2} = \frac{\nu_{12}}{E_1} \quad , \quad \frac{\nu_{31}}{E_3} = \frac{\nu_{13}}{E_1} \quad , \quad \frac{\nu_{23}}{E_2} = \frac{\nu_{32}}{E_3} \quad (2.42)$$

The relation in equation (2.42) is known as the reciprocal relations. These relations are also written as equation (2.43).

$$E_1 \nu_{21} = E_2 \nu_{12} \quad , \quad E_1 \nu_{31} = E_3 \nu_{13} \quad , \quad E_3 \nu_{23} = E_2 \nu_{32} \quad (2.43)$$

2.3.2.2 Stress-strain relations for a lamina of arbitrary orientation

The stresses and strains were defined in the principal material coordinates for an orthotropic material up to now. However, the principal directions of lamina often do not coincide with the global coordinate directions. Therefore, it is required to transform the stress and strain tensor from one coordinate axes system to another. It should be known that the stress and strain tensors are second order tensors and hence, tensor transformation rules should be followed. Consider an orthotropic lamina with its principal material axes oriented at an angle θ with the reference coordinate axes as shown in Figure (2.21).

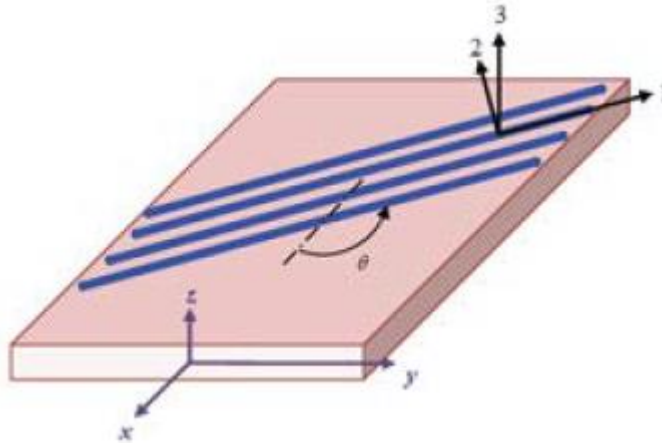


Figure 2.21 : Unidirectional lamina with global xyz directions and principal material 1-2-3 directions.

Primarily, two notations should be introduced the subscripts 123 and xyz. The subscripts 123 indicates a quantity such as engineering constants, constitutive equation

in principal material directions, while the subscripts xyz means the corresponding quantity in global coordinate directions. Here, 1-2 plane is rotated about direction 3 to x-y plane. Here, direction z and direction 3 are in same directions, that is, along the thickness direction of lamina. Therefore, in order to introduce transformation of stresses and strains easily, plane stress state can be considered. A plane stress state is defined by setting $\sigma_3=0$, $\tau_{23}=0$, and $\tau_{31}=0$. For orthotropic materials, plane stress assumptions cause implied out-of-plane strains of

$$\varepsilon_3 = S_{13}\sigma_1 + S_{23}\sigma_2 \quad \gamma_{23} = 0 \quad , \quad \gamma_{31} = 0 \quad (2.44)$$

$$S_{13} = -\frac{\nu_{13}}{E_1} = -\frac{\nu_{31}}{E_3} \quad , \quad S_{23} = -\frac{\nu_{23}}{E_2} = -\frac{\nu_{32}}{E_3} \quad (2.45)$$

Furthermore, the strain-stress relations in equation (2.35) contract to using by equation (2.44) and the strain-stress relations for plane stress approach can be written as equation (2.46).

$$\begin{bmatrix} \varepsilon_1 \\ \varepsilon_2 \\ \gamma_{12} \end{bmatrix} = \begin{bmatrix} S_{11} & S_{12} & 0 \\ S_{12} & S_{22} & 0 \\ 0 & 0 & S_{66} \end{bmatrix} \begin{bmatrix} \sigma_1 \\ \sigma_2 \\ \tau_{12} \end{bmatrix} \quad , \quad \begin{bmatrix} \sigma_1 \\ \sigma_2 \\ \tau_{12} \end{bmatrix} = \begin{bmatrix} Q_{11} & Q_{12} & 0 \\ Q_{12} & Q_{22} & 0 \\ 0 & 0 & Q_{66} \end{bmatrix} \begin{bmatrix} \varepsilon_1 \\ \varepsilon_2 \\ \gamma_{12} \end{bmatrix} \quad (2.46)$$

$$S_{11} = \frac{1}{E_1} \quad , \quad S_{11} = -\frac{\nu_{12}}{E_1} = -\frac{\nu_{21}}{E_2} \quad , \quad S_{22} = \frac{1}{E_2} \quad S_{66} = \frac{1}{G_{12}} \quad (2.47)$$

The Q_{ij} are the reduced stiffnesses for a plane stress state, which are determined as the components of the inverted compliance matrix in equation (2.46).

The transformation equations for obtaining stresses in a x-y coordinate system in terms of stresses in a 1-2 coordinate system, which is given as equation (2.48).

$$\begin{bmatrix} \sigma_x \\ \sigma_y \\ \tau_{xy} \end{bmatrix} = \begin{bmatrix} \cos^2\theta & \sin^2\theta & -2\sin\theta\cos\theta \\ \sin^2\theta & \cos^2\theta & 2\sin\theta\cos\theta \\ \sin\theta\cos\theta & -\sin\theta\cos\theta & \cos^2\theta - \sin^2\theta \end{bmatrix} \begin{bmatrix} \sigma_1 \\ \sigma_2 \\ \tau_{12} \end{bmatrix} \quad (2.48)$$

Similarly, the strain transformation is given as equation (2.49).

$$\begin{bmatrix} \varepsilon_x \\ \varepsilon_y \\ \frac{\gamma_{xy}}{2} \end{bmatrix} = \begin{bmatrix} \cos^2\theta & \sin^2\theta & -2\sin\theta\cos\theta \\ \sin^2\theta & \cos^2\theta & 2\sin\theta\cos\theta \\ \sin\theta\cos\theta & -\sin\theta\cos\theta & \cos^2\theta - \sin^2\theta \end{bmatrix} \begin{bmatrix} \varepsilon_1 \\ \varepsilon_2 \\ \frac{\gamma_{12}}{2} \end{bmatrix} \quad (2.49)$$

It is noticed that the direction of the rotation is significant and θ is the angle from the x- axis to the 1-axis (Figure 2.22)

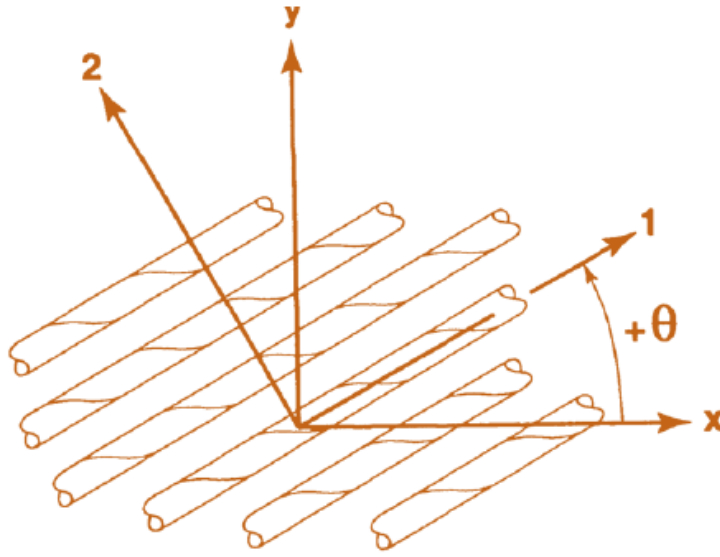


Figure 2.22 : Positive rotation of principal material axes from x-y axes.

The transformations are generally written as equation (2.50) for stresses and strains.

$$\begin{bmatrix} \sigma_x \\ \sigma_y \\ \tau_{xy} \end{bmatrix} = [T]^{-1} \begin{bmatrix} \sigma_1 \\ \sigma_2 \\ \tau_{12} \end{bmatrix}, \quad \begin{bmatrix} \varepsilon_x \\ \varepsilon_y \\ \frac{\gamma_{xy}}{2} \end{bmatrix} = [T]^{-1} \begin{bmatrix} \varepsilon_1 \\ \varepsilon_2 \\ \frac{\gamma_{12}}{2} \end{bmatrix} \quad (2.50)$$

The superscripts -1 indicates the matrix inverse and the transformation matrix is given equation (2.51).

$$[T] = \begin{bmatrix} \cos^2\theta & \sin^2\theta & 2\sin\theta\cos\theta \\ \sin^2\theta & \cos^2\theta & -2\sin\theta\cos\theta \\ -\sin\theta\cos\theta & \sin\theta\cos\theta & \cos^2\theta - \sin^2\theta \end{bmatrix} \quad (2.51)$$

Due to differences between engineering shear strains and tensor shear strains, the ordinary expressions for stiffness and compliance matrices with factors of 1/2 and 2 in several rows and columns can be avoided by introducing Reuter's transformation which is given as equation (2.52) with engineering strain vectors.

$$[R] = \begin{bmatrix} 1 & 0 & 0 \\ 0 & 1 & 0 \\ 0 & 0 & 2 \end{bmatrix} \quad \text{and} \quad \begin{bmatrix} \varepsilon_x \\ \varepsilon_y \\ \gamma_{xy} \end{bmatrix} = [R] \begin{bmatrix} \varepsilon_x \\ \varepsilon_y \\ \frac{\gamma_{xy}}{2} \end{bmatrix}, \quad \begin{bmatrix} \varepsilon_1 \\ \varepsilon_2 \\ \gamma_{12} \end{bmatrix} = [R] \begin{bmatrix} \varepsilon_1 \\ \varepsilon_2 \\ \frac{\gamma_{12}}{2} \end{bmatrix} \quad (2.52)$$

An orthotropic lamina with principal material axes are aligned with the same direction of the global coordinate system whose stresses definition is given as equation (2.53).

$$\begin{bmatrix} \sigma_x \\ \sigma_y \\ \tau_{xy} \end{bmatrix} = \begin{bmatrix} \sigma_1 \\ \sigma_2 \\ \tau_{12} \end{bmatrix} = \begin{bmatrix} Q_{11} & Q_{12} & 0 \\ Q_{12} & Q_{22} & 0 \\ 0 & 0 & Q_{66} \end{bmatrix} \begin{bmatrix} \varepsilon_1 \\ \varepsilon_2 \\ \gamma_{12} \end{bmatrix} \quad (2.53)$$

As it has been mentioned before, orthotropic laminae are generally built that the principal material orientation local coordinates do not coincide with the global coordinates system. Then, the fundamental question is: what are the stress-strain relations in x-y global coordinate systems when the stress-strain relations in the principal material local coordinates are known, and vice versa.

Accordingly, equation (2.50) and along with the Reuter' matrix, which is given in equation (2.52) and after abbreviating equation (2.54) in order to obtain stress-strain relations in x-y coordinates system, which is given as equation (2.55).

$$\begin{bmatrix} \sigma_1 \\ \sigma_2 \\ \tau_{12} \end{bmatrix} = [Q] \begin{bmatrix} \varepsilon_1 \\ \varepsilon_2 \\ \gamma_{12} \end{bmatrix} \quad (2.54)$$

$$\begin{bmatrix} \sigma_x \\ \sigma_y \\ \tau_{xy} \end{bmatrix} = [T]^{-1} \begin{bmatrix} \sigma_1 \\ \sigma_2 \\ \tau_{12} \end{bmatrix} = [T]^{-1}[Q][R][T][R]^{-1} \begin{bmatrix} \varepsilon_x \\ \varepsilon_y \\ \gamma_{xy} \end{bmatrix} \quad (2.55)$$

$[R][T][R]^{-1}$ can be written as $[T]^T$ where the superscript T means the matrix transpose. $\bar{Q}_{ij}$ can be defined as equation (2.56), which denotes the transformed reduced stiffnesses instead of the reduced stiffnesses, Q_{ij} by using abbreviation and $[T]^T$.

$$[\bar{Q}] = [T]^{-1}[Q][T]^T \quad (2.56)$$

Now, it can be defined the stress-strain relations in x-y coordinates.

$$\begin{bmatrix} \sigma_x \\ \sigma_y \\ \tau_{xy} \end{bmatrix} = [\bar{Q}] \begin{bmatrix} \varepsilon_x \\ \varepsilon_y \\ \gamma_{xy} \end{bmatrix} = \begin{bmatrix} \bar{Q}_{11} & \bar{Q}_{12} & \bar{Q}_{16} \\ \bar{Q}_{12} & \bar{Q}_{22} & \bar{Q}_{26} \\ \bar{Q}_{16} & \bar{Q}_{26} & \bar{Q}_{66} \end{bmatrix} \begin{bmatrix} \varepsilon_x \\ \varepsilon_y \\ \gamma_{xy} \end{bmatrix} \quad (2.57)$$

The transformed reduced stiffnesses can be calculated via reduced stiffnesses.

$$\begin{aligned} \bar{Q}_{11} &= Q_{11}\cos^4\theta + 2(Q_{12} + 2Q_{66})\sin^2\theta \cos^2\theta + Q_{22}\sin^4\theta \\ \bar{Q}_{12} &= (Q_{11} + Q_{22} - 4Q_{66})\sin^2\theta \cos^2\theta + Q_{12}(\sin^4\theta + \cos^4\theta) \\ \bar{Q}_{22} &= Q_{11}\sin^4\theta + 2(Q_{12} + 2Q_{66})\sin^2\theta \cos^2\theta + Q_{22}\cos^4\theta \\ \bar{Q}_{16} &= (Q_{11} - Q_{12} - 2Q_{66})\sin\theta \cos^3\theta + (Q_{12} - Q_{22} + 2Q_{66})\sin^3\theta \cos\theta \\ \bar{Q}_{26} &= (Q_{11} - Q_{12} - 2Q_{66})\sin^3\theta \cos\theta + (Q_{12} - Q_{22} + 2Q_{66})\sin\theta \cos^3\theta \\ \bar{Q}_{66} &= (Q_{11} + Q_{22} - 2Q_{12} - 2Q_{66})\sin^2\theta \cos^2\theta + Q_{66}(\sin^4\theta + \cos^4\theta) \end{aligned} \quad (2.58)$$

It can be realized that the transformed reduced stiffness matrix $\bar{Q}_{ij}$ has terms in all nine positions but the presence of zeros in the reduced stiffness matrix Q_{ij} inversely.

However, due to fact that the lamina is orthotropic, there are still only four independent material constant.

In order to express the strains in terms of the stresses in x-y global coordinates, there are two methods. One is inversion of the stress-strain relations in equation (2.57), and the other is transformation of the strain-stress relations in principal material local coordinates, which has almost same steps and procedure. Therefore, only the equations are given step by step as expressing the stress-strain relations in x-y global coordinates.

$$\begin{bmatrix} \epsilon_1 \\ \epsilon_2 \\ \gamma_{12} \end{bmatrix} = \begin{bmatrix} S_{11} & S_{12} & 0 \\ S_{12} & S_{22} & 0 \\ 0 & 0 & S_{66} \end{bmatrix} \begin{bmatrix} \sigma_1 \\ \sigma_2 \\ \tau_{12} \end{bmatrix} \quad (2.59)$$

$$[T]^T = [R][T]^{-1}[R]^{-1} \quad (2.60)$$

$$\begin{bmatrix} \epsilon_x \\ \epsilon_y \\ \gamma_{xy} \end{bmatrix} = [T]^T [S] [T] \begin{bmatrix} \sigma_x \\ \sigma_y \\ \tau_{xy} \end{bmatrix} = \begin{bmatrix} \bar{S}_{11} & \bar{S}_{12} & \bar{S}_{16} \\ \bar{S}_{12} & \bar{S}_{22} & \bar{S}_{26} \\ \bar{S}_{16} & \bar{S}_{26} & \bar{S}_{66} \end{bmatrix} \begin{bmatrix} \sigma_x \\ \sigma_y \\ \tau_{xy} \end{bmatrix} \quad (2.61)$$

$$\begin{aligned} \bar{S}_{11} &= S_{11} \cos^4 \theta + (2S_{12} + S_{66}) \sin^2 \theta \cos^2 \theta + S_{22} \sin^4 \theta \\ \bar{S}_{12} &= (S_{11} + S_{22} - S_{66}) \sin^2 \theta \cos^2 \theta + S_{12} (\sin^4 \theta + \cos^4 \theta) \\ \bar{S}_{22} &= S_{11} \sin^4 \theta + (2S_{12} + S_{66}) \sin^2 \theta \cos^2 \theta + S_{22} \cos^4 \theta \\ \bar{S}_{16} &= (2S_{11} - 2S_{12} - S_{66}) \sin \theta \cos^3 \theta - (2S_{22} - 2S_{12} - S_{66}) \sin^3 \theta \cos \theta \\ \bar{S}_{26} &= (2S_{11} - 2S_{12} - S_{66}) \sin^3 \theta \cos \theta - (2S_{22} - 2S_{12} - S_{66}) \sin \theta \cos^3 \theta \\ \bar{S}_{66} &= 2(2S_{11} + 2S_{22} - 4S_{12} - S_{66}) \sin^2 \theta \cos^2 \theta + S_{66} (\sin^4 \theta + \cos^4 \theta) \end{aligned} \quad (2.62)$$

2.4 Failure Theories for Composite Lamina

Composite lamina failure theories can be categorize in the following three groups:

- Non-interactive or limit theories: The failure modes are predicted by comparing individual stresses or strains respective to their ultimate stresses and strains. However, such theories do not deal with interply between different stress components. Maximum stress criteria, and maximum strain criteria can be given examples of such theories.
- Interactive theories: These theories go a step further than limit theories, and also consider interaction between various stress/strain components. Tsai-Wu

and Tsai-Hill can be given examples of such theories. They are able to estimate overall failure, but cannot estimate the exact failure mode.

- Failure mode based theories: These theories provide separate criteria for failure of matrix, fiber and interface. Puck, and Hashim-Rotem theories can be given examples.

Maximum stress, maximum strain, and Tsai-Hill failure theories are examined in this study.

2.4.1 Maximum stress theory

This theory calculates stresses in material directions on a layer-by-layer basis. The stresses acting on a lamina are caused the local axes normal and shear stresses. If any of the normal or shear stresses in local axes of a lamina is satisfied the related ultimate strengths of the unidirectional lamina, failure is occurred in the lamina. If the stresses or strains of the global axes of a lamina are known, then it can be found the stresses in the material local axes by using equation (2.50). The conditions of the maximum stress failure are given in equation (2.63). It can be first defined strength parameters, which is given Table 2.4.

Table 2.4 : Definitions of strength parameters for maximum stress theory.

Strength Parameters	Definitions
$(\sigma_1^T)_{ult}$	Ultimate longitudinal tensile strength (in direction 1),
$(\sigma_1^C)_{ult}$	Ultimate longitudinal compressive strength (in direction 1),
$(\sigma_2^T)_{ult}$	Ultimate transverse tensile strength (in direction 2),
$(\sigma_2^C)_{ult}$	Ultimate transverse compressive strength (in direction 2),
$(\tau_{12})_{ult}$	Ultimate in-plane shear strength (in plane 1-2)

These strength parameters cannot be changed for an angle lamina. Therefore, the failure theories such as maximum stress theory which examines local axes stress, then assess these five strength parameters of a unidirectional lamina whether the lamina has failed.

$$\begin{aligned}
-(\sigma_1^C)_{ult} < \sigma_1 < (\sigma_1^T)_{ult} \quad , \text{ or} \\
-(\sigma_2^C)_{ult} < \sigma_2 < (\sigma_2^T)_{ult} \quad , \text{ or} \\
-(\tau_{12})_{ult} < \tau_{12} < (\tau_{12})_{ult}
\end{aligned} \tag{2.63}$$

It can be realised that all five strength values are treated as positive, and the normal stresses are positive if tensile and negative if compressive.

2.4.2 Maximum strain failure theory

This theory calculates strains in material directions on a layer-by-layer basis. The strains acting on a lamina are caused the local axes normal and shear strains. If any of the normal or shear strains in local axes of a lamina is satisfied the related ultimate strains of the unidirectional lamina, failure is occurred in the lamina. If the stresses or strains of the global axes of a lamina are known, then it can be found the stresses in the material local axes by using equation (2.50). The conditions of the maximum strain failure are given in equation (2.64). It can be first defined strain parameters, which is given Table 2.5.

Table 2.5 : Definitions of strain parameters for maximum strain theory.

Strains Parameters	Definitions
$(\epsilon_1^T)_{ult}$	Ultimate longitudinal tensile strain (in direction 1),
$(\epsilon_1^C)_{ult}$	Ultimate longitudinal compressive strain (in direction 1),
$(\epsilon_2^T)_{ult}$	Ultimate transverse tensile strain (in direction 2),
$(\epsilon_2^C)_{ult}$	Ultimate transverse compressive strain (in direction 2),
$(\gamma_{12})_{ult}$	Ultimate in-plane shear strain (in plane 1-2)

These strain parameters cannot be changed for an angle lamina. Therefore, the failure theories such as maximum strain theory which examines local axes strains, then assess these five strains parameters of a unidirectional lamina whether the lamina has failed.

$$\begin{aligned}
-(\epsilon_1^C)_{ult} < \epsilon_1 < (\epsilon_1^T)_{ult} \quad , \text{ or} \\
-(\epsilon_2^C)_{ult} < \epsilon_2 < (\epsilon_2^T)_{ult} \quad , \text{ or} \\
-(\gamma_{12})_{ult} < \gamma_{12} < (\gamma_{12})_{ult}
\end{aligned} \tag{2.64}$$

The maximum stress and strain theories are similar, both have no interaction between several components of strain or stress. However, due to local strains in a lamina including Poisson's ratio effects, the two failure theories have different results. If the Poisson's ratio is zero for the unidirectional lamina, then they will give identical results.

2.4.3 Tsai-Hill failure theory

This theory is based on distortion energy yield criterion for anisotropic materials. The total strain energy, which contains distortion energy, consists of two parts. One is dilation energy which is caused change in volume and the second is distortion energy, which is induced a change in shape. If the distortion energy is greater than the failure distortion energy of the material, failure in the material occurs. Hill first studied von-Mises' distortional energy yield criterion for anisotropic materials. Then, Tsai developed it for a unidirectional lamina. He proposed equation (2.65) that based on the distortion energy theory whether lamina has failed.

$$\begin{aligned}
(G_2 + G_3)\sigma_1^2 + (G_1 + G_3)\sigma_2^2 + (G_1 + G_2)\sigma_3^2 - 2G_3\sigma_1\sigma_2 - 2G_2\sigma_1\sigma_3 \\
- 2G_1\sigma_2\sigma_3 + 2G_4\tau_{23}^2 + 2G_5\tau_{13}^2 + 2G_6\tau_{12}^2 < 1
\end{aligned} \tag{2.65}$$

The all G components in equation (2.65) are depend on and determined by the failure strengths of the materials. The equation (2.65) is used and calculated via applied defined stress and defined directions which are given as below.

- Apply $\sigma_1 = (\sigma_1^T)_{ult}$ to a unidirectional lamina, and then failure of the lamina will occur. Therefore, equation (2.65) is changed as given in equation (2.66).

$$(G_2 + G_3)(\sigma_1^T)_{ult}^2 = 1 \tag{2.66}$$

- Apply $\sigma_2 = (\sigma_2^T)_{ult}$ to a unidirectional lamina, and then failure of the lamina will occur. Therefore, equation (2.65) is changed as given in equation (2.67).

$$(G_1 + G_3)(\sigma_2^T)_{ult}^2 = 1 \quad (2.67)$$

- Apply $\sigma_3 = (\sigma_2^T)_{ult}$ to a unidirectional lamina, and then failure of the lamina will occur assuming that the normal tensile failure strength is same directions (2) and (3). Thus, equation (2.65) is changed as given in equation (2.68).

$$(G_1 + G_2)(\sigma_2^T)_{ult}^2 = 1 \quad (2.68)$$

- Apply $\tau_{12} = (\tau_{12})_{ult}$ to a unidirectional lamina, and then failure of the lamina will occur. Thereby, equation (2.65) is changed as given in equation (2.69).

$$2G_6(\tau_{12})_{ult}^2 = 1 \quad (2.69)$$

G_1, G_2, G_3 , and G_6 are found by using equations from (2.66) to (2.69). Which are given as equation (2.70).

$$G_1 = \frac{1}{2} \left(\frac{2}{[(\sigma_2^T)_{ult}]^2} - \frac{1}{[(\sigma_1^T)_{ult}]^2} \right), \quad G_2 = G_3 = \frac{1}{2} \left(\frac{1}{[(\sigma_1^T)_{ult}]^2} \right), \quad (2.70)$$

$$G_6 = \frac{1}{2} \left(\frac{1}{[(\tau_{12})_{ult}]^2} \right)$$

Due to plane stress assumption of the unidirectional lamina, that is, $\sigma_3 = \tau_{31} = \tau_{13} = 0$, then equation (2.65) can be simplified through equation (2.70) as given in equation (2.71).

$$\left[\frac{\sigma_1}{(\sigma_1^T)_{ult}} \right]^2 - \left[\frac{\sigma_1 \sigma_2}{(\sigma_1^T)_{ult}^2} \right] + \left[\frac{\sigma_2}{[(\sigma_2^T)_{ult}]} \right]^2 + \left[\frac{\tau_{12}}{(\tau_{12})_{ult}} \right]^2 < 1 \quad (2.71)$$

The lamina's global stresses are known, then the local stresses in a lamina can be found and determine whether the lamina has failed by applying the corresponding failure theory.

3. ADHESIVE BONDED JOINTS

3.1 Introduction

Adhesive joints play a significant role in the lightweight construction fields. In particular, the use of composites favors the structural adhesive bonding technology that stimulates growing research in adhesive technology nowadays. Adhesives are not generally as strong as the materials they join and therefore use is often made of an overlap to enhance the load carrying capacity of a joint. In general, the most common approach is to provide that the bonded joint is stronger than the parts being joined when designing a bonded joint. However, the load carrying capacity of an overlap joint is not proportional to the length of the overlap. The simple single lap joint which is illustrated in Figure 3.1, can be thought over. The axial load is transferred from one adherend to the other via shear in the adhesive. Due to adhesive nature, this shear occurs only finite region of the overlap and hence increasing overlap length does not cause any enhancements of the joint strength. However, it can be said that there must be optimum length of the overlap region.

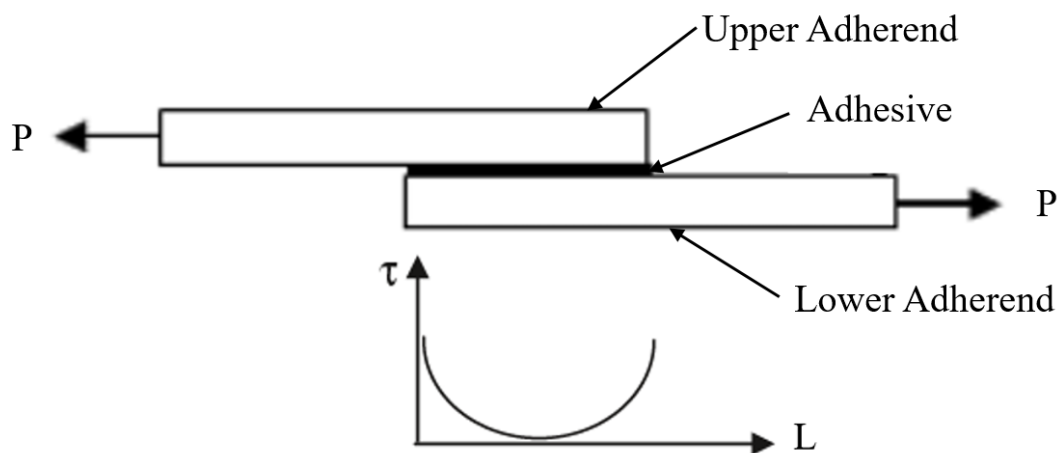


Figure 3.1 : Shear stress distribution in the adhesive layer for single-lap joint.

In practice, due to occurring the adherends deformation that means they are not rigid, as the load transfers from the other adherend, a non-uniform shear stress distribution is occurred, which is shown in Figure 3.2.

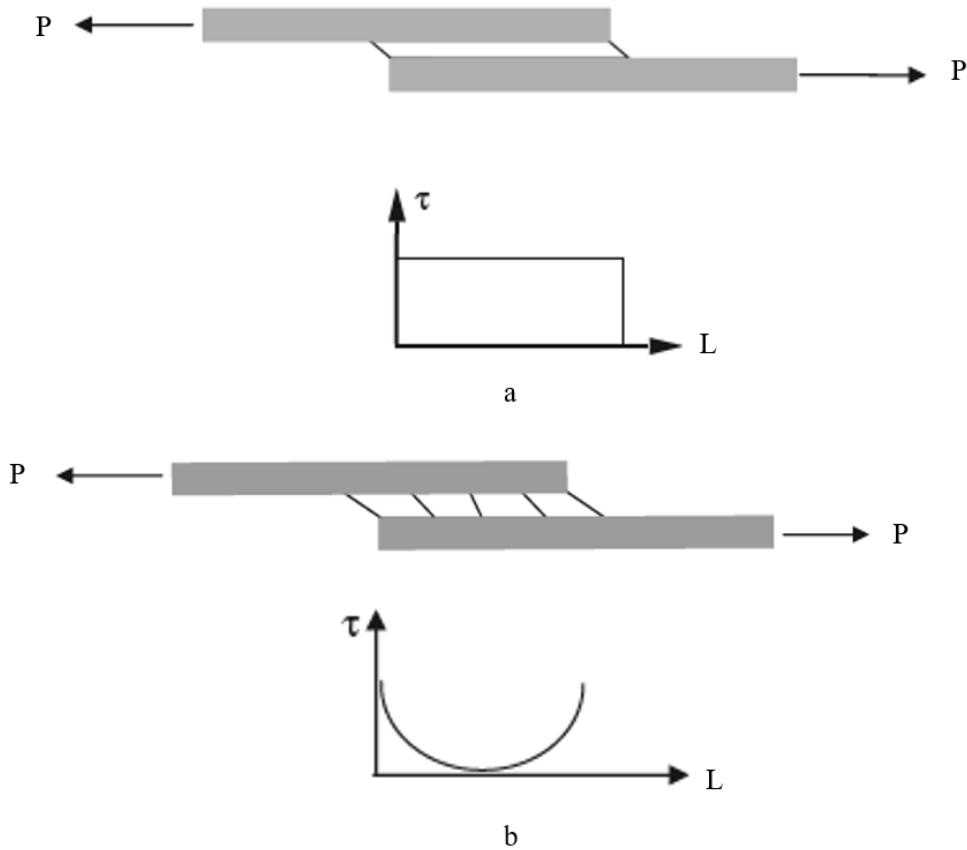


Figure 3.2 : Uniform (a) and non-uniform (b) shear stress distributions.

Besides the shearing the adhesive, the offset loading results in bending in the loaded adherend, which struggles to split from the unloaded adherend. This bending deformation causes increasing transverse direct stresses which are called cleavage or peel stresses, addition to non-uniform shear stress. Maximum peel stresses occur at the ends of the overlap region, which is shown in Figure 3.3. Although the single lap joint has simple geometry, it has fairly complex state of adhesive stresses.

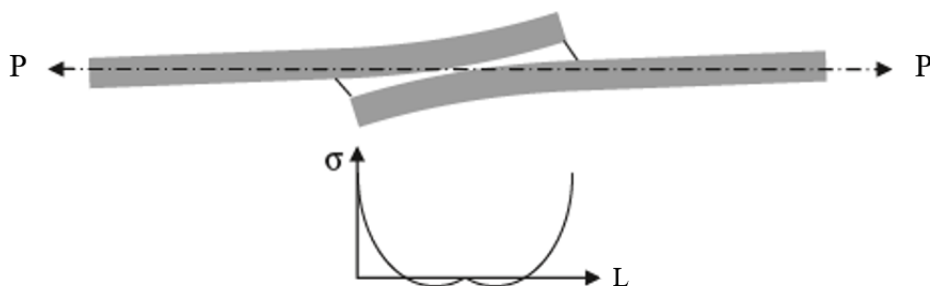


Figure 3.3 : Peel stress distribution in the adhesive layer for single-lap joint.

Double lap joints (Figure 3.4), which are basically two single laps-back to-back, can prevent joint rotation due to no net bending moment on a symmetrical double lap

bending joint. Because the force is applied through the adhesive to the adherends away from their neutral axes, but internal bending moments arise in the outer adherends. Therefore, direct stresses acting normal to the adhesive layer, that is, peel stresses occur and it can play a key role in failure examination.

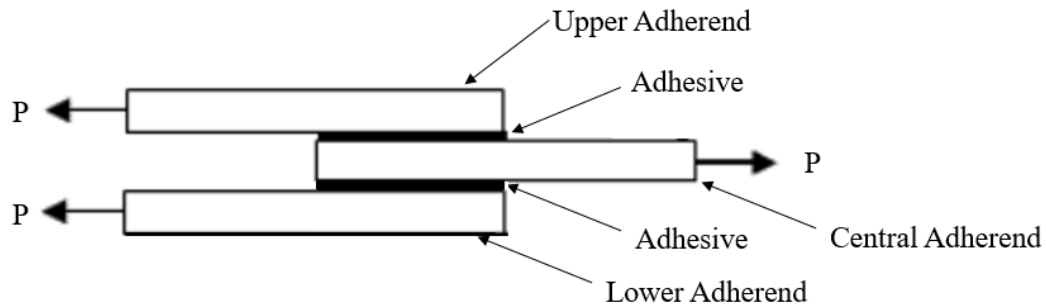


Figure 3.4 : A double lap joint.

Adhesive failure can be defined as follows: When the adherends are subjected to a load, failure, or de-bonding, failure happens when the parts separate prematurely. Adhesive or cohesive are two types of failure, which are shown in Figure 3.5. The adhesive failure mode occurs when the adhesive pulls away from one or both of the adherends, but does not tear or split. When cohesive failure occurs, you can see a physical separation of the adhesive material.

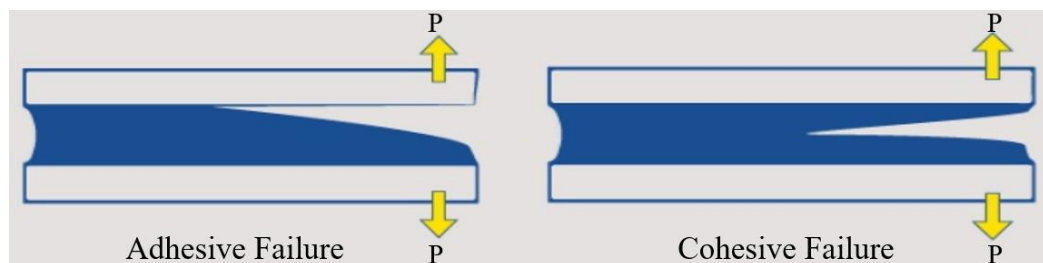


Figure 3.5 : Adhesive and cohesive failures.

There are most common reasons of those failures such as environmental factors, inappropriate adhesive, lack of elasticity and strength, surface preparation, and improper curing of the adhesive.

1. **Environmental factors:** Even if the type of adhesive is appropriate for the application, an unexpected change in the environment can result in the adhesive to fail. Heat, cold, moisture, and the introduction of chemicals are all potential failure factors.

2. **Inappropriate adhesive:** When selecting an adhesive, you should take into account all aspect of the conditions and the adherends being joined, the environment it will be applied & used in, and how long it will have to be operational.
3. **Lack of elasticity and strength:** When selecting an adhesive, flexibility and adhesion levels are crucial components to think about and optimum value must be defined according to the specific conditions. The specific type and amount of stress on the joint must be taken into consideration.
4. **Surface preparation:** Careful consideration must be given to the cleanliness of the all adherends. Grease, oil, dust, or dirt are a few examples that can cause poor bonding. They can be caused any failures in adhesively bonded joints substantially.
5. **Improper curing of the adhesive:** Adhesives require specific actions in order to cure. Some of those include time, air flow, and the amount of pressure used during application. If any or all of these requirements are not carried out, then failure can be possible occurred.

The adhesive performance in a joint changes according to many factors, such as strength and thickness of the adherends, joint design, the conditions of surfaces being bounded, the adhesive thickness, the bounding technique employed. Joint design defines the types and degree of the stress. Bonds may be stressed in shear, tension, compression, cleavage or peel, or any of these stresses and most adhesives display optimum strength properties in tension or compression [27]. The joint stress types are shown in Figure 3.6.

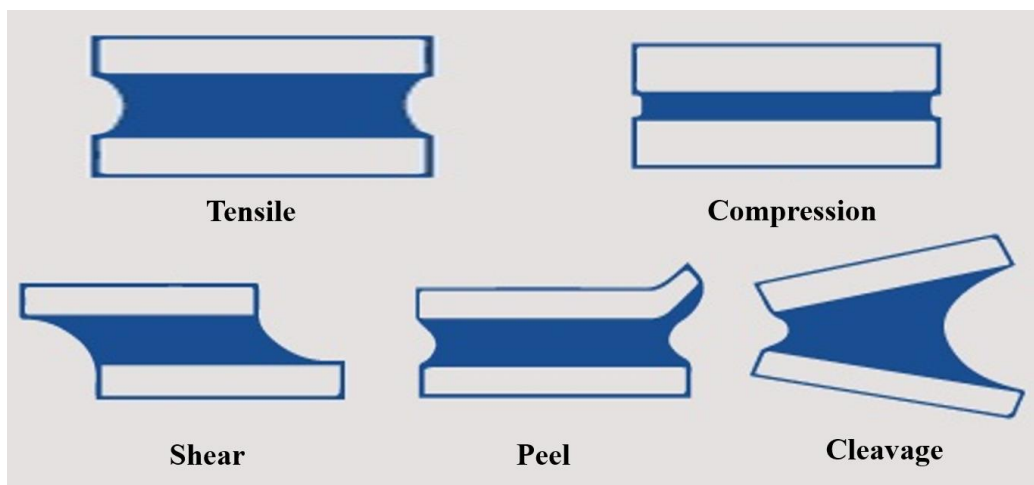


Figure 3.6 : Types of the joint stress.

Some of the typical bonded joint configurations are shown in Figure 3.7 [28]. Although the largest users of the adhesive materials are generally construction industries, some applications contain adhesive bonded joints which are required to endure large externally applied loads. Adhesive bonding plays key role for the utilization of new material and for the development of novel and reliable design concepts and structural configurations. As stated previously, the single lap joint is largely used in industry. Because, it is cheap and easy to manufacture and inspect ultrasonically. Moreover, the single lap joint under tension loading causes a problem which is often used to indicate the accuracy of theories for predicting the adhesively bonded joint stresses. In this study, single lap and double lap joints are examined in details.

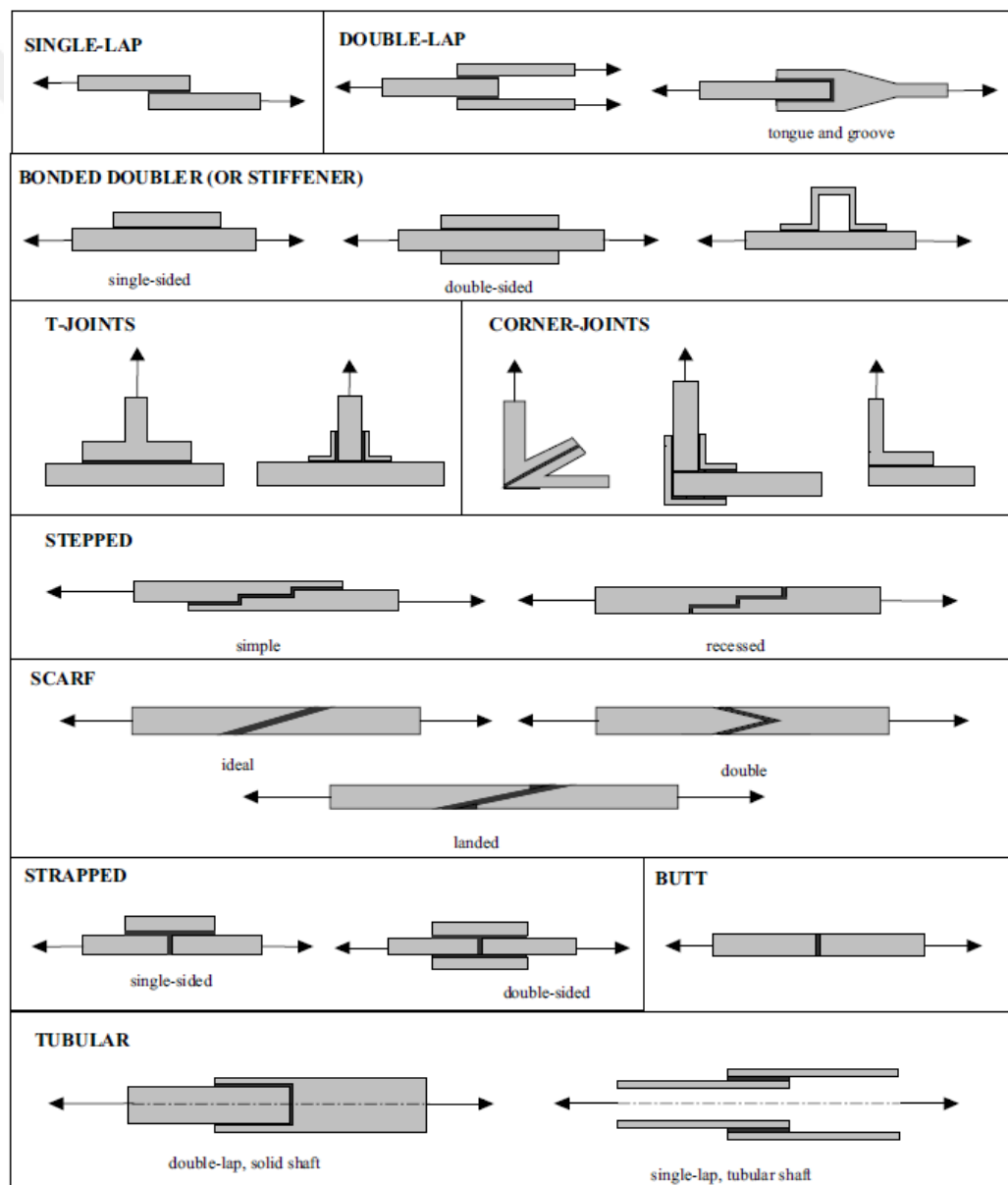


Figure 3.7 : Typical structural adhesive joint configurations.

3.2 Analytical Models

3.2.1 Volkersen's shear lag analysis

The first bonded joint analytical approach was developed by Volkersen (1938) and examined differential shear stress distribution, which is shown in Figure 3.8. The adhesive shear stress, τ , is defined in terms of the shear strain, γ , at distance x from the centre of the overlap region in equation (3.1), and adhesive length and thickness are defined as L and η , respectively. Modulus of rigidity is represented by G .

$$\tau = G\gamma = G\frac{\delta}{\eta} = G\frac{(\delta_0 + u_2 - u_1)}{\eta} \quad (3.1)$$

Adherends strains (ϵ_1 and ϵ_2) can be derived via extension of the adherends as u_1 and u_2 which is given as equation (3.2).

$$u_1 = \int_{-L/2}^x \epsilon_1 dx, \quad u_2 = \int_{-L/2}^x \epsilon_2 dx \quad (3.2)$$

These strains at distance x can be obtained lower and upper adherends' stresses, which can be stated by the adherends force P , tensile modulus, E_i and thickness, t_i , as given in equation (3.3) and shown in Figure 3.8.

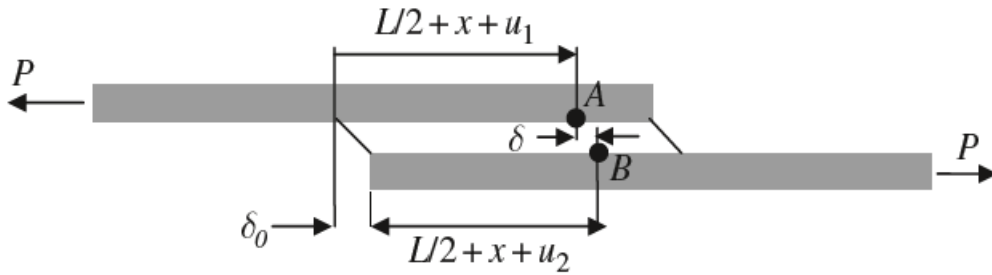


Figure 3.8 : Volkersen's shear lag analysis deformation [32].

$$\epsilon_1 = \frac{[P - F_\tau]}{E_1 t_1} = \frac{\left[P - \int_{-L/2}^x \tau dx \right]}{E_1 t_1}, \quad (3.3)$$

$$\epsilon_2 = \frac{F_\tau}{E_2 t_2} = \left[\frac{\int_{-L/2}^x \tau dx}{E_2 t_2} \right]$$

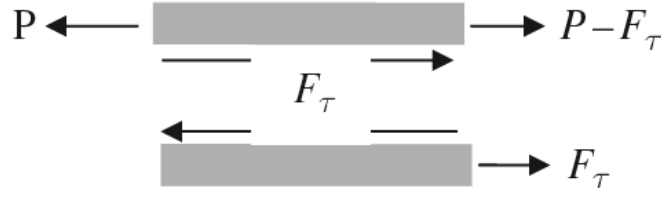


Figure 3.9 : Adherends forces.

If the adherends are considered same material then $E_1=E_2=E_A$. By substituting first equation (3.1) into equation (3.3), then equation (3.3) into equation (3.2) and then in order to obtain the adhesive shear stress differentiating twice the second order differential governing equation, which is given by equation (3.4).

$$\frac{d^2\tau}{dx^2} = \left[\frac{w}{L}\right]^2 \tau, \quad (3.4)$$

$$w^2 = \left[1 + \left(\frac{t_1}{t_2}\right)\right] k, \text{ and } k = \frac{GL^2}{(E_A t_1 \eta)}$$

This can be solved and non-dimensionalized to give a solution.

$$\frac{\tau}{\tau_a} = \frac{w \cosh(wX)}{2 \sinh(\frac{w}{2})} + \left(\frac{t_1 - t_2}{t_1 + t_2}\right) \frac{w \sinh(wX)}{2 \cosh(\frac{w}{2})} \quad (3.5)$$

τ_a is the average shear stress (P/L) and $X=x/L$. By considering only the peak stress at the joint ends ($X=\pm 0.5$) and suggesting that the joint is adequately long such that $w/2 = \cosh(w/2)$, then it can be simplified as equation (3.6).

$$\frac{\tau}{\tau_a} = \frac{w}{2} + \left(\frac{t_1 - t_2}{t_1 + t_2}\right) \quad (3.6)$$

It can be realised that, the adhesive stress is maximum and hence failure most likely at the overlap end where the loaded adherend is thinnest for a joint with different adherends.

Moreover, when the adherends are identical ($t_1=t_2=t$), the lowest adhesive stress are obtained, which is given as equation (3.7).

$$\frac{\tau}{\tau_a} = \frac{w}{2} = \sqrt{\frac{GL^2}{2E_A t \eta}} \quad (3.7)$$

It can be deduced from Equation (3.7) which is a useful formula. Because it gives some information about effects of the size of the peak adhesive stress in single lap joint. If the length of joint is long then, the size of the peak adhesive stress is independent of the joint length. The size of the peak stress increases with decreasing adherends thickness, elastic modulus and adhesive thickness, and it increases with increasing adhesive shear modulus [32].

3.2.2 Goland and Reissner model

The first bonded joint analytical model to take account of the effects owing to adherends rotation. The model approach is divided into two stages. The first is determination of the loads at the ends of the overlap region by using the finite deflection of cylindrically bent plate theory. The second is determination of joints stresses because of the applied loads. The assumptions of the Goland and Reissner method are the normal stress parallel to the layer in the adhesive not taken into account, and all other normal stresses in the adhesive do not change through the thickness and the problem is one of the plane strain (Figure 3.10).

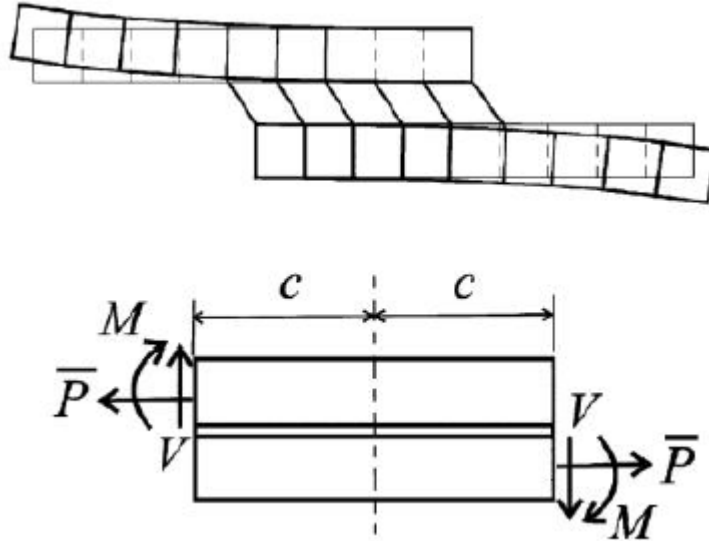


Figure 3.10 : Goland and Reissner analytical model [33].

The shear stress distribution of Goland and Reissner is given by equation (3.8).

$$\tau = \left(-\frac{1}{8} \frac{\bar{P}}{c} \right) \left[\frac{\beta c}{t} (1 + 3k) \frac{\cosh\left(\frac{\beta c}{t} \frac{x}{c}\right)}{\sinh\left(\frac{\beta c}{t}\right)} + 3(1 - k) \right] \quad (3.8)$$

$$k = \frac{\cosh(u_2 c)}{\cosh(u_2 c) + 2\sqrt{2} \sinh(u_2 c)}, \quad u_2 = \frac{1}{t} \sqrt{\left(\frac{\bar{P}}{tE}\right) \left(\frac{3(1-v^2)}{2}\right)} \quad (3.9)$$

$$\beta^2 = 8 \frac{G_a}{E} \frac{t}{t_a}$$

$\bar{P}$ is the applied tensile load per unit width, t and t_a are the thickness of the adherend and adhesive, respectively, k is the bending moment factor, c is half of the overlap length, G_a is adhesive shear modulus, and E is the elastic modulus of the adherend.

The adhesive peel stress (σ) distribution is given by equation (3.10).

$$\sigma = \frac{1}{\Delta} \frac{Pt}{c^2} (A + B) \quad (3.10)$$

$$\begin{aligned} A &= \left(R_2 \lambda^2 \frac{k}{2} + \lambda k' \cosh(\lambda) \cos(\lambda) \right) \cosh\left(\frac{\lambda x}{c}\right) \cos\left(\frac{\lambda x}{c}\right), \\ B &= \left(R_1 \lambda^2 \frac{k}{2} + \lambda k' \sinh(\lambda) \sin(\lambda) \right) \sinh\left(\frac{\lambda x}{c}\right) \sin\left(\frac{\lambda x}{c}\right), \\ k' &= \frac{kc}{t} \sqrt{\left(\frac{\bar{P}}{tE}\right) (3(1-v^2))}, \quad k': \text{The transverse force factor} \end{aligned} \quad (3.11)$$

$$\lambda = \gamma \frac{c}{t}, \quad \gamma^4 = 6 \frac{E_a}{E} \frac{t}{t_a}, \quad \Delta = \frac{1}{2} (\sin(2\lambda) + \sinh(\lambda)),$$

$$R_1 = \cosh(\lambda) \sin(\lambda) + \sinh(\lambda) \cos(\lambda),$$

$$R_2 = -\cosh(\lambda) \sin(\lambda) + \sinh(\lambda) \cos(\lambda)$$

Equation (3.11) is defined some variables which are used in equation (3.10). The studies of Volkersen and Goland Reissner were established a basic approach in stress analysis of adhesively bonded joint, there are also some limitations [34]. The proper model can be used regarding the stresses and conditions of applicability. For instance, where the joint bending is not taken into account and the adhesive is brittle, it can be suggested to use Volkersen's analysis method. However, when peel and yield stresses of the adherends and/or the adhesive are examined, then it is necessary to perform more complex model.



4. FINITE ELEMENT MODEL

4.1 Introduction

The finite element analysis has become more attractive in recent years, especially detailed analysis of bonded joints or complex geometries, for failure load prediction. It is essential to obtain sufficiently accurate results by refining the mesh in the high stress concentration regions. The challenging problem is in the interpretation of the data due to the stress singularity. In the bonded joints models due to presence of the sharp corners, stress singularities occur at points A, F, C, and D in Figure 4.1. Because of the stress singularities, mesh density is critical factor especially in these points region. In this study, fine mesh scheme was used in these critical region and similarly coarse mesh scheme was used in non-critical region, in order to decrease computing cost.

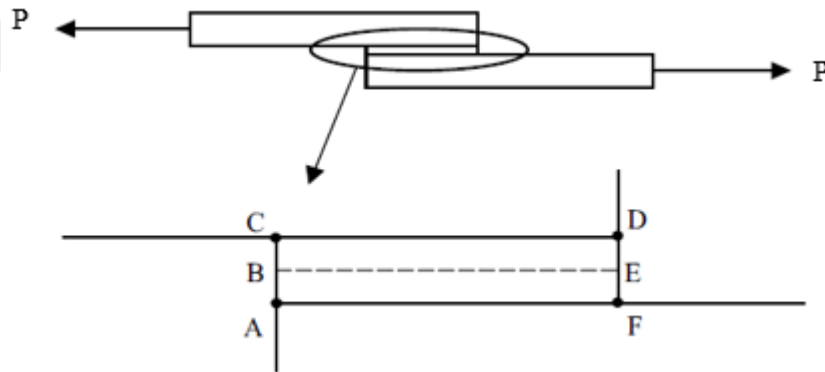


Figure 4.1 : The points of singularity and centerline of the bondline.

A single lap adhesive bonded joint used by Tsai and Morton [35] and a new material model for 2D FE analysis, which is proposed by Zhao [6] were based on for this study. An arbitrary unidirectional fiber reinforced composite ply with principal axes 1 (longitudinal, fiber direction), 2 (transverse direction), and 3 (interlaminar direction) for both single lap joint and double lap joint. Zhao's material model for 2D FE analysis is expressed and examined in detail, as shown in Figure 4.2.

A ply can be presumed as transversely isotropic materials according to the orthotropic elastic mechanics theory, and the constitute relationship was already given by equation (2.37).

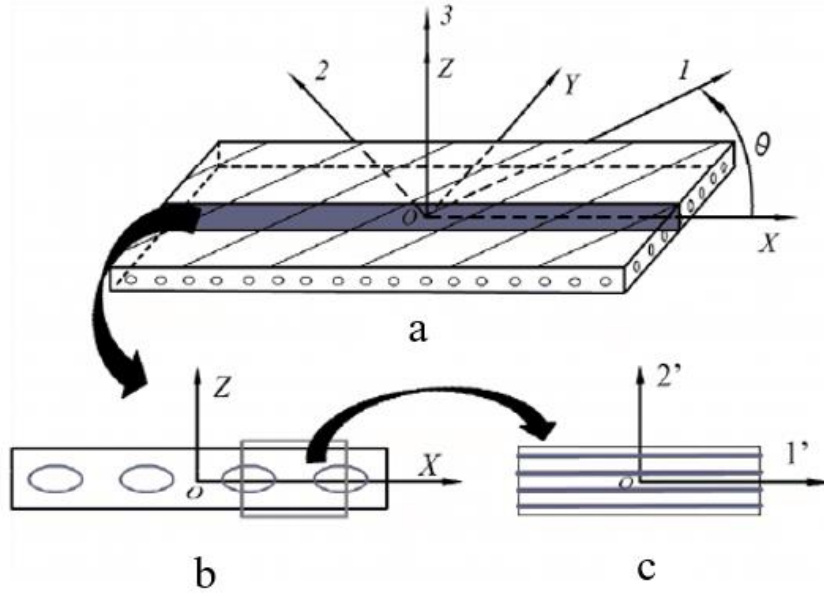


Figure 4.2 : Sketch map of proposed material model: a- composite ply with an arbitrary angle; b- middle plane; c- equivalent ply.

It can be seen from Figure 4.2, a section, referred to the coordinate system O-XYZ, the ply is an orthotropic with one symmetric plane X-O-Y, and the off-axis stiffness matrix $[\bar{Q}]$ is given by equation (4.1) [6].

$$\begin{bmatrix} \bar{Q}_{11} & \bar{Q}_{12} & \bar{Q}_{13} & 0 & 0 & \bar{Q}_{16} \\ \bar{Q}_{32} & \bar{Q}_{22} & \bar{Q}_{23} & 0 & 0 & \bar{Q}_{26} \\ \bar{Q}_{31} & \bar{Q}_{32} & \bar{Q}_{33} & 0 & 0 & \bar{Q}_{36} \\ 0 & 0 & 0 & \bar{Q}_{44} & \bar{Q}_{45} & 0 \\ 0 & 0 & 0 & \bar{Q}_{54} & \bar{Q}_{55} & 0 \\ \bar{Q}_{61} & \bar{Q}_{62} & \bar{Q}_{63} & 0 & 0 & \bar{Q}_{66} \end{bmatrix} \quad (4.1)$$

As the composite laminates are always symmetrical balanced when the bending twist coupling of ply is ignored. Furthermore, adhesively bonded joints are commonly wide compared with its thickness. Therefore, the plane strain assumption can be considered in the stress analysis of the joints in the plane X-O-Z, and $\varepsilon_y, \gamma_{yz}, \gamma_{xy}$ are assumed to be zero. Due to plane strain assumption, the constitutive equation in X-O-Z plane, which is given in Figure 4.2, b section, can be written as equation (4.2) [6].

$$\begin{bmatrix} \sigma_x \\ \sigma_z \\ \tau_{xz} \end{bmatrix} = \begin{bmatrix} \bar{Q}_{11} & \bar{Q}_{13} & 0 \\ \bar{Q}_{31} & \bar{Q}_{33} & 0 \\ 0 & 0 & \bar{Q}_{55} \end{bmatrix} \begin{bmatrix} \varepsilon_x \\ \varepsilon_z \\ \gamma_{xz} \end{bmatrix}, \quad \begin{bmatrix} \varepsilon_x \\ \varepsilon_z \\ \gamma_{xz} \end{bmatrix} = \begin{bmatrix} \bar{S}_{11} & \bar{S}_{12} & 0 \\ \bar{S}_{12} & \bar{S}_{22} & 0 \\ 0 & 0 & \bar{S}_{33} \end{bmatrix} \begin{bmatrix} \sigma_x \\ \sigma_z \\ \tau_{xz} \end{bmatrix} \quad (4.2)$$

In Zhao's study, an equivalent lamina (Figure 4.2, c section) is proposed to characterize the material properties in X-O-Z plane. Therefore, the proposed

equivalent lamina has the same constitutive relationship as equation (4.2). The principal axes of the equivalent lamina consider as 1' (longitudinal, fiber direction) and 2' (transverse direction), and the engineering constants can be represented by $E_{1'}$, $E_{2'}$, $\nu_{1'2'}$, and $G_{1'2'}$. The strain-stress relationship of the equivalent lamina is given by equation (4.3).

$$\begin{bmatrix} \varepsilon_{1'} \\ \varepsilon_{2'} \\ \gamma_{1'2'} \end{bmatrix} = \begin{bmatrix} 1/E_{1'} & \nu_{1'2'}/E_{1'} & 0 \\ \nu_{1'2'}/E_{1'} & 1/E_{2'} & 0 \\ 0 & 0 & 1/G_{1'2'} \end{bmatrix} \begin{bmatrix} \sigma_{1'} \\ \sigma_{2'} \\ \tau_{1'2'} \end{bmatrix} \quad (4.3)$$

Engineering constants now can be defined as equation (4.4).

$$E_{1'} = \frac{1}{\bar{S}_{11}}, \quad E_{2'} = \frac{1}{\bar{S}_{22}}, \quad \nu_{1'2'} = \frac{-\bar{S}_{12}}{\bar{S}_{11}}, \quad G_{1'2'} = \frac{1}{\bar{S}_{33}} \quad (4.4)$$

The Zhao's material model provides at the level of physical composite ply with arbitrary angle via equation (4.4). Therefore 2D FE models of the adhesively bonded composite joints can be established with the FE software program.

4.2 Description of the Finite Element Models

4.2.1 Finite element models of single lap and double lap joints

The joints configuration dimensions for both SLJ and DLJ are shown in Figure 4.3 and Figure 4.4, respectively. A uniform tensile force $P = 4448 \text{ N}$ [35] was applied at the right end of the upper adherend longitudinal direction for SLJ. A uniform tensile force $P = 8896 \text{ N}$, which are two times of SLJ's force, was applied at the right end of central adherend along the longitudinal direction for DLJ. The lower composite adherend left end was totally fixed with a length of 20 mm, while the upper composite adherend right end was only limited in transverse and normal direction with the length of 20 mm for SLJ (Figure 4.5). The left end of both lower and upper adherend were totally fixed with a length of 20 mm, while the right end of middle or central adherend was only restricted in transverse and normal direction with the length of 20 mm for DLJ (Figure 4.6). The both dimensions of the joints are listed in Table 4.1. The tie contact is defined for all contact regions which are upper adherend - adhesive and lower adherend - adhesive. In general, all models contact regions are defined as tie contact in this study. The adherends for both joints were made of graphite/epoxy (XAS/914C)

unidirectional composite materials with a lay-up of $[0/45/45/0]_s$. Epoxy adhesive (Hexcel Redux 308A) was used for the adhesive layer for stress analysis of SLJ and DLJ. In order to generate configuration of the adhesively bonded joints, three other adhesive material are used in the effect of the rigidity and pattern shape of the adhesive. The detail information will be given in effect of rigidity and pattern shape of the adhesive chapter.

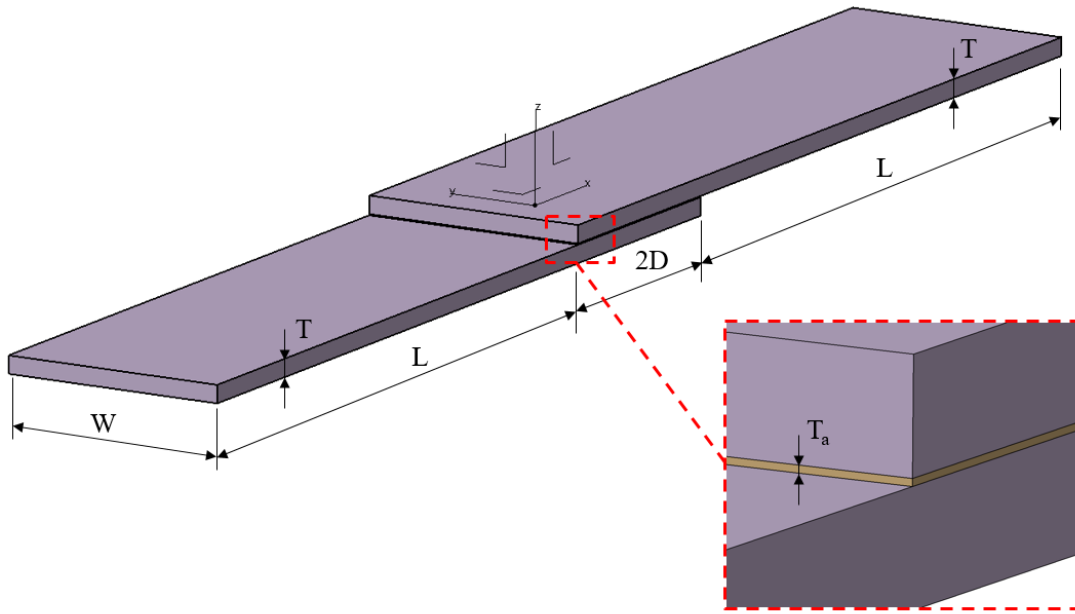


Figure 4.3 : SLJ configuration and dimensions.

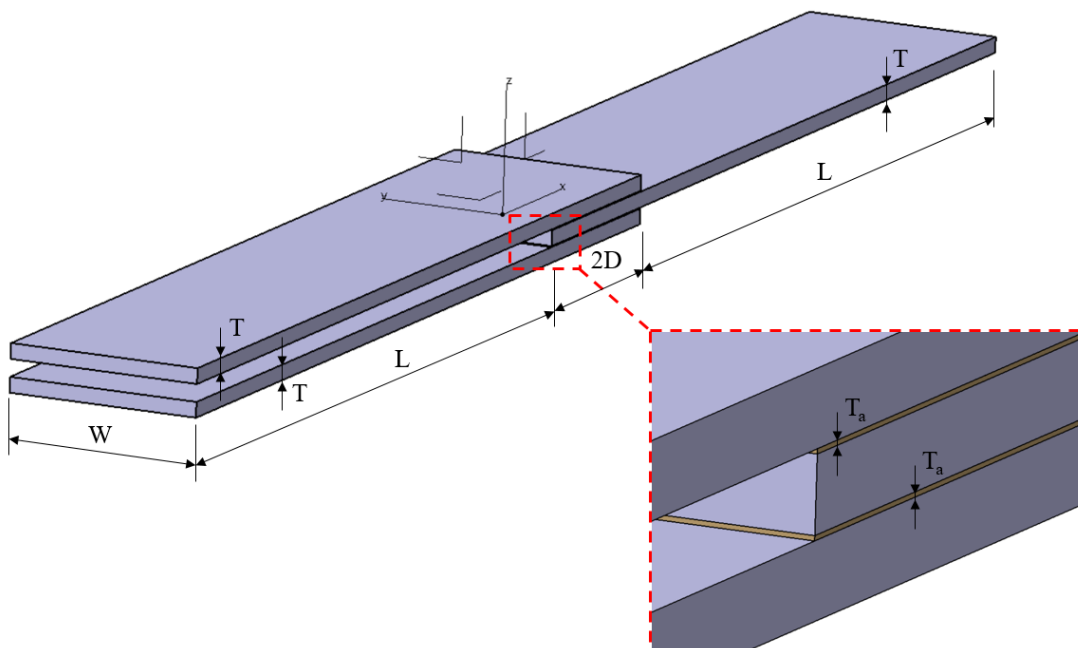


Figure 4.4 : DLJ configuration and dimensions.

The material properties of the adhesive and the composite adherends are listed in Table 4.2. The adhesive is generally considered as isotropic elastic material. The interlaminar properties of composite lamina were given by transverse isotropic assumption or engineering experiences: $E_2=E_3$, $\nu_{12}=\nu_{13}=\nu_{23}$, $G_{12}=G_{13}=G_{23}$ [6]. In the joint symmetrical plane for plies with different angles, the equivalent engineering constants are listed in Table 4.3.

Table 4.1 : Dimension of both SLJ and DLJ.

L (mm)	2D (mm)	W (mm)	T (mm)	T _a (mm)
101.6	25.4	25.4	2	0.13

Table 4.2: Composite adherend and adhesive material properties.

Properties	E ₁ (GPa)	E ₂ (GPa)	G ₁₂ (GPa)	ν_{12}
Adherend	138	9.4	6.7	0.32
Adhesive	E=3 GPa		1.15	0.31

Table 4.3 : Equivalent engineering constants for different plies.

Ply orientation	E _{1'} (GPa)	E _{2'} (GPa)	G _{1'2'} (GPa)	$\nu_{1'2'}$
0°	138.97	10.47	6.7	0.43
±45°	45.32	10.47	5.13	0.38

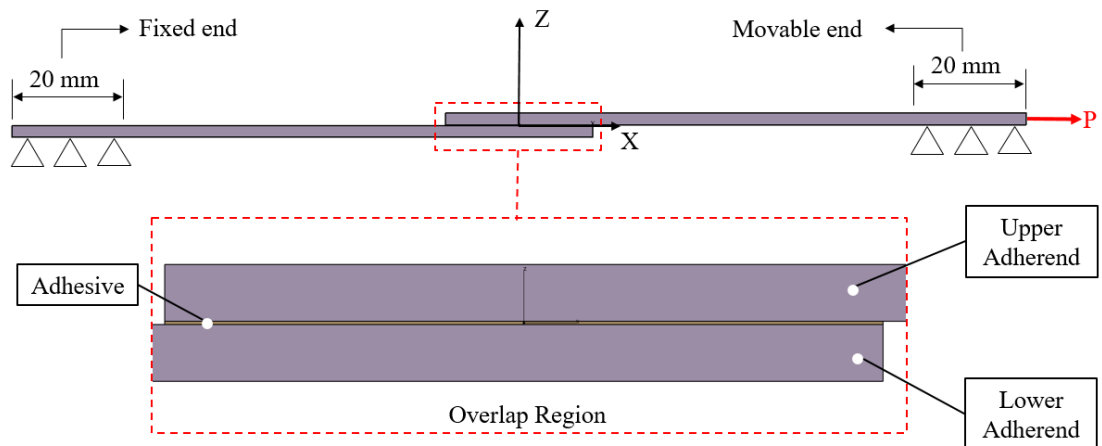


Figure 4.5 : Boundary conditions and overlap region of SLJ configuration.

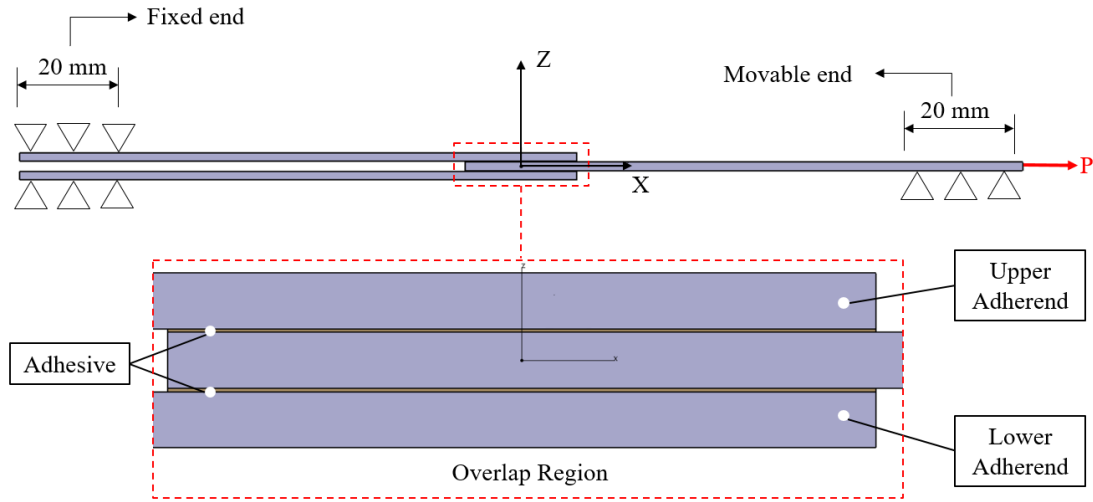


Figure 4.6 : Boundary conditions and overlap region of DLJ configuration.

The adherends for both SLJ and DLJ, 2D FE models with employing the Zhao's material model were generated in this study. The material properties of composite plies with different angles were assigned to related ply section respectively and S4R shell elements were adopted in ABAQUS. The adhesive layer was consisted of 6 C3D8R solid elements through thickness. Different mesh densities were applied in different regions of the both SLJ and DLJ. In particular, potential stress concentration region such as two ends of the adhesive, refined meshes were used and in the other region relatively coarse meshes were generated.

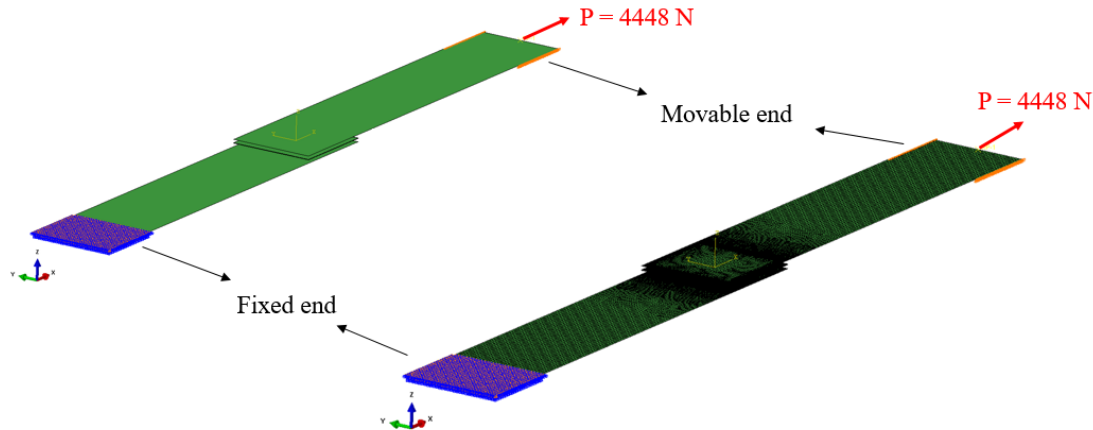


Figure 4.7 : Finite element model and boundary conditions of SLJ.

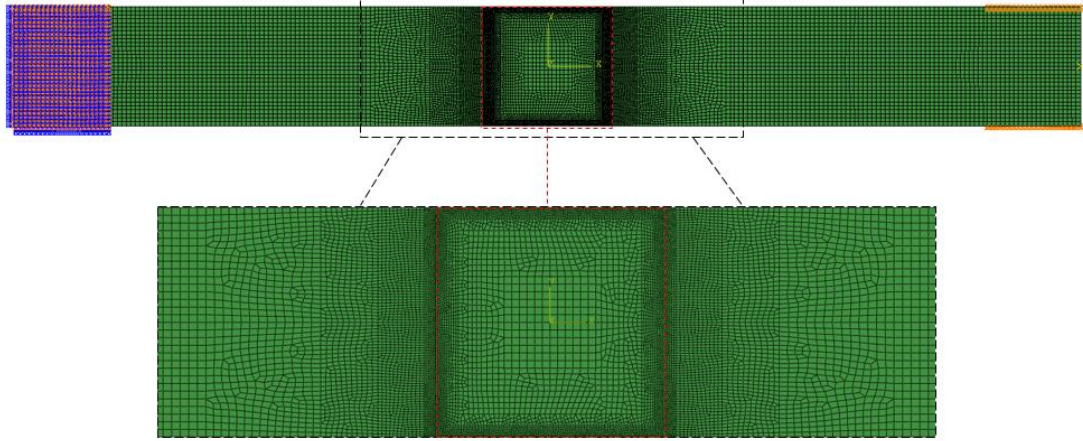


Figure 4.8 : Overlap region and different mesh densities of SLJ.

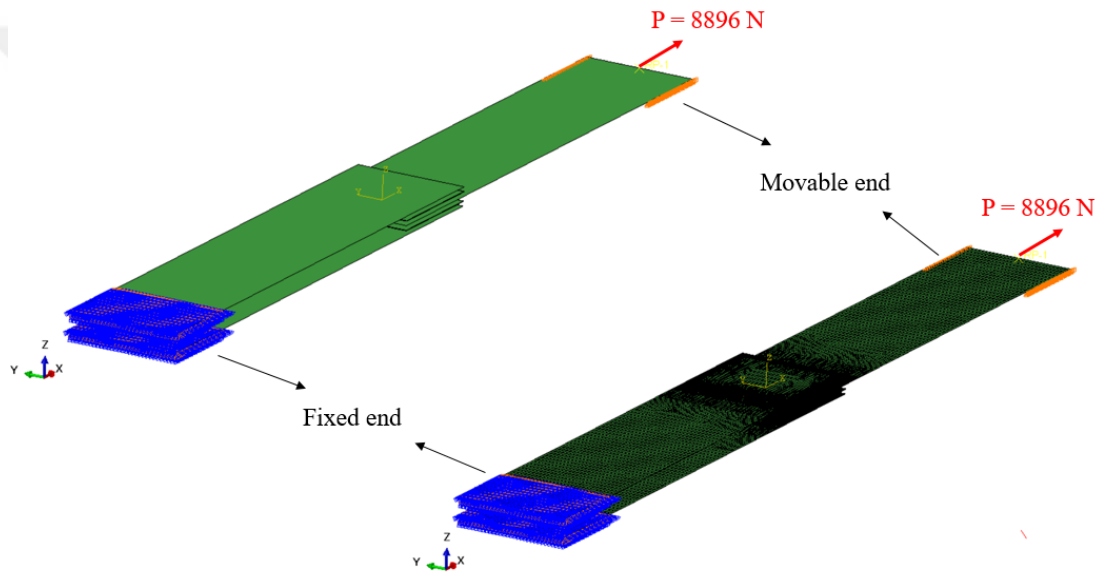


Figure 4.9 : Finite element model and boundary conditions of DLJ.

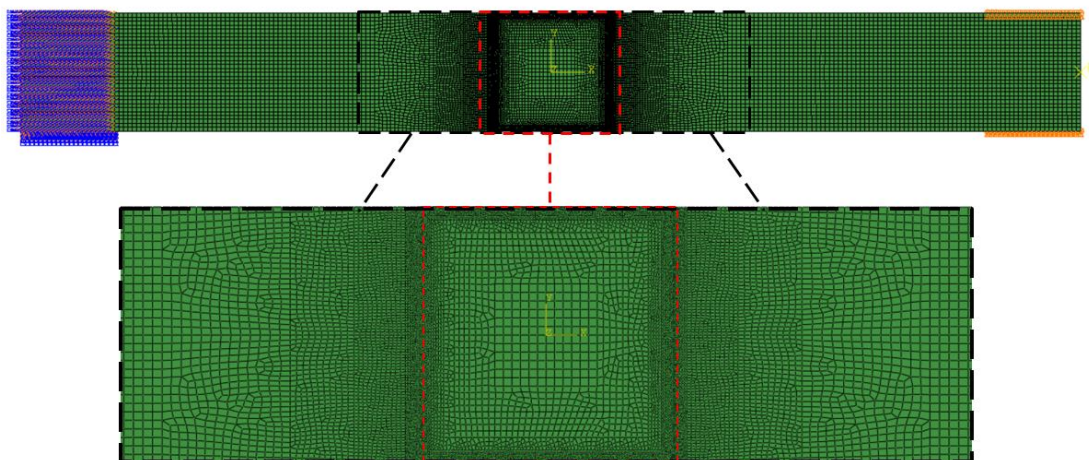


Figure 4.10 : Overlap region and different mesh densities of DLJ.

In order to contain geometric nonlinearity, the 'NLgeom' was set the 'on' state in ABAQUS. The composite plies for adherends were generated by ABAQUS composite layups command and were defined by corresponding fibers directions, ply thickness and material (Figure 4.11).

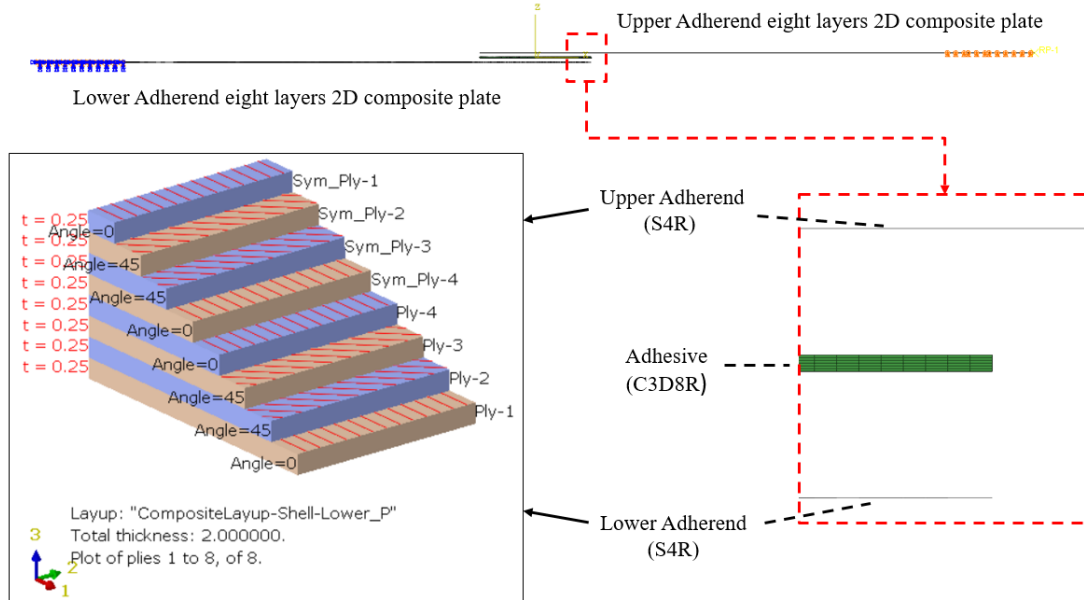


Figure 4.11 : Ply-based visualization of adherends.

In order to realize deformation differences according to joint types, both SLJ and DLJ stress analysis model were generated and stress results will be given in analysis results section.

4.2.2 Pattern and rigidity effects of adhesive FE models for SLJ

Due to simply design and manufacturing, low cost and easy test abilities of the SLJ, too many research and studies have been carried out. Therefore, this study focuses on the SLJ stress analysis, especially aiming to decrease stress concentration of the peak stress locations and to propose more uniform stress in the adhesive. There are five different pattern shapes and a master adhesive shape which is commonly used and has continuous volume, and three different materials for adhesive is used. Therefore, six different adhesive models are defined totally. All different adhesive cad models and dimensions of a single cell are shown in Figure 4.12 and Figure 4.13, respectively. The single cell dimensions are listed in Table 4.4. Except adhesives, each models have same adherends geometry, material and boundary conditions. Therefore, the main structures of the models are same previous SLJ and the main dimensions such as

adherends length, thickness and width, ply angles and layers numbers, adhesive thickness and boundary conditions. Thereby, Figure 4.3 and 4.5 can be indicated as model dimensions and overlap region and Figure 4.7 and 4.8 can be indicated as finite element model and boundary conditions.

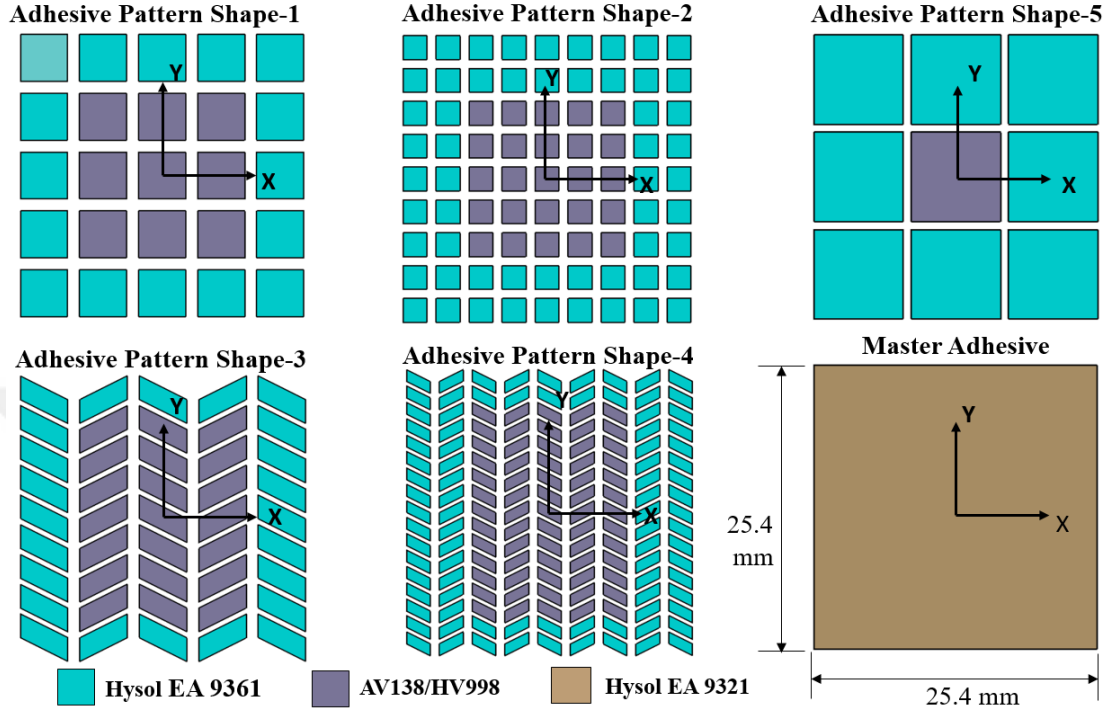


Figure 4.12 : Adhesive pattern shape configurations.

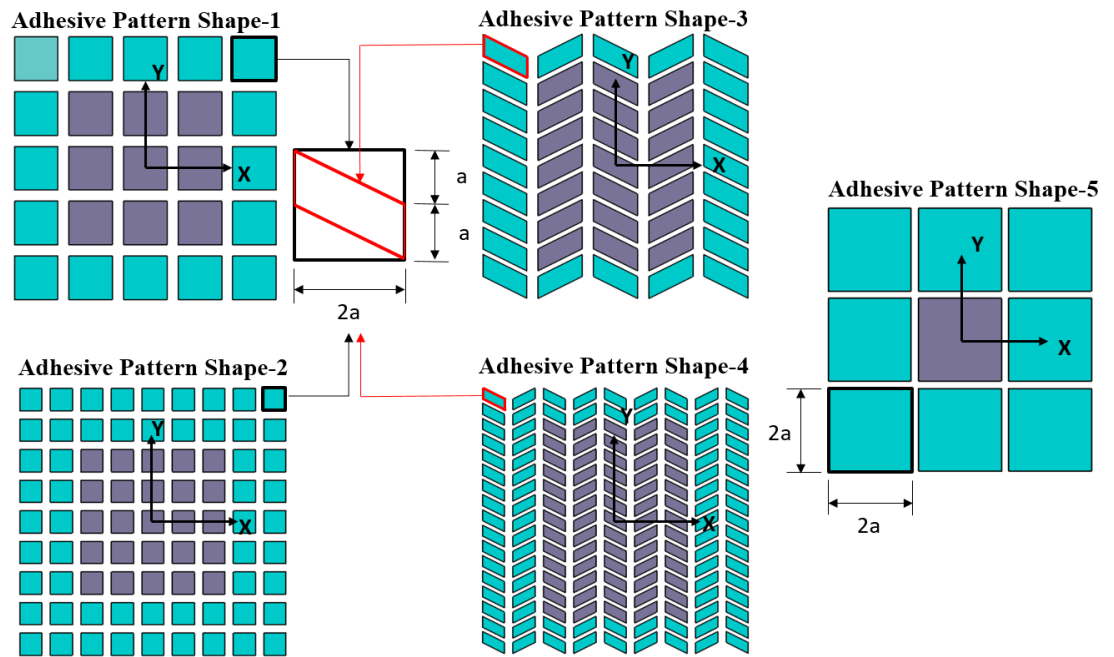


Figure 4.13 : The single cell dimensions of adhesive pattern shape configurations.

Table 4.4 : The single cell dimensions of adhesive pattern shapes.

Adhesives Models	a (mm)
Adhesive Pattern Shape-1	25.4/12
Adhesive Pattern Shape-2	25.4/12
Adhesive Pattern Shape-3	25.4/24
Adhesive Pattern Shape-4	25.4/24
Adhesive Pattern Shape-5	4

In order to reduce stress concentration and provide better uniform stress on the adhesive, mixed-adhesive joints are proposed. Mixed-adhesive joint is consisted of at least two different adhesives, one has low elastic modulus and the other has high elastic modulus. Generally, the low elastic modulus or low strength adhesive to be placed at the outer region of the overlap and acting as decreasing stress levels. On the other hand, the high elastic modulus or high strength adhesive is commonly located at the central region of the overlap, being far from the stress concentrations regions such as overlap ends. Three types of adhesives (ductile, intermediate and brittle) were selected whose material properties are listed in Table 4.5 [36].

Table 4.5 : Material of adhesives.

Adhesives	E_1 (GPa)	G_{12} (GPa)	ν
Hysol EA 9361	0.67	0.24	0.4
Hysol EA 9321	3.87	1.42	0.36
AV138/HV998	4.59	1.71	0.34

Furthermore, all different adhesive mesh models are shown in Figure 4.14. In order to reveal mixed adhesives and pattern shapes effects, overlap regions mesh for all adhesive models are generated specific elements length, which is 0.2646 mm. Therefore, the mesh size effects are minimized for all models. The stress results will be given in analysis results section.

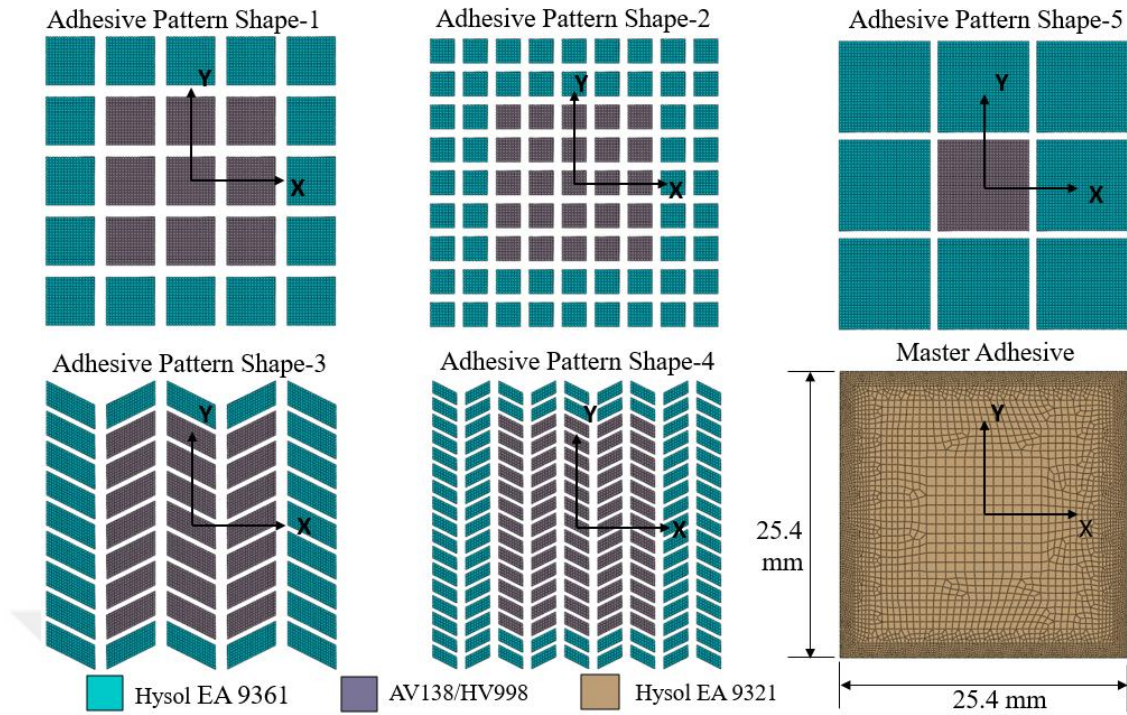


Figure 4.14 : Adhesive pattern shape FE models.

4.2.3 Spew fillet SLJ and tapered SLJ FE models

The main objective of this chapter is to investigate adhesive stress distribution with effects of tapered end adherends and with effects of the spew fillets. For this purpose, two models, one is tapered end adherends model and the other is spew fillets model, are generated. The general dimensions, adherends layers, fibers angles, materials and boundary conditions of models are same with previous SLJ model. The adhesive mesh layer for both model in overlap region are divided into 4 layers through the thickness of the adhesives. The material properties of the adhesives and the composite adherends for both models are listed in Table 4.2 and Table 4.3, which were used in the previous SLJ model. Likewise previous SLJ model, the same assumptions and steps are considered in here. The adhesive is generally considered as isotropic elastic material. The interlaminar properties of composite lamina were given by transverse isotropic assumption or engineering experiences: $E_2=E_3$, $\nu_{12}=\nu_{13}=\nu_{23}$, $G_{12}=G_{13}=G_{23}$ [6]. However, for tapered end adherends, only the end of the adherends geometries are changed and for spew fillet models, only the ends of the adhesive geometries are changed. Both models dimensions are shown in Figure 4.15 and Figure 4.16, and are listed in Table 4.6, and Table 4.7, respectively.

Table 4.6 : Tapered SLJ dimensions.

L (mm)	2D (mm)	W (mm)	T (mm)	T _a (mm)	T _b (mm)	α°
101.6	25.4	25.4	2	0.13	0.435	30°

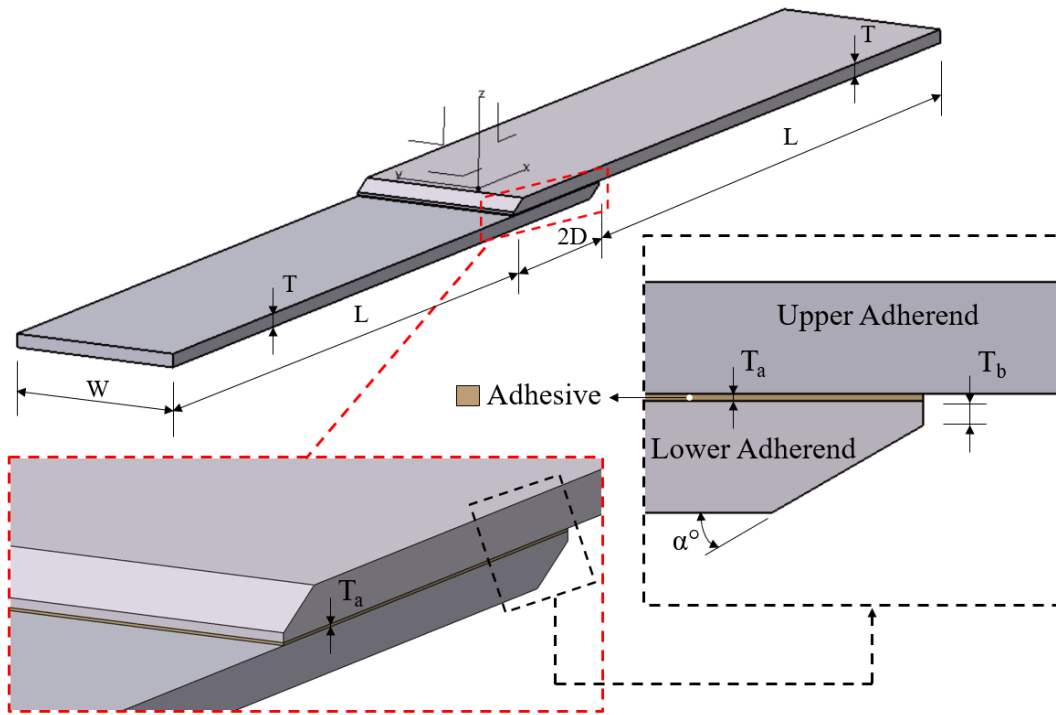


Figure 4.15 : Tapered SLJ configuration and dimensions

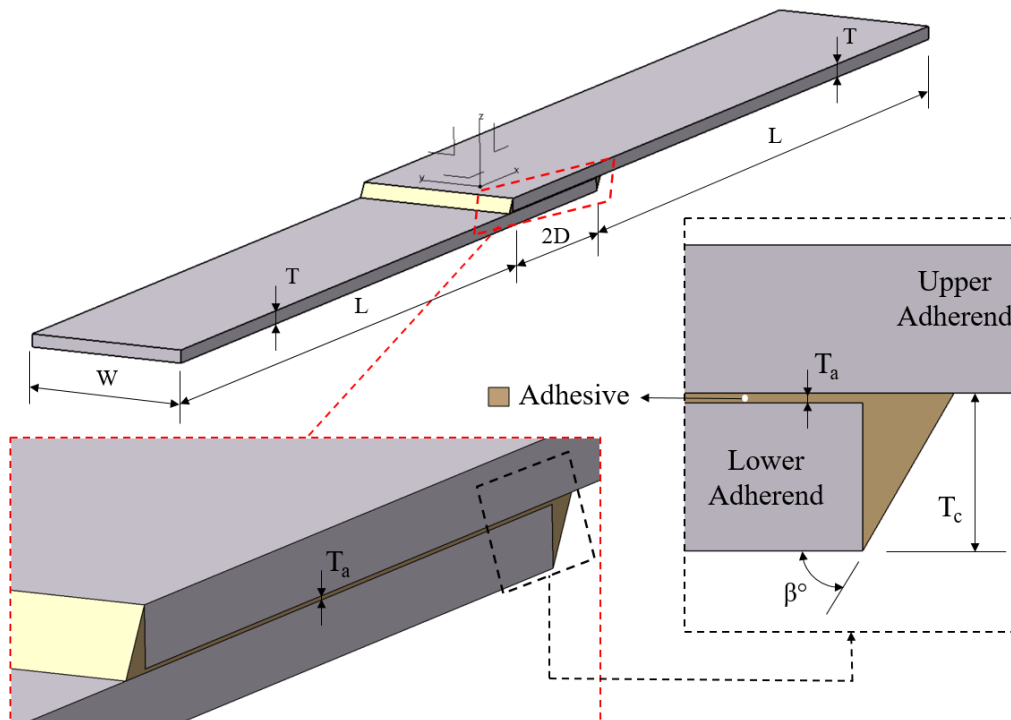


Figure 4.16 : Spew fillet SLJ configuration and dimensions.

Table 4.7 : Spew fillet SLJ dimensions.

L (mm)	2D (mm)	W (mm)	T (mm)	T _a (mm)	T _c (mm)	β°
101.6	25.4	25.4	2	0.13	2.13	60°

The adherends are generated by using continuum shell elements in ABAQUS, due to modelling the thickness of the adherends. Continuum shell model in ABAQUS provides 3D modelling with low computing cost and effective to obtain accurate stress distributions. The FE model of tapered and spew fillet SLJ are shown in Figure 4.17 and 4.18, respectively.

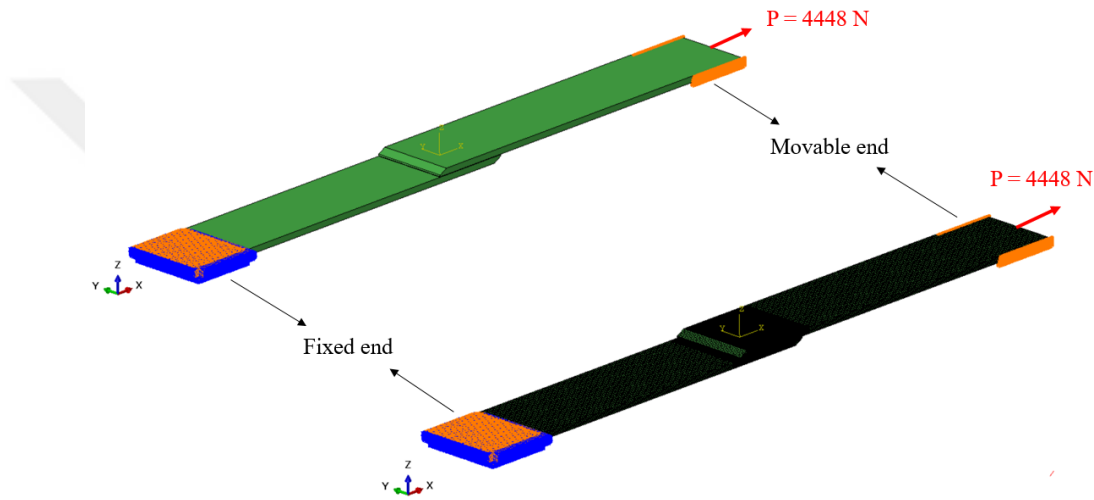


Figure 4.17 : Finite element model and boundary conditions of tapered SLJ.

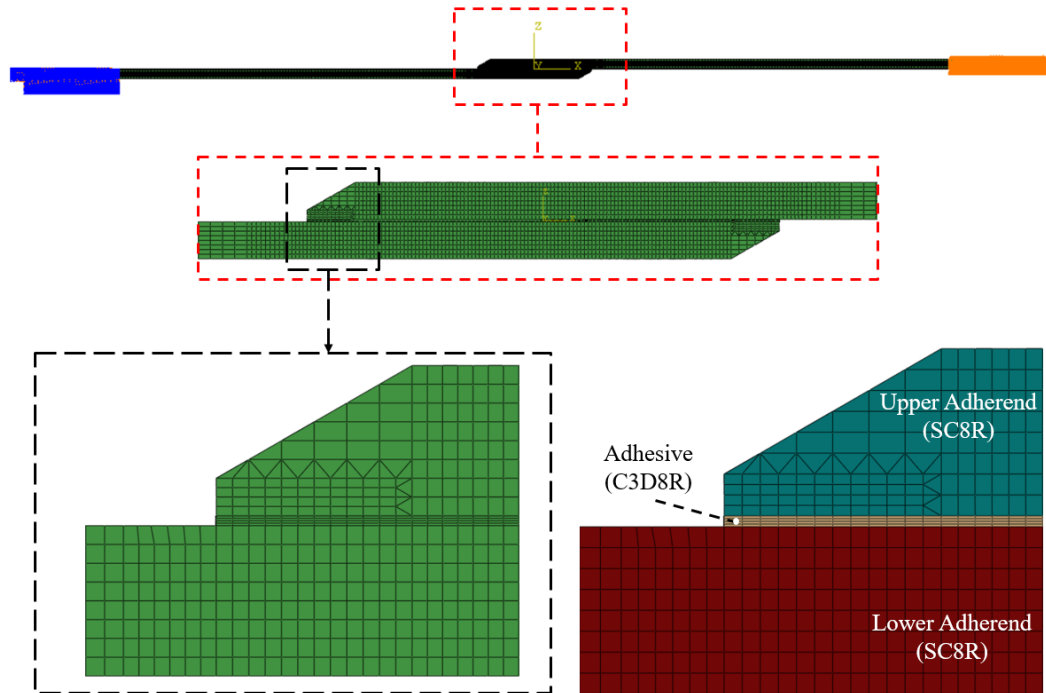


Figure 4.18 : Overlap region, different mesh densities, and parts of the tapered SLJ.

The spew fillet SLJ model is same boundary conditions and hence, only the parts of the spew fillet SLJ model is shown in Figure 4.19.

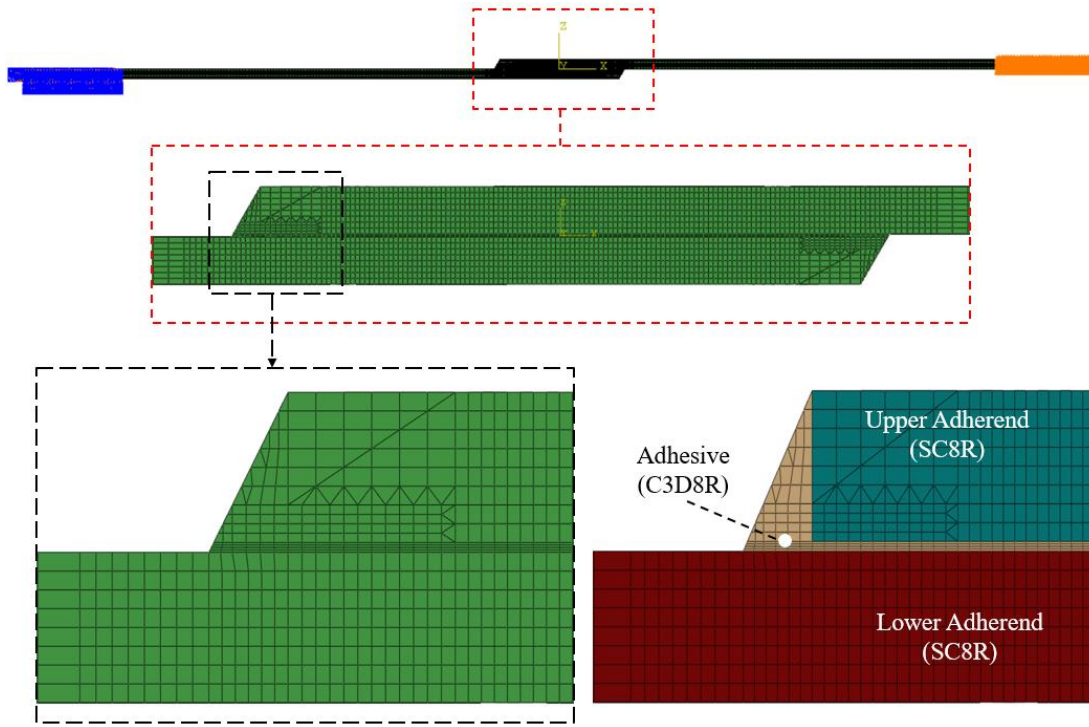


Figure 4.19 : Overlap region, different mesh densities, and parts of the spew fillet SLJ.

It can be realized that spew fillet mesh model is generated by using tapered SLJ model, and hence the mesh size effects are minimized for all models. The stress results will be given in analysis results section.

4.2.4 The fibers direction angles effects on the stress of the adhesive FE models for SLJ

In order to examine fiber directions effects on the adhesive, a great numbers of SLJ FE models were generated, analyzed, and revealed and read results with the help of several codes which are performed by MATLAB and Python. The Matlab is dedicated to mathematical and technical computing. In contrast, Python is a general-purpose programming language.

The main structures of the models are same previous SLJ except only fibers directions angles. Thereby, Figure 4.3 and 4.5 can be indicated as model dimensions and overlap region and Figure 4.7 and 4.8 can be indicated as finite element model and boundary conditions.

5. ANALYSIS RESULTS AND DISCUSSIONS

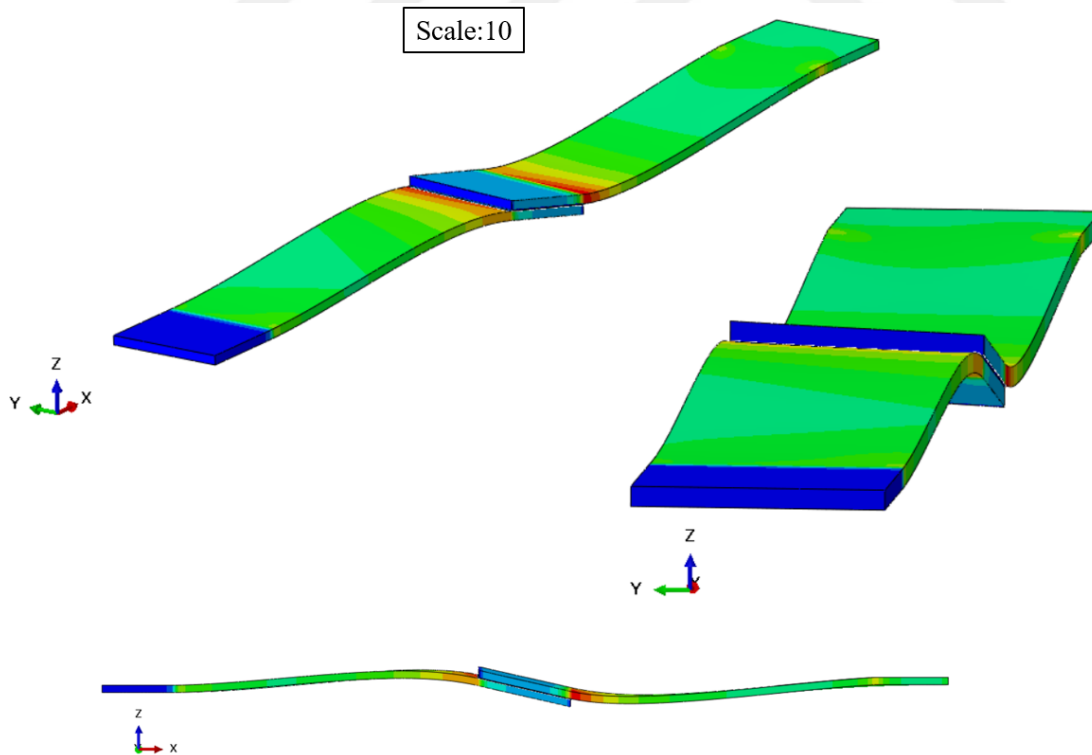
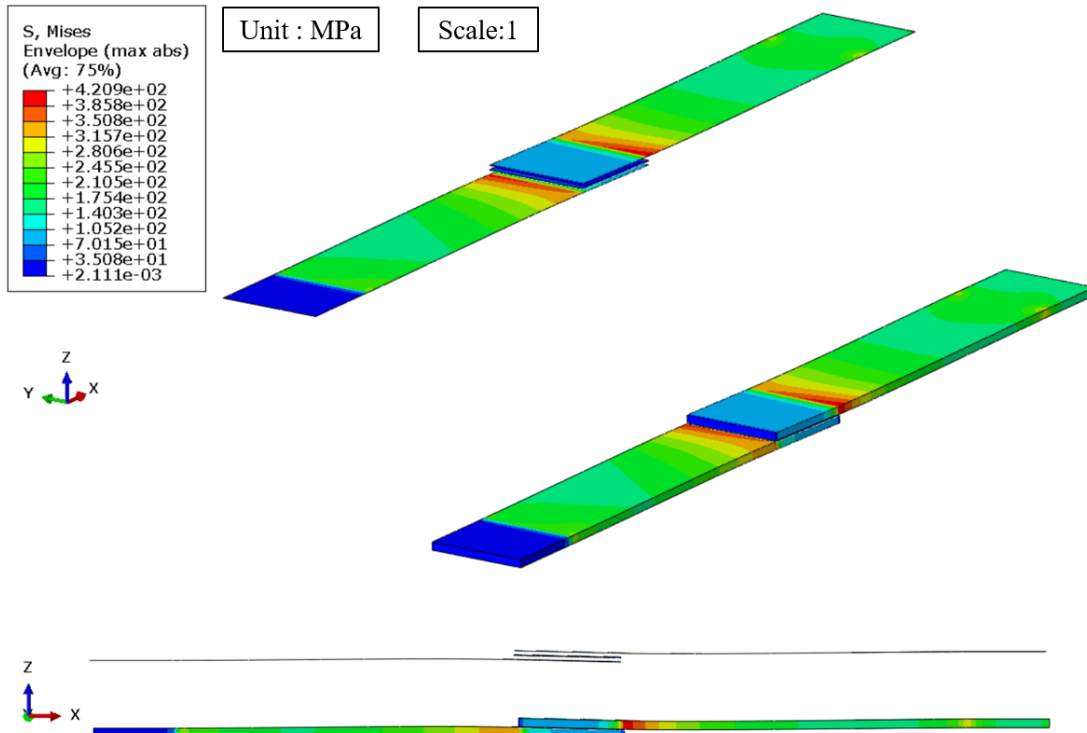
Basically, there are four different analysis cases which are examined in details. These cases were introduced at chapter 4 and it can be given below as a summary.

- 1) Single lap joints and double lap joints stress analysis,
- 2) Revealing pattern and rigidity effects of the adhesives for single lap joints,
- 3) The tapered adherends, and using spew fillets effect on the stress distribution of the adhesive for SLJ,
- 4) The fibers direction angles effects on the stress concentration of the adhesive for SLJ.

In summary, while investigating adhesives stress distribution, direct stresses acting normal to the adhesive layer, that is, peel stresses occur. In practice, due to occurring the adherends deformation that means they are not rigid, as the load transfers from the other adherend, a non-uniform shear stress distribution is occurred and the maximum stress for both peel and shear stress are observed both ends of the adhesive in all models. For the adhesively bonded joint, the peel stress and shear stress in the adhesive, which always lead to adhesive failure, and play an important role in estimating joint strength.

5.1 Single and Double Lap Joints Stress Results

The main purpose of these analyses is to reveal stress and deformations differences of the single lap and double lap joints and comparison of the stress distributions. The SLJ stress analysis results are presented in assembly level and adhesive level. In order to comprehend the behaviour of the SLJ under tensile load, stress results in assembly level are given as both scale factor 1 and 10, in Figure 5.1 and 5.2, respectively. Both maximum peel and shear stresses are observed both ends of the adhesive. As stated previously, the adhesive layer was divided into six elements through thickness. The shear and peel stress distributions in the middle of adhesive layer are obtained and compared with the corresponding DLJ stress results through the finite element analysis.



The stress distribution of the adhesive is investigated, especially peel stress and shear stress, both are indicated together in Figure 5.3. It can be realised that due to single lap

joints unbalanced characteristics, stress concentration and deformation levels are severe according to other adhesively bonded joint types. Because the force is applied through the adhesive to the adherends away from their neutral axes.

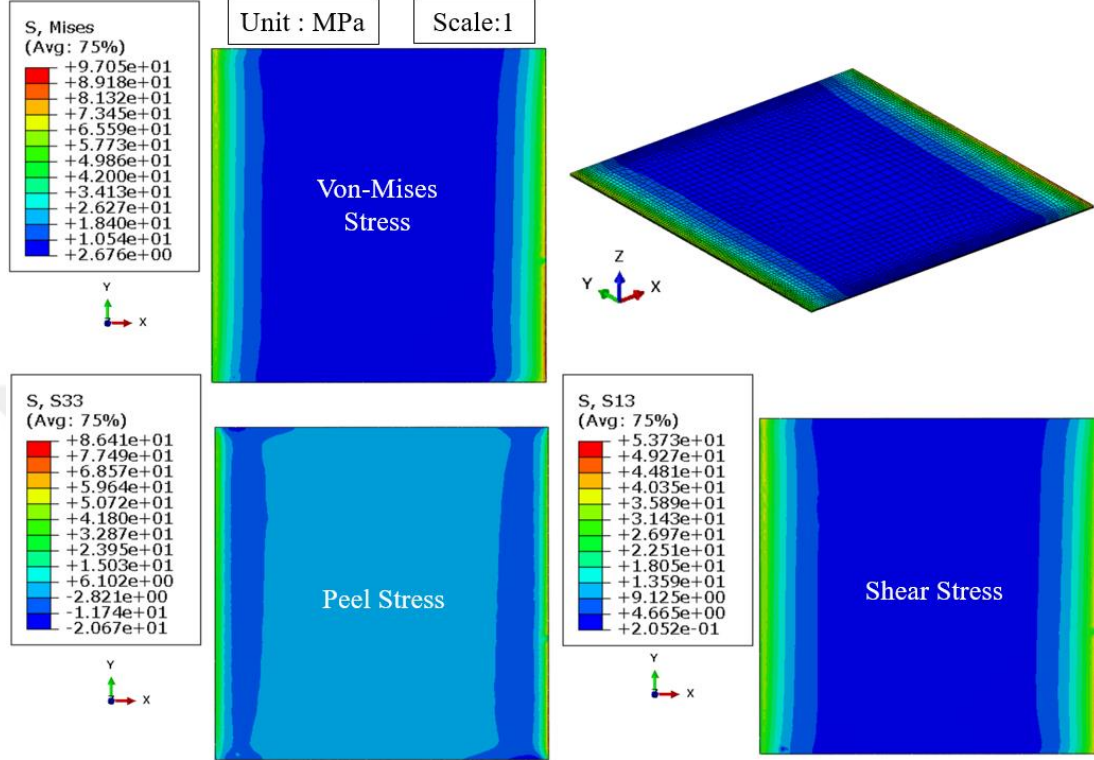


Figure 5.3 : Stress results and distributions of the adhesive in SLJ.

In order to obtain a better understanding, the stress levels on the adhesive can be given as graph. To avoid FE based misdirection due to contact region, the middle mesh layer is considered and hence, upper three mesh layers of the adhesive are omitted in order to obtain the middle of the adhesive layer stresses. Then, center line of the middle mesh layer's elements are selected through the adhesive and plotted as peel and shear stresses via ABAQUS. All steps are shown in Figure 5.4 and Figure 5.5.

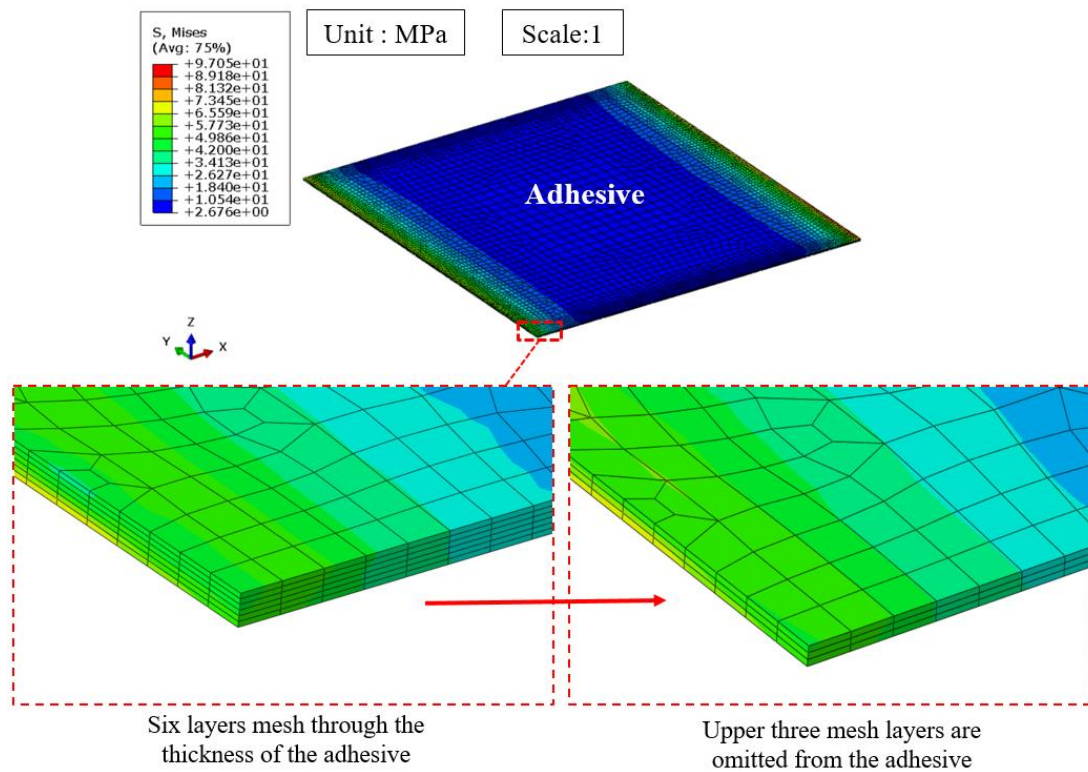


Figure 5.4 : The removal of the upper three mesh layers.

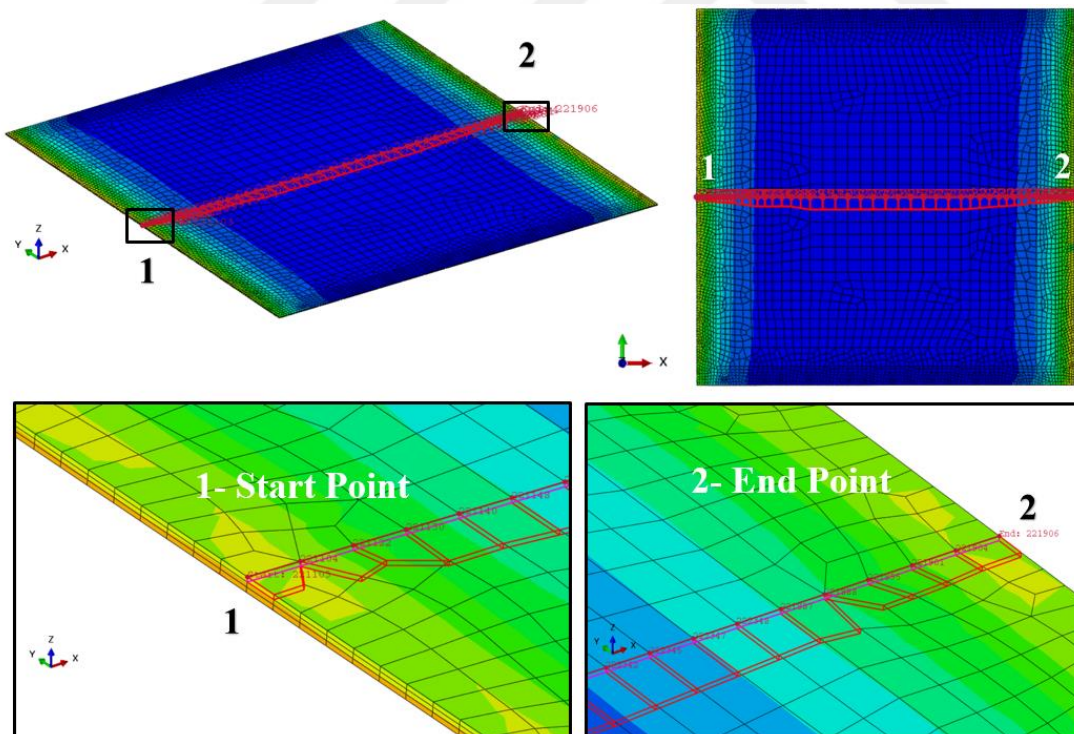


Figure 5.5 : The reading stresses path of the SLJ model.

In Figure 5.6, it can be seen easily, the stress concentration regions are both ends of the overlap region. For both peel and shear stresses of the overlap middle region are very low according to the both ends of the overlap regions.

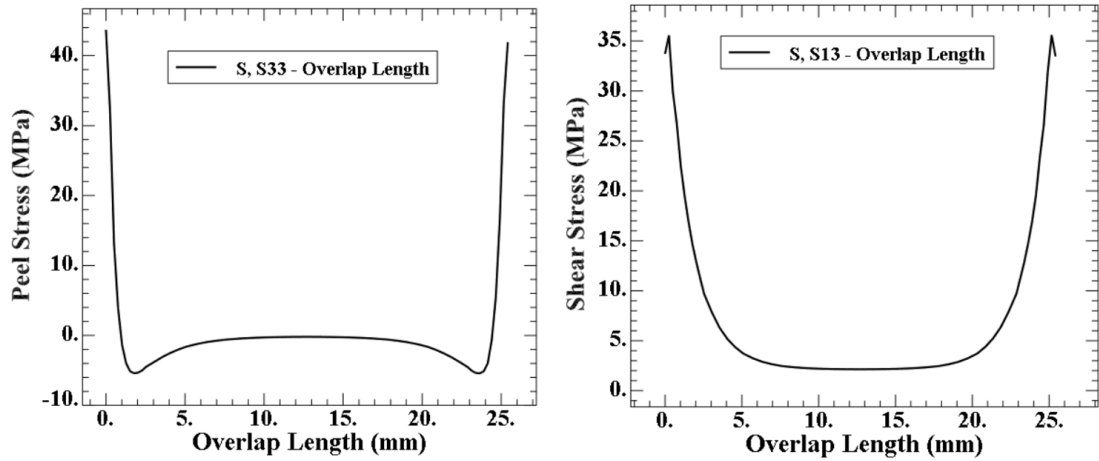


Figure 5.6 : Peel and shear stresses along the centerline of the adhesive.

The DLJ stress analysis results will be given as same steps of the SLJ. The DLJ stress analysis results are presented in assembly level and adhesive level. In order to comprehend the behaviour of the DLJ under tensile load, stress results in assembly level are given as both scale factor 1 and 10, in Figure 5.7 and 5.8, respectively. Both maximum peel and shear stresses are observed both ends of the adhesive.

It can be shown that stress and deformation levels are lower than SLJ's stress and deformations level, due to having more balanced structure. Even if scale factor is increased 10, the deformations are not easily realised in Figure 5.8.

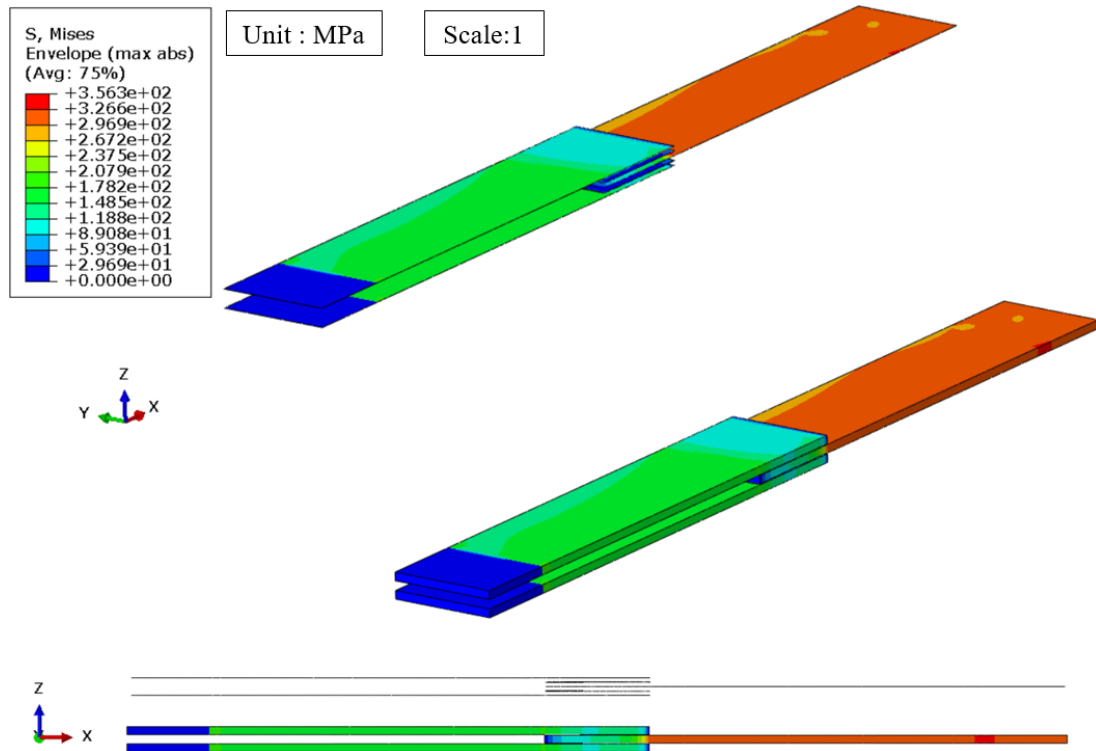


Figure 5.7 : Stress results of the DLJ assembly in scale 1.

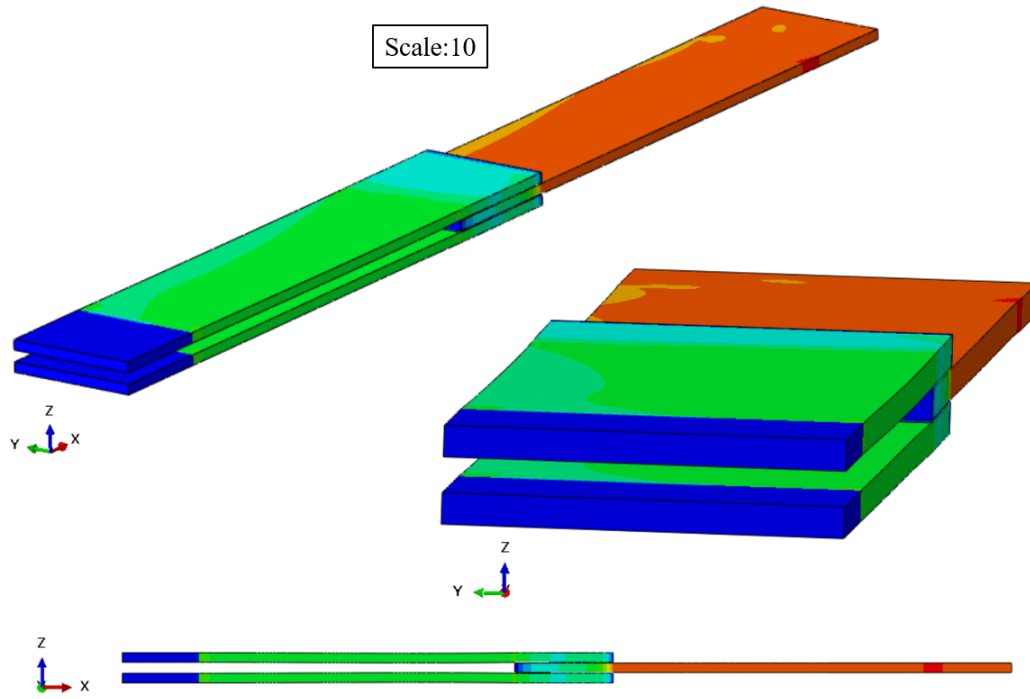


Figure 5.8 : Stress results of the SLJ assembly in scale 10.

The stress distribution of the adhesive is investigated, especially peel stress and shear stress, both are indicated together in Figure 5.9. Likewise assembly stress levels, the stress levels on the adhesive are low according to SLJ's stress levels on the adhesive owing to double lap joints balanced characteristics. Due to FE singularities, the stress level limits are reduced for von-Mises stress, peel stress, and shear stress.

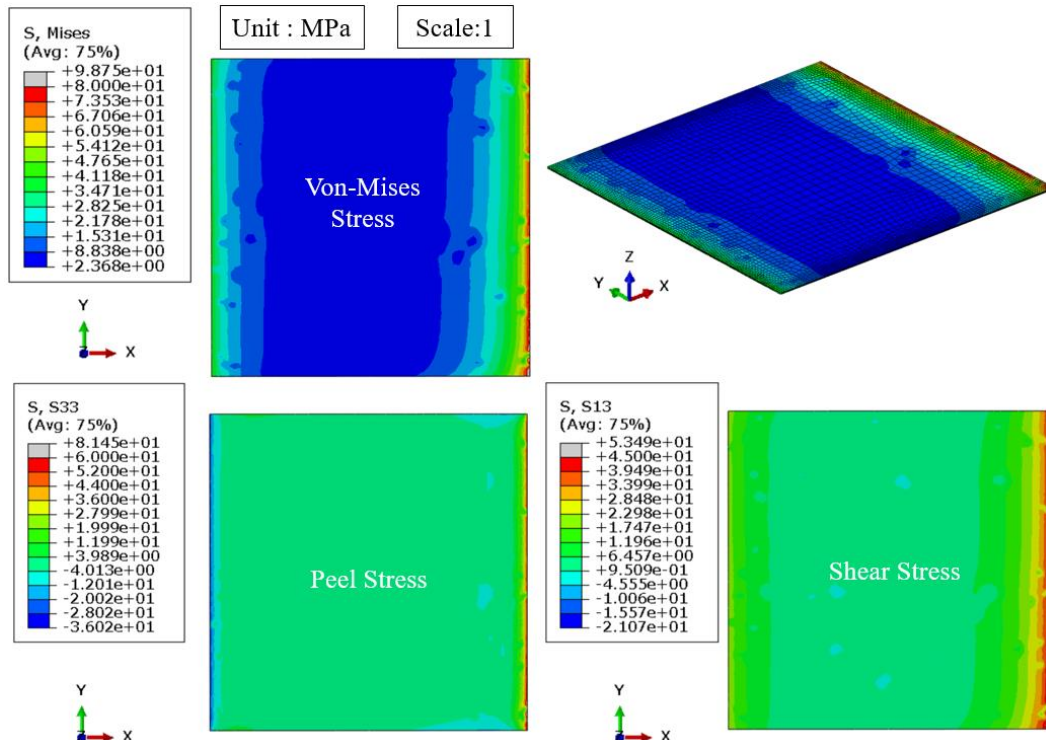


Figure 5.9 : Stress results and distributions of the lower adhesive in DLJ.

In order to obtain a better understanding, the stress levels on the adhesive can be given as graph, likewise it was done for SLJ adhesive stresses. It should be realized that there are two adhesives in the double lap joints, lower adhesive and upper adhesive. Therefore, only the lower adhesive is investigated. Furthermore, it was examined that both adhesives have nearly same behavior and stress levels. To avoid FE based misdirection due to contact region, the middle mesh layer is considered and hence, upper three mesh layers of the adhesive are omitted in order to obtain the middle of the adhesive layer stresses. Then, center line of the middle mesh layer's elements are selected through the adhesive and plotted as peel and shear stresses via ABAQUS. All steps are same with the SLJ and hence, it can be indicated in Figure 5.4 and Figure 5.5.

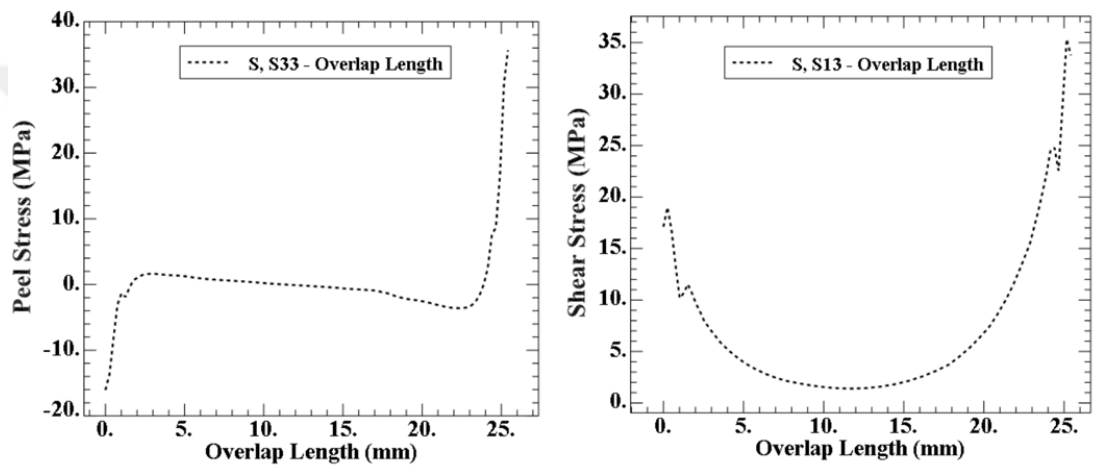


Figure 5.10 : Peel and shear stresses along the centerline of the adhesive.

In Figure 5.10, it can be seen easily, the stress concentration regions are both ends of the overlap region, but especially end of the applied load side. Due to balanced characteristic of DLJ, the both stress levels are low in the constrained side of the overlap end. Furthermore, the peel stress in the constrained side of the overlap end occurs as compressive stress, a certain level of compressive stress can be increased durability of the joints in some cases. For both peel and shear stresses of the overlap middle region are very low according to the both ends of the overlap regions.

The peel and shear stresses comparison graph of the adhesives for both SLJ and DLJ can be seen in Figure 5.11. It can be seen that due to balanced structures of the DLJ, which is more durable than SLJ.

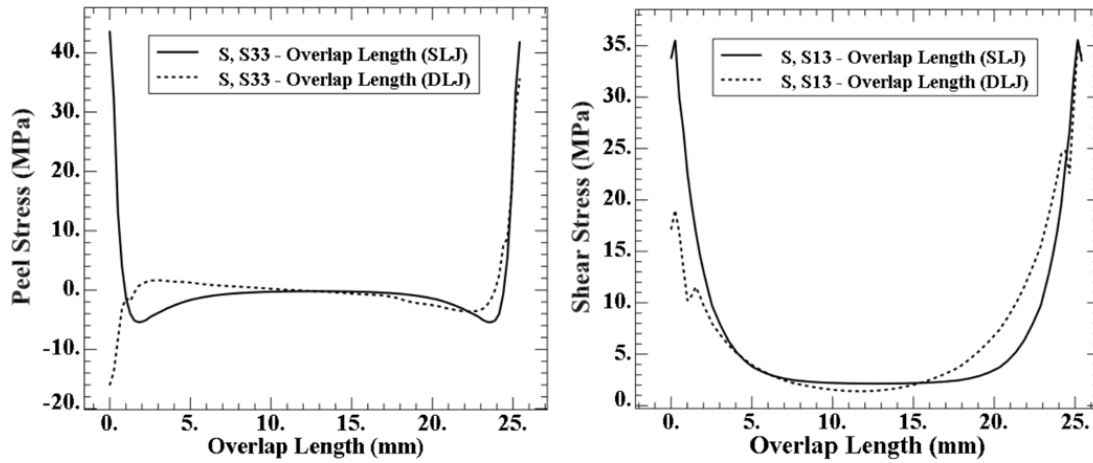


Figure 5.11 : The peel and shear stresses comparison of SLJ and DLJ along the centerline of adhesives.

5.2 Pattern and Rigidity Effects of Adhesive FE Models Stress Results for SLJ

The main objective of this study is to obtain low stress levels especially, ends of the overlap region, which are considered stress concentration region of the adhesive, and more uniform stress distribution on the adhesive. Previous study revealed that SLJ is more delicate than other types of bonded joints, and there have been much more researches and studies on the SLJ. Therefore, SLJ is based on the pattern and rigidity effects of the adhesive.

As stated previously, there are five mixed adhesives and a master adhesive model which is a conventional adhesive. Peel and shear stresses of adhesives are investigated and compared each other and master adhesive model. The whole models' stress distributions on the centerline of the middle layer of the adhesive are presented via graphs and 3D stress distributions on the middle layer surface of the adhesives are also presented by the help of MATLAB.

The stress results reading will not be mentioned in this chapter. Because in previous chapter which explained in details of stresses analysis and results reading steps. Thereby, it is not necessary to mention all steps again. The stress results of mixed adhesives and master adhesive will be presented in this chapter. The master adhesive model's analysis results will be given first, and then adhesive pattern shape-1, 2, 3, 4, and 5 will be given respectively.

The stress distribution of the master adhesive model is investigated in terms of von-Mises stress, peel stress and shear stress, all are indicated together in Figure 5.12. Likewise previous SLJ stress results, the stress concentrations on the adhesive occur on the both ends of the adhesive owing to single lap joints unbalanced characteristics.

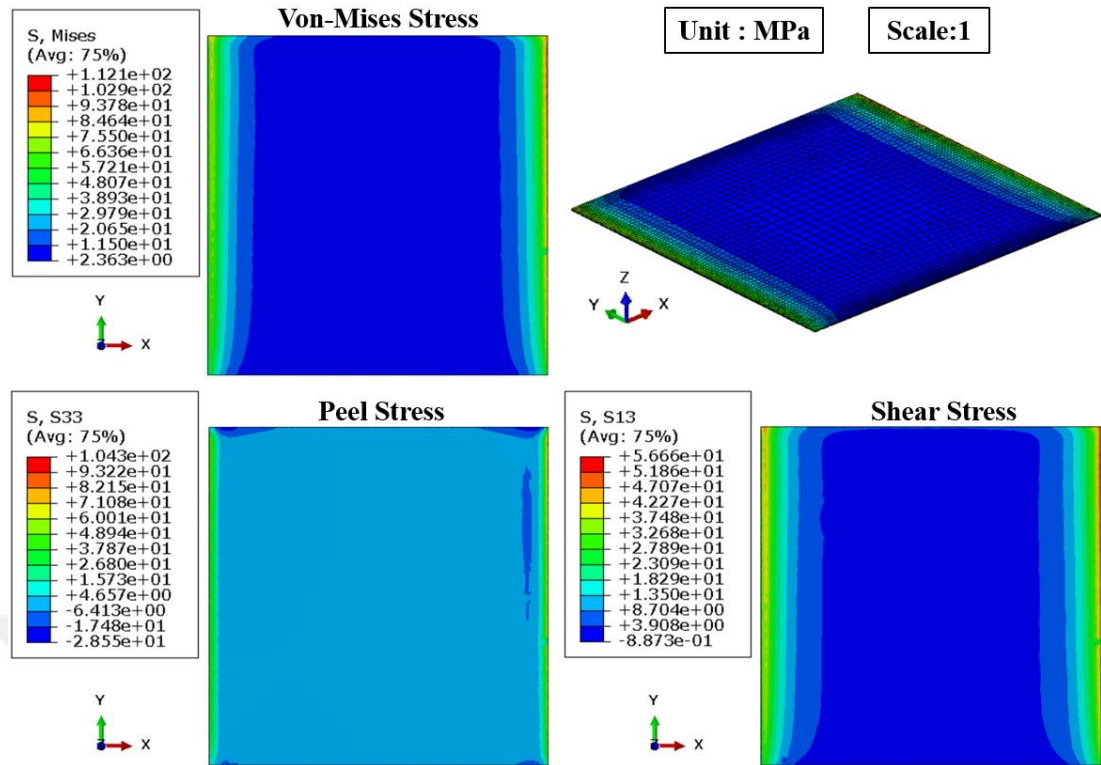


Figure 5.12 : Stress results and distributions of the master adhesive model.

The 3D peel and shear stress distributions on the adhesive for master adhesive model are given in Figure 5.13 and Figure 5.14, respectively.

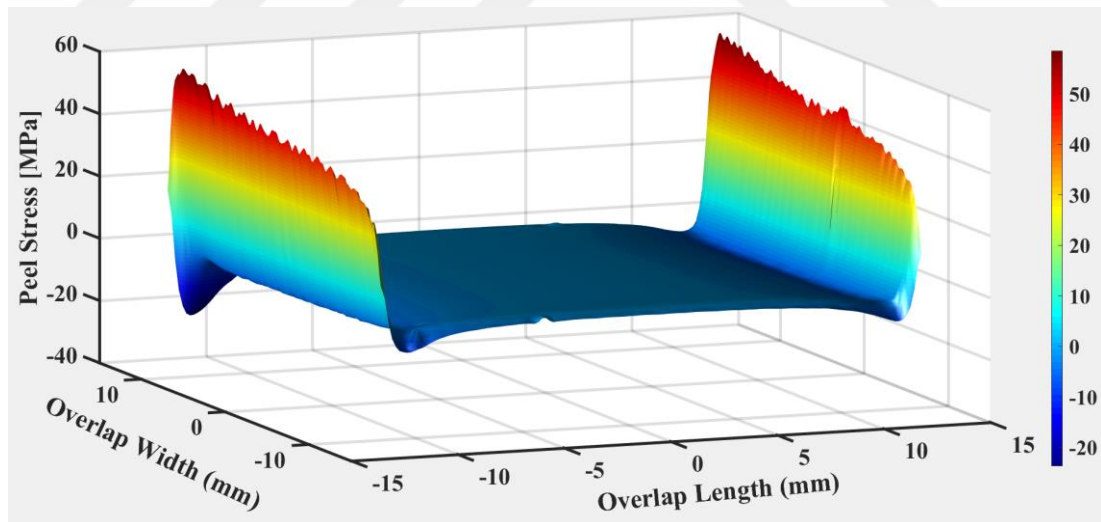


Figure 5.13 : 3D peel stress distribution on the adhesive for master adhesive model.

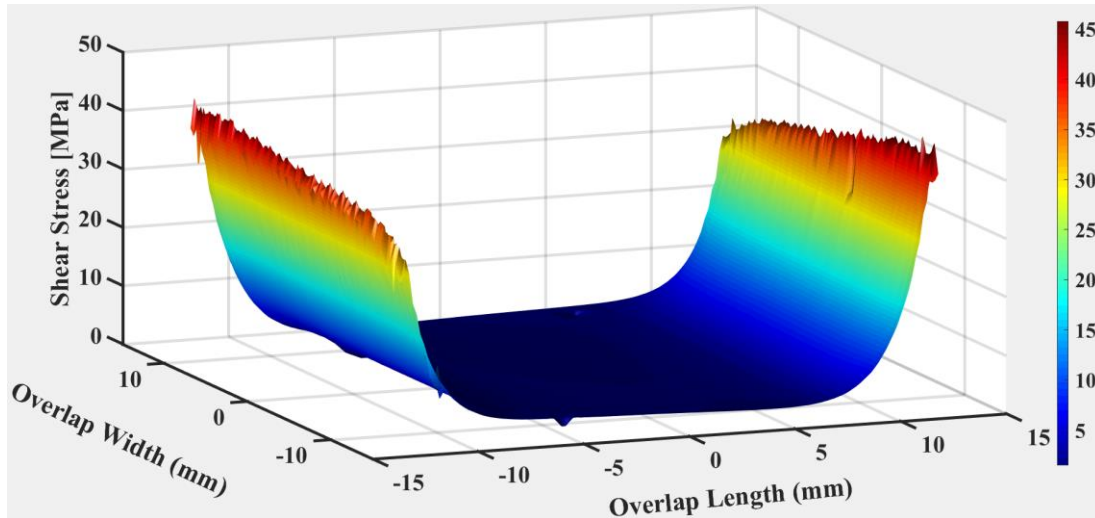


Figure 5.14 : 3D shear stress distribution on the adhesive for master adhesive model.

The stress distribution of the adhesive pattern shape-1 model is investigated in terms of von-Mises stress, peel stress and shear stress, all are indicated together in Figure 5.15. It can be realized that the stress levels of all types of stress are decreased significantly. The von-Mises, peel and shear stresses results are reduced from 112 MPa to 66 MPa, from 104 MPa to 38 MPa and from 57 MPa to 37 MPa, respectively according to master adhesive model. It can be also realized that the stress distribution profiles are changed for all stresses, due to consisting of different stiffnesses of adhesives and pattern shapes.

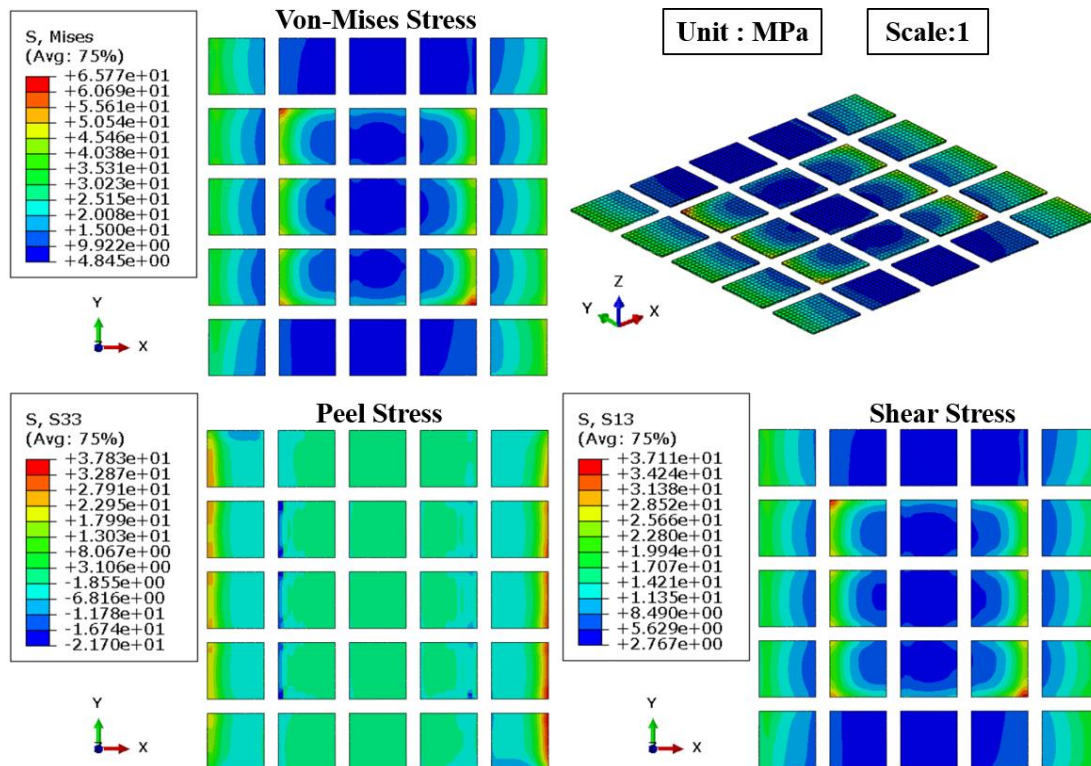


Figure 5.15 : Stress results and distributions of the adhesive pattern shape-1 model.

The 3D peel stress and the 3D shear stress distributions on the adhesive for adhesive pattern shape-1 model are given in Figure 5.16 and Figure 5.17, respectively.

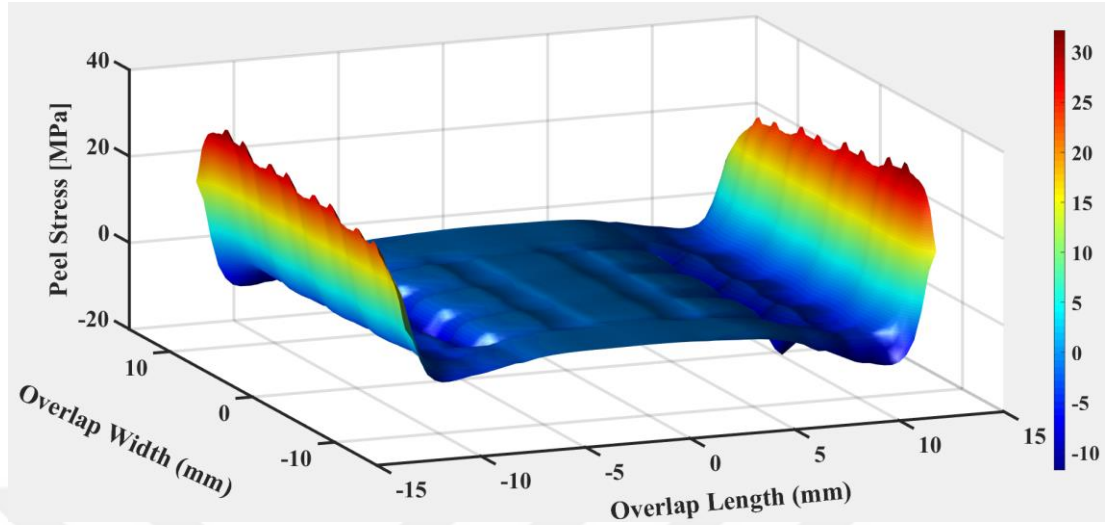


Figure 5.16 : 3D peel stress distribution on the adhesive for adhesive pattern shape-1 model.

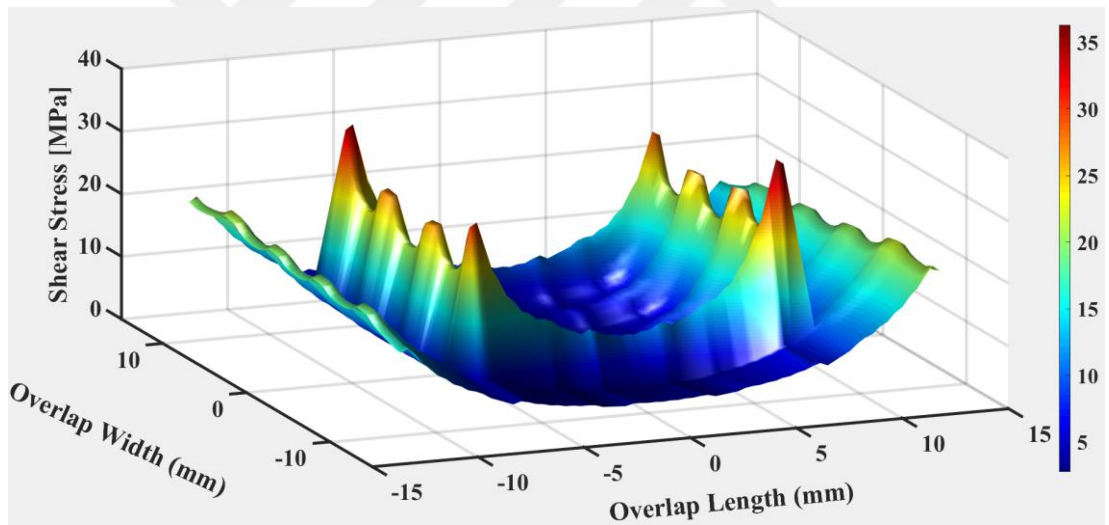


Figure 5.17 : 3D shear stress distribution on the adhesive for adhesive pattern shape-1 model.

The stress distribution of the adhesive pattern shape-2 model is investigated in terms of von-Mises stress, peel stress and shear stress, all are indicated together in Figure 5.18. It can be realized that the stress levels of all types of stress are decreased significantly according to master adhesive model. The results of von-Mises, peel and shear stresses are reduced from 112 MPa to 72 MPa, from 104 MPa to 40 MPa and from 57 MPa to 40 MPa, respectively according to master adhesive model. It can be also realized that the stress distribution profiles are changed for all stresses, due to

consisting of different stiffnesses of adhesives and pattern shapes, likewise adhesive pattern shape-1. Even if stress distributions profiles seem similar, the whole stress levels of adhesive pattern shape-2 are a little higher than the adhesive pattern shape-1 because of the different size of cells.

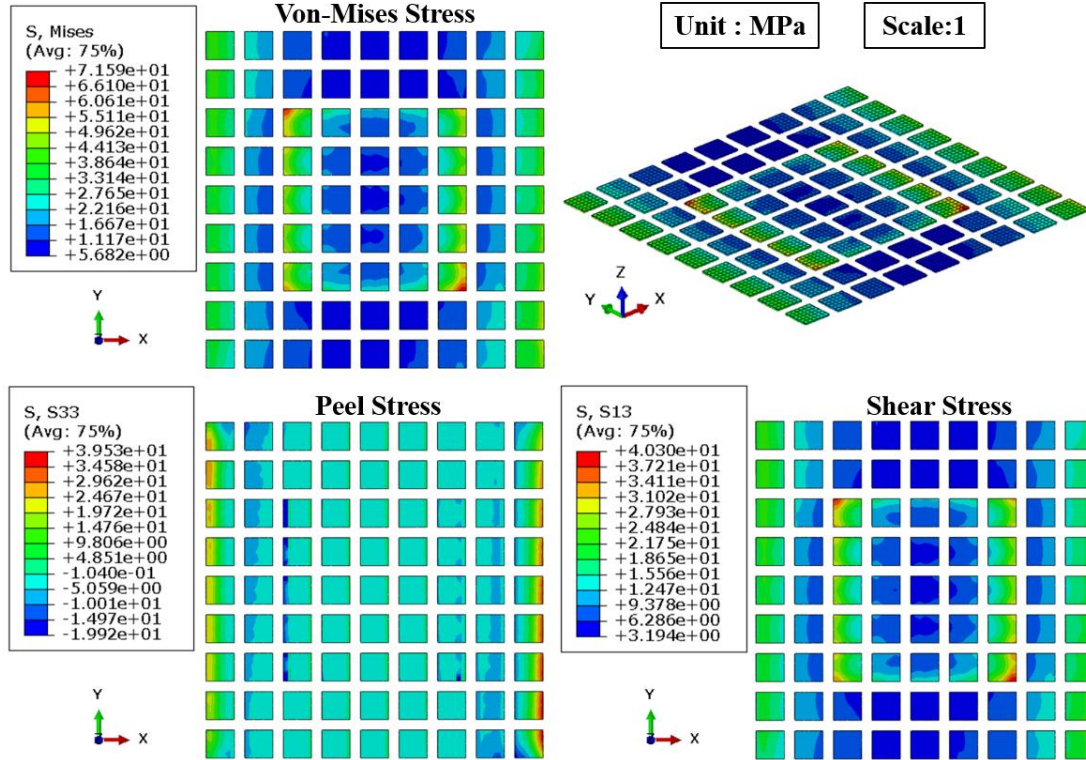


Figure 5.18 : Stress results and distributions of the adhesive pattern shape-2 model.

The 3D peel stress and the 3D shear stress distributions on the adhesive for adhesive pattern shape-2 model are given in Figure 5.19 and Figure 5.20, respectively.

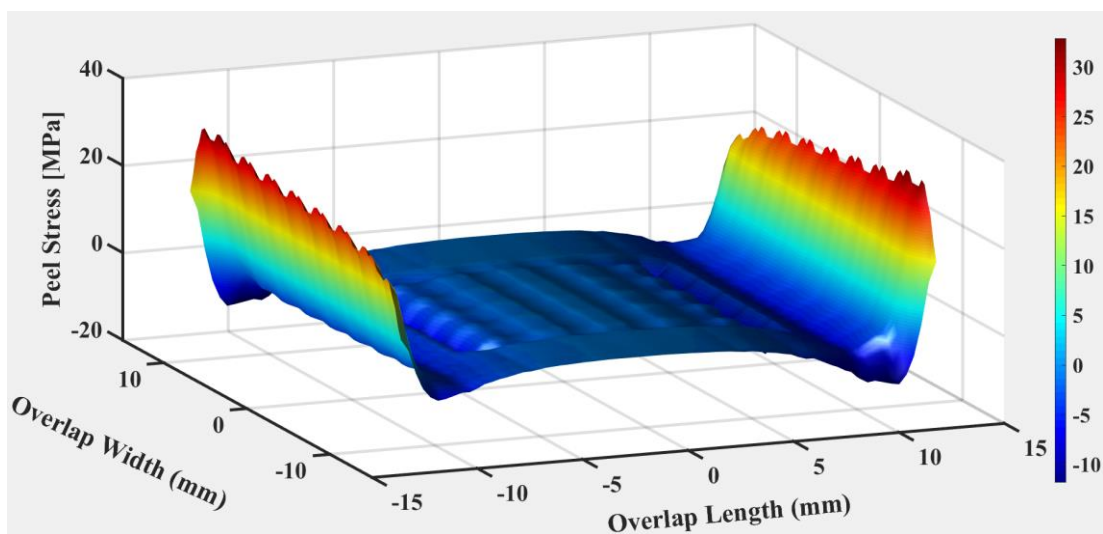


Figure 5.19 : 3D peel stress distribution on the adhesive for adhesive pattern shape-2 model.

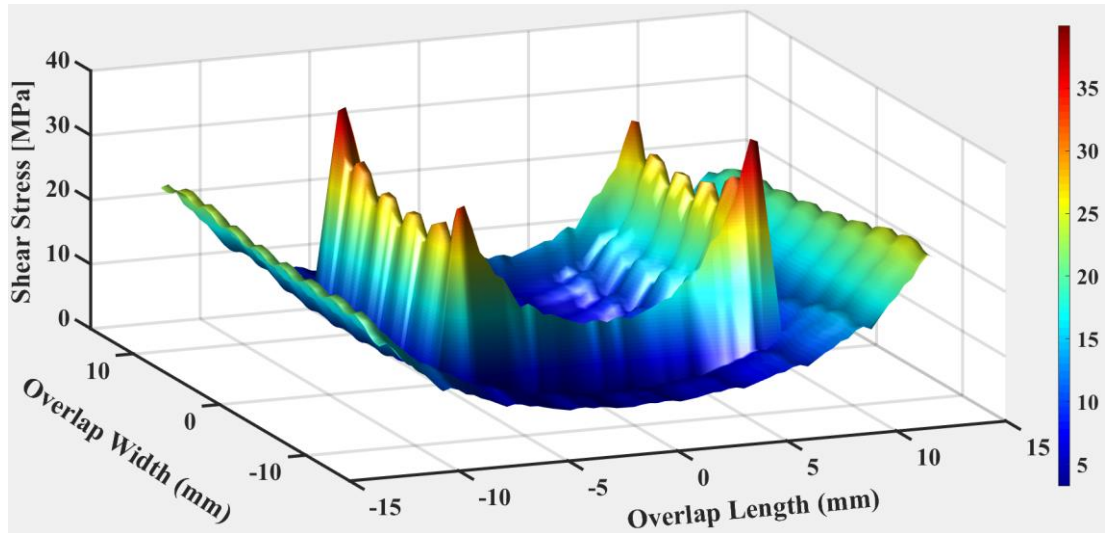


Figure 5.20 : 3D shear stress distribution on the adhesive for adhesive pattern shape-2 model.

The stress distribution of the adhesive pattern shape-3 model is investigated in terms of von-Mises stress, peel stress and shear stress, all are indicated together in Figure 5.21. It can be realized that the stress levels of all types of stress are decreased significantly according to master adhesive model. The results of von-Mises, peel and shear stresses are reduced from 112 MPa to 63 MPa, from 104 MPa to 41 MPa and from 57 MPa to 36 MPa, respectively according to master adhesive model. It can be also realized that the stress distribution profiles are changed for all stresses, due to consisting of different stiffnesses of adhesives and pattern shapes, likewise adhesive pattern shape-1, and 2. In addition, the stress distributions profiles of the adhesive pattern shape-3 are a little bit different from the adhesive pattern shape-1, and 2, due to different pattern shape of cells. The von-Mises and shear stress levels of adhesive pattern shape-3 are a little lower than the adhesive pattern shape-1 but peel stress of the adhesive pattern shape-3 is a little higher than adhesive pattern shape-1. The von-Mises and shear stresses levels of adhesive pattern shape-3 are a little lower than the adhesive pattern shape-2 but the peel stress are nearly same.

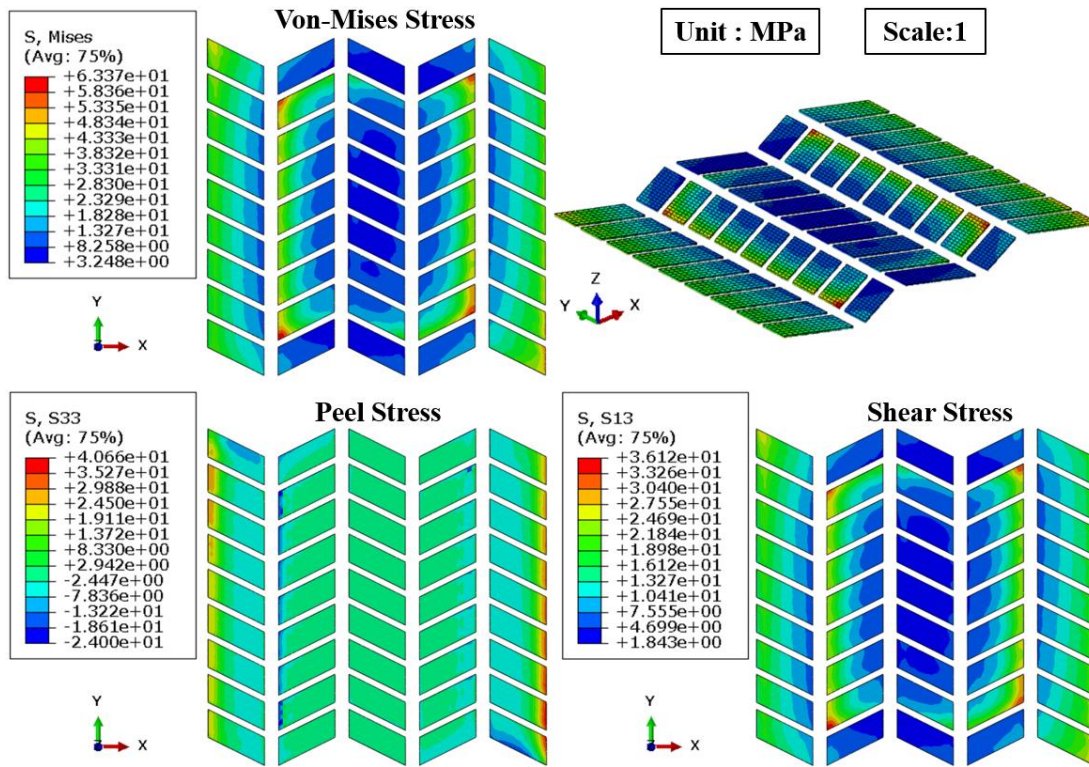


Figure 5.21 : Stress results and distributions of the adhesive pattern shape-3 model.

The 3D peel stress and the 3D shear stress distributions on the adhesive for adhesive pattern shape-3 model are given in Figure 5.22 and Figure 5.23, respectively.

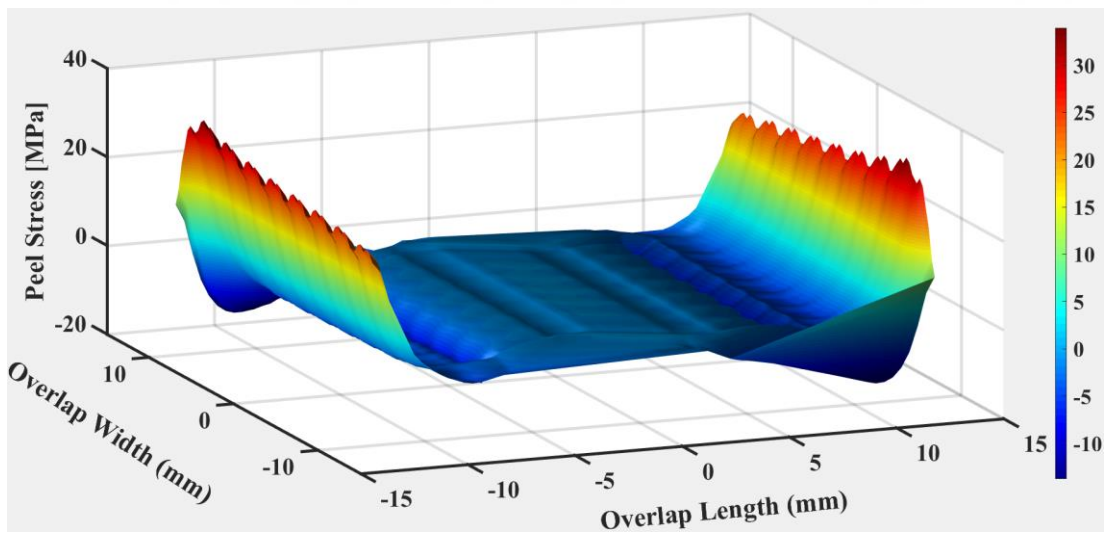


Figure 5.22 : 3D peel stress distribution on the adhesive for adhesive pattern shape-3 model.

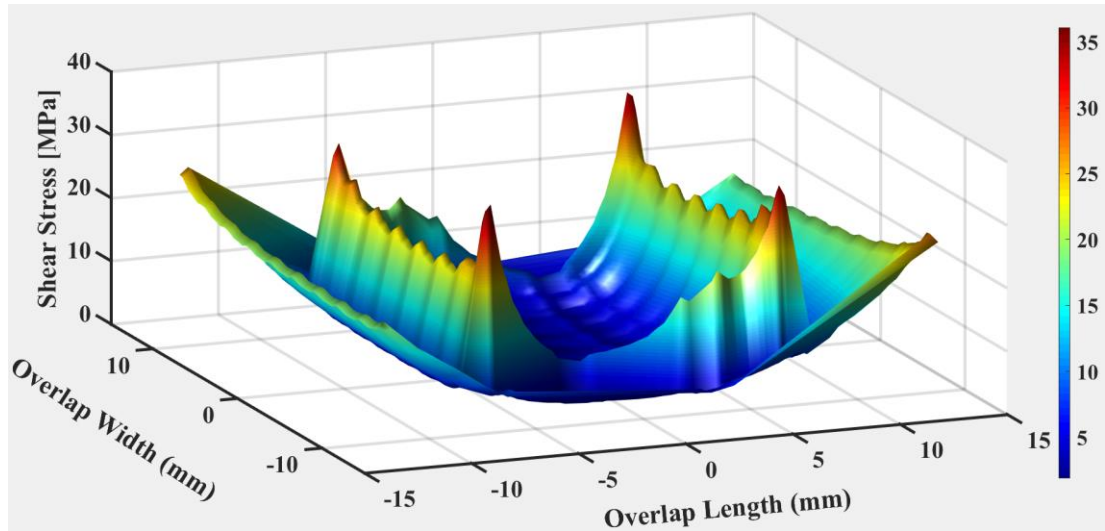


Figure 5.23 : 3D shear stress distribution on the adhesive for adhesive pattern shape-3 model.

The stress distribution of the adhesive pattern shape-4 model is investigated in terms of von-Mises stress, peel stress and shear stress, all are indicated together in Figure 5.24. It can be realized that the stress levels of all types of stress are decreased significantly according to master adhesive model. The results of von-Mises, peel and shear stresses are reduced from 112 MPa to 82 MPa, from 104 MPa to 42 MPa and from 57 MPa to 46 MPa, respectively according to master adhesive model. It can be also realized that the stress distribution profiles are changed for all stresses, due to using different stiffness adhesives and pattern shapes, likewise adhesive pattern shape-1, 2, and 3. In addition, the stress distributions profiles of the adhesive pattern shape-4 are a little bit different from the adhesive pattern shape-1, and 2, due to different pattern shape of cells. The whole stress levels of adhesive pattern shape-4 are higher than the adhesive pattern shape-1, 2, and 3, especially von-Mises and shear stresses of the adhesive pattern shape-4. The specific size of the cells can be a reason of this situation.

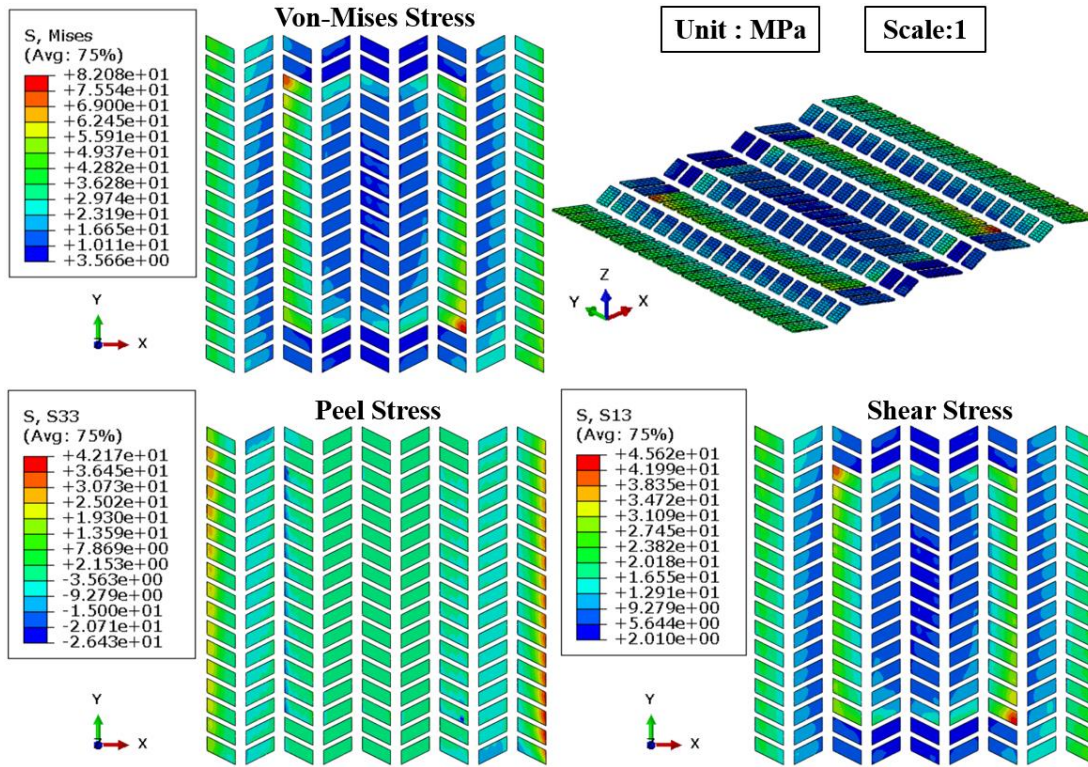


Figure 5.24 : Stress results and distributions of the adhesive pattern shape-4 model.

The 3D peel stress and the 3D shear stress distributions on the adhesive for adhesive pattern shape-4 model are given in Figure 5.25 and Figure 5.26, respectively.

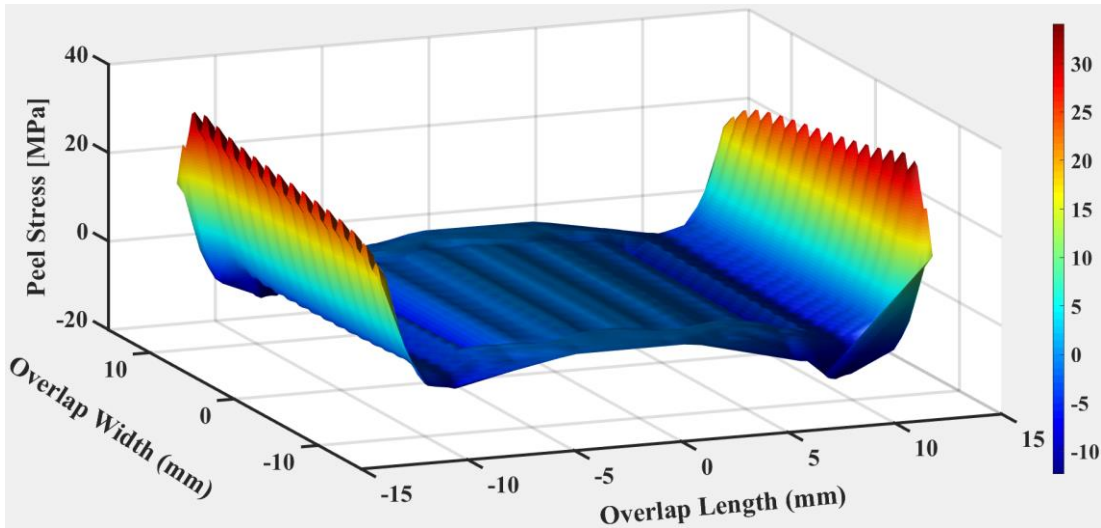


Figure 5.25 : 3D peel stress distribution on the adhesive for adhesive pattern shape-4 model.

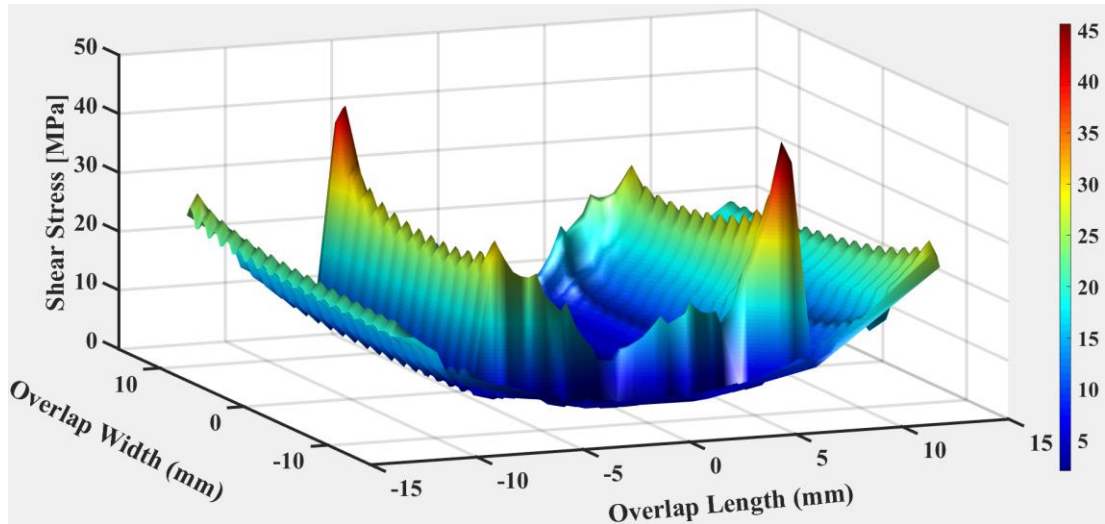


Figure 5.26 : 3D shear stress distribution on the adhesive for adhesive pattern shape-4 model.

The stress distribution of the adhesive pattern shape-5 model is investigated in terms of von-Mises stress, peel stress and shear stress, all are indicated together in Figure 5.27. It can be realized that the stress levels of all types of stress are decreased significantly according to master adhesive model. Furthermore, the whole stress types of the adhesive pattern shape-5 model is the lowest in all other adhesive models, due to taking place high rate of ductile adhesive in the overlap region. The results of von-Mises, peel and shear stresses are reduced from 112 MPa to 43 MPa, from 104 MPa to 34 MPa and from 57 MPa to 24 MPa, respectively according to master adhesive model. It can be also realized that the stress distribution profiles seem similar to adhesive pattern shape-1, and 2. However, stress distributions are more uniform than them. It is known that due to low strength capability of ductile adhesives, there is a limit to use them in the joints. The best way to define ductile and brittle adhesive proportion of using, the master model can be a reference to decide the proportion of using them.

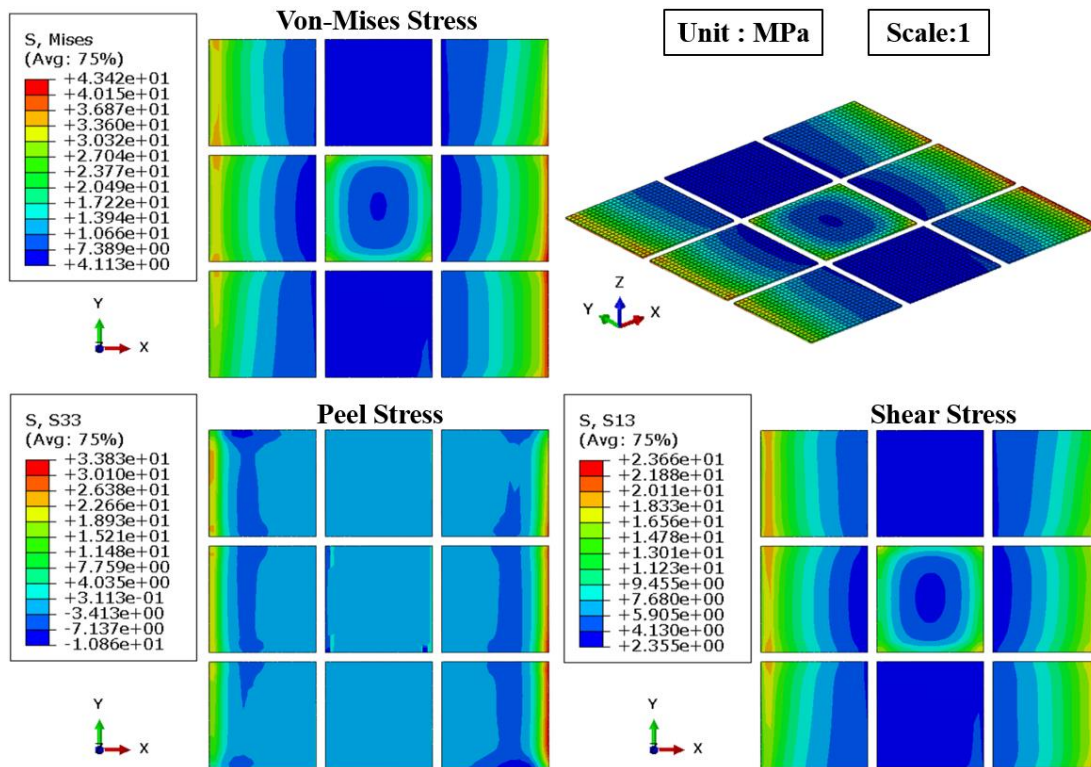


Figure 5.27 : Stress results and distributions of the adhesive pattern shape-5 model.

The 3D peel stress and the 3D shear stress distributions on the adhesive for adhesive pattern shape-4 model are given in Figure 5.28 and Figure 5.29, respectively.

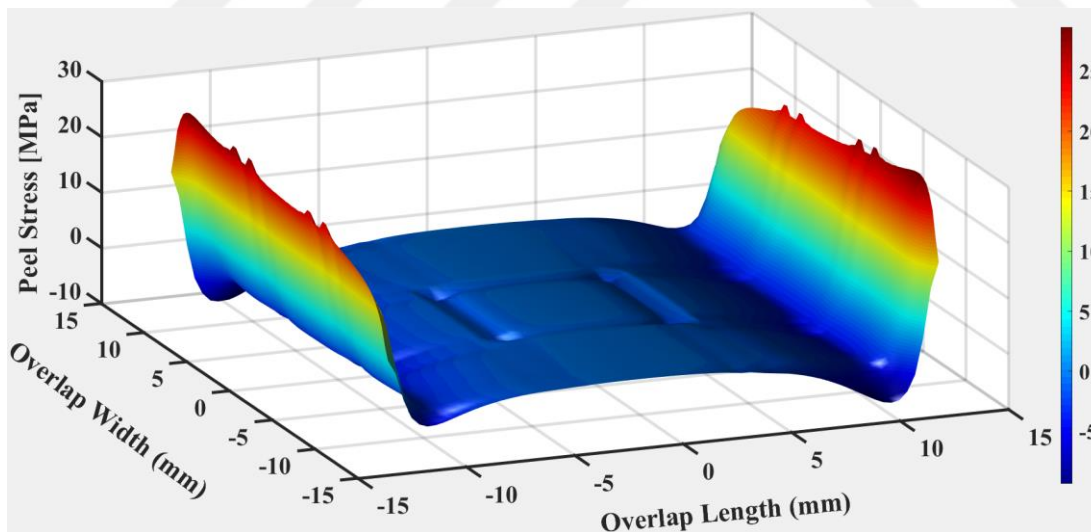


Figure 5.28 : 3D peel stress distribution on the adhesive for adhesive pattern shape-5 model.

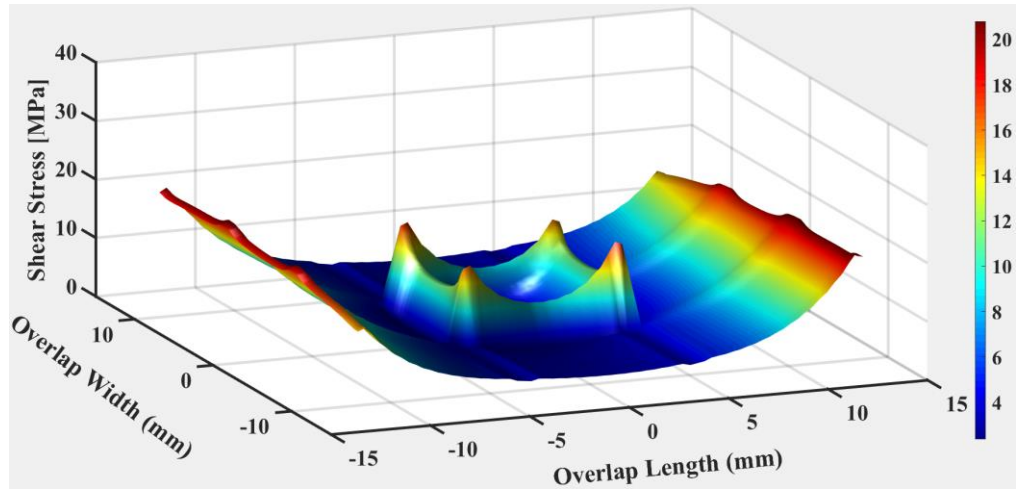


Figure 5.29 : 3D shear stress distribution on the adhesive for adhesive pattern shape-5 model.

The stresses results of all adhesive models are listed in Table 5.1. It can be noticed that due to high ratio of ductile adhesive usage in the overlap region, the lowest stress results occurs in the adhesive pattern shape-5 model as it is expected. However, adhesive pattern shape-1, and 2 are more feasible due to strength of the joints. The joint has the more ductile region, tends to deformed easily.

Table 5.1 : The stress results of all adhesive models.

Adhesives Models	von-Mises Stress (MPa)	Peel Stress (MPa)	Shear Stress (MPa)
Master adhesive model	112.1	104.3	56.7
Adhesive pattern shape-1	65.8	38.8	37.1
Adhesive pattern shape-2	71.6	39.5	40.3
Adhesive pattern shape-3	63.4	40.7	36.1
Adhesive pattern shape-4	82.1	42.2	45.6
Adhesive pattern shape-5	43.4	33.8	23.7

The shear stress distributions through the centerline of overlap region for all adhesive models are presented as a graph in Figure 5.30. It can be easily realized that the peel stress levels of all adhesive pattern shape models in the stress concentration region, which are both ends of the overlap, are decreased almost between 40% - 50% percent of the master adhesive model's peel stress. It can be seen in Figure 5.30, the region of

A, peel stresses of all adhesive pattern shape models are between 25 MPa - 30 MPa, on the other hand, peel stress level of master adhesive model is around 50 MPa.

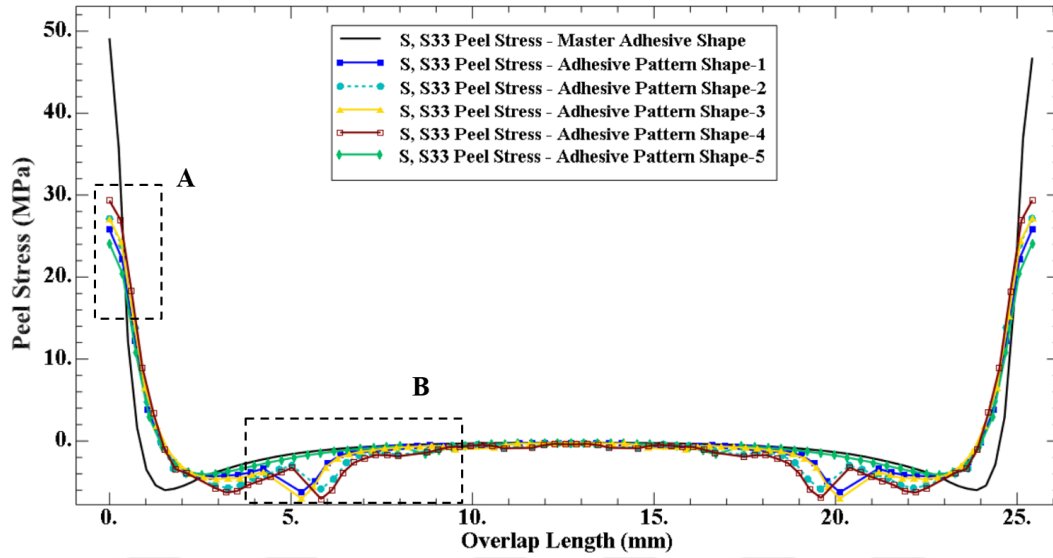


Figure 5.30 : The comparison of peel stress distribution for all adhesives models.

In the region of B, all adhesive shape models, except adhesive pattern shape-5 model, there is a fall and rise on the stress level suddenly, due to ductile-brittle adhesive transition region. It can be realized that adhesive pattern shape-5 model has more ductile region length (approximately 8 mm) of the others and that is why transition region do not exist between 5 mm to 6 mm overlap length but exists between 8 mm to 9 mm, are shown in Figure 5.31 clearly. Brittle adhesive region peel stress levels of all models are nearly zero. It can be noticed that peel stress distributions of the whole models are symmetrical.

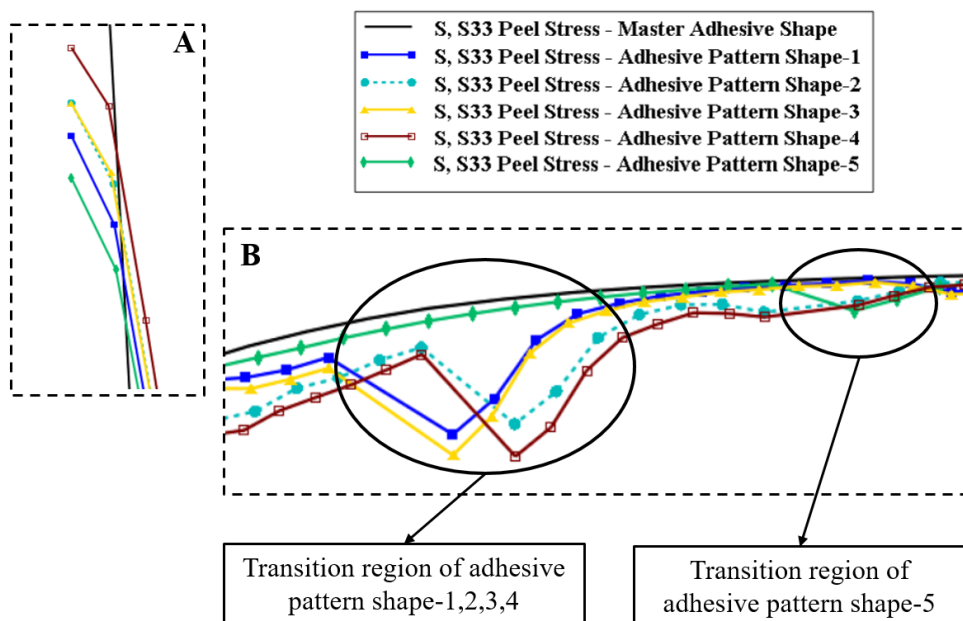


Figure 5.31 : Ductile-brittle adhesives transition regions.

The shear stress distributions through the centerline of overlap region for all adhesive models are presented as a graph in Figure 5.32. It can be easily realized that shear stress levels of all adhesive pattern shape models in the stress concentration region, which are both ends of the overlap, are reduced almost 50% percent of the master adhesive model's shear stress. It can be seen in Figure 5.32, the region of A, shear stresses of all adhesive pattern shape models are between 17 MPa - 22 MPa, on the other hand, shear stress level of master adhesive model is around 40 MPa.

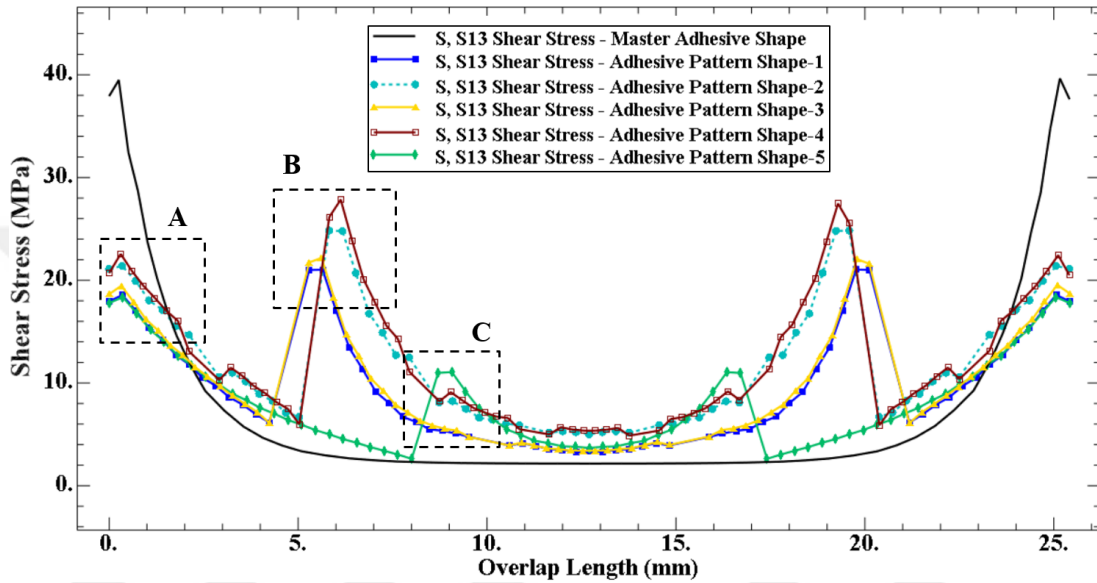


Figure 5.32 : The comparison of shear stress distribution for all adhesives models.

In the region of B, all adhesive shape models, except adhesive pattern shape-5 model, the stress levels suddenly increase to between 20 MPa – 30 MPa, due to brittle adhesive region. It can be also realized that adhesive pattern shape-5 model has more ductile region length (approximately 8 mm) of the others and that is why shear stress levels rises limited according to the others and the transition region of all adhesive pattern shapes models can be appeared clearly. The regions A, B, and C, are shown in the figure 5.33 clearly. Brittle adhesive region peel stress levels of all models are nearly zero. It can be also noticed that shear stress distributions of the whole models are symmetrical, likewise peel stress distributions.

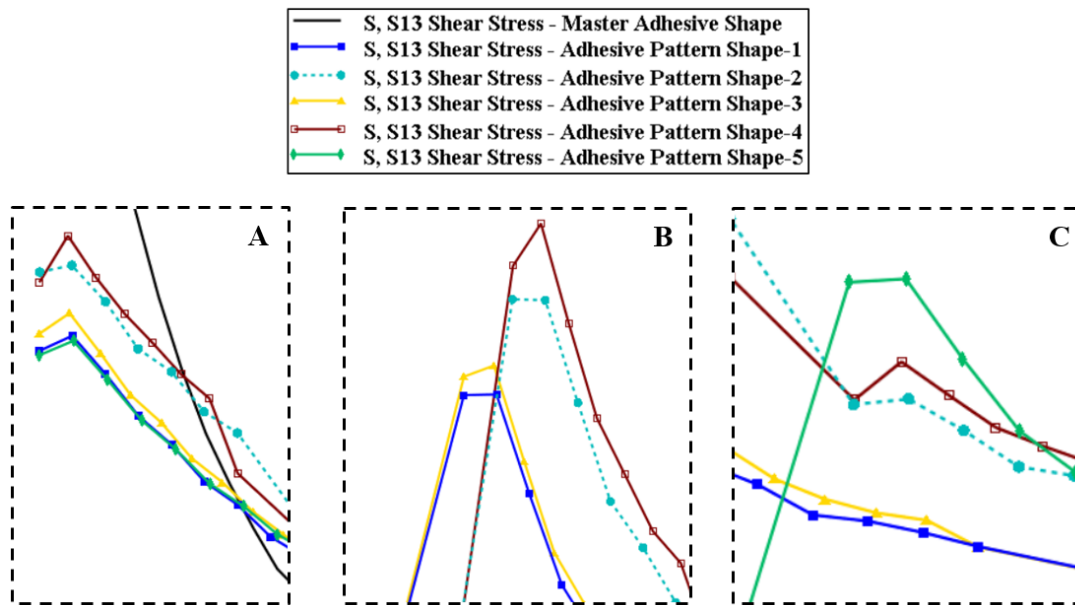


Figure 5.33 : The selected regions A, B, and C, for ductile - brittle adhesives.

For both peel and shear stress distribution in the all adhesive pattern shape models occur more uniform stresses according to the master adhesive model. As it is known, the more uniform stress takes place on the joint, the more the joint is durable.

5.3 Spew Fillet SLJ and Tapered SLJ Effect on the Stress Results for SLJ

The main objective of this study is to obtain low stress levels especially, ends of the overlap region, which are considered stress concentration region of the adhesive, and to obtain more uniform stress distribution on the adhesive. Previous study indicated that SLJ is more delicate than other types of bonded joints, and there have been much more researches and studies on the SLJ. Therefore, SLJ is based on the effect of the spew fillet on the adhesive and tapered adherends stress results. The stress results of both tapered and spew fillet models will be compared each other and master adhesive models, which was given in chapter 5.1, SLJ stress results. Assembly stress results for tapered SLJ models with deformation scale 1 and scale 10 are shown in Figure 5.34 and Figure 5.35, respectively.

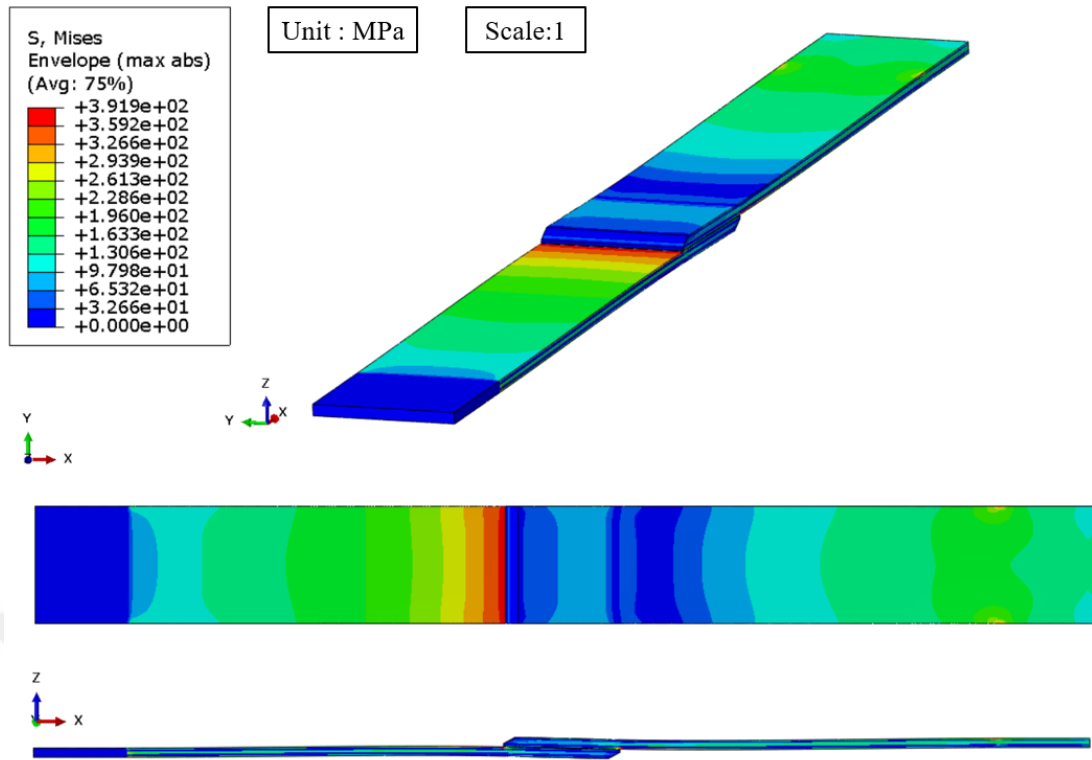


Figure 5.34 : Stress results of the tapered SLJ assembly in scale 1.

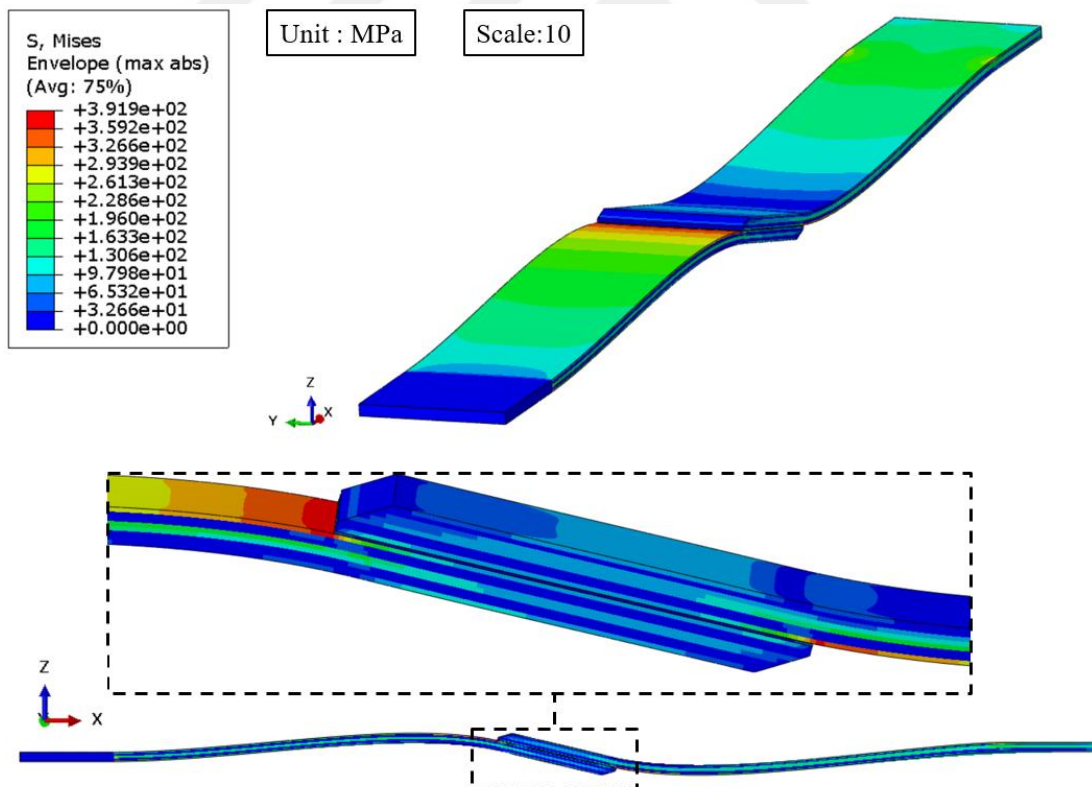


Figure 5.35 : Stress results of the tapered SLJ assembly in scale 10.

Assembly stress results for tapered SLJ models with deformation scale 1 and scale 10 are shown in Figure 5.36 and Figure 5.37, respectively.

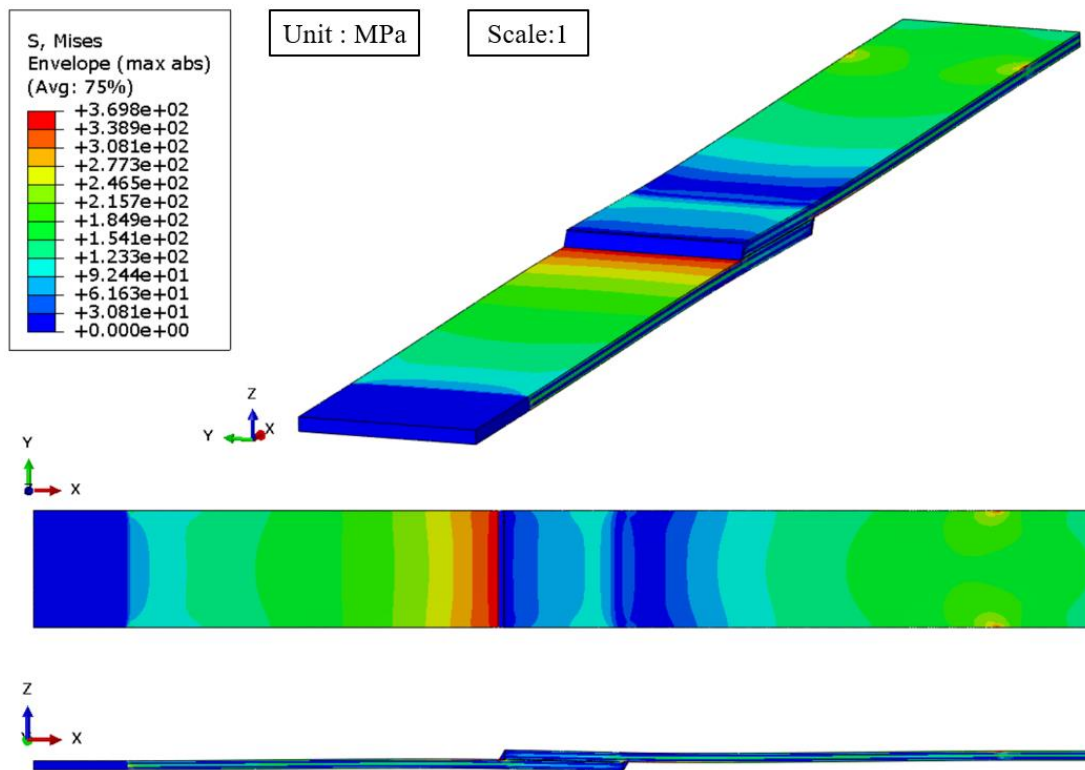


Figure 5.36 : Stress results of the spew fillet SLJ assembly in scale 1.

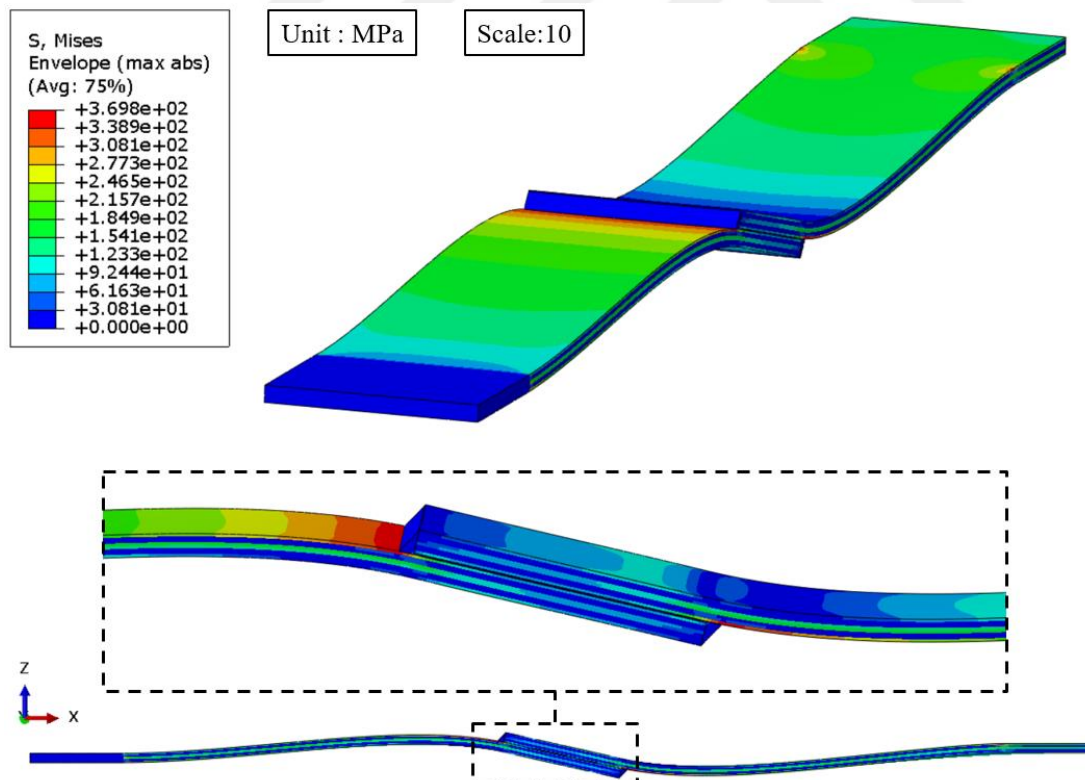


Figure 5.37 : Stress results of the spew fillet SLJ assembly in scale 10.

The stress distribution of the assembly for both models are similar except that the stress levels of the spew fillet SLJ model (370 MPa) is lower than the tapered SLJ models

(392 MPa). On the other hand, both models' stress level are also lower than the master SLJ model, which was given in Figure 5.1 (421 MPa) previously.

Furthermore, the stress distribution of the adhesive in tapered SLJ model is investigated. The von-Mises stress, peel stress and shear stress, are shown in Figure 5.38 together.

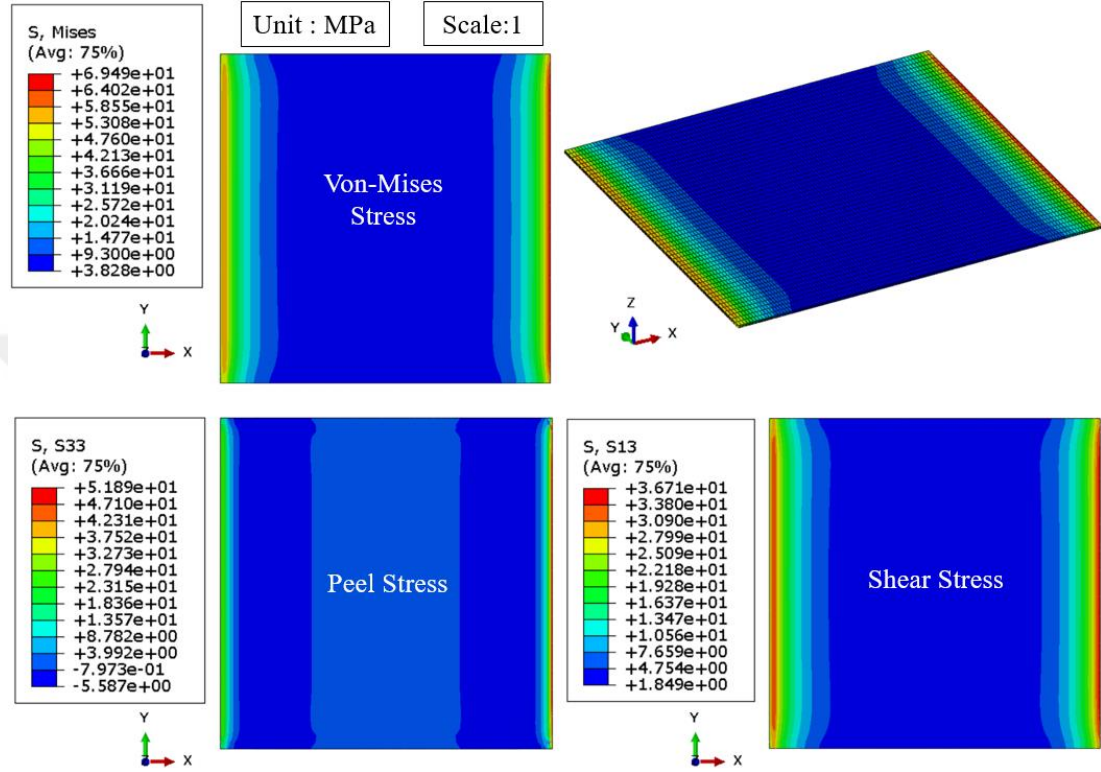


Figure 5.38 : Stress results and distributions of the adhesive in tapered SLJ.

It can be realized that all stress values of the tapered SLJ are below the master SLJ model stress results which were given in Figure 5.3. Due to less peeling effect because of strain, as a result of the tapering. Tapered SLJ can be loaded in shear as well as in tension. It is significant to note that the bending moment that occurs in master SLJ is still present. The taper angle that is also called bevelling angle, effects directly the stress distribution and stress concentration significantly. When the bevelling angle rises, the stress levels reduce considerably and so it gives a clear indication of the importance of the bevelling angle. The maximum stresses of the von-Mises, peel and shear stresses are observed around 70 MPa, 52 MPa, and 37 MPa, respectively. Even if stress concentration is decreased both ends of the overlap region, non-uniform stress distribution is still occurred. The centre region of the adhesive is still not exposed to load, and the stress levels for all stress types are below the 10 MPa. The 3D peel stress

and the 3D shear stress distributions on the adhesive for tapered SLJ model are given in Figure 5.39 and Figure 5.40, respectively.

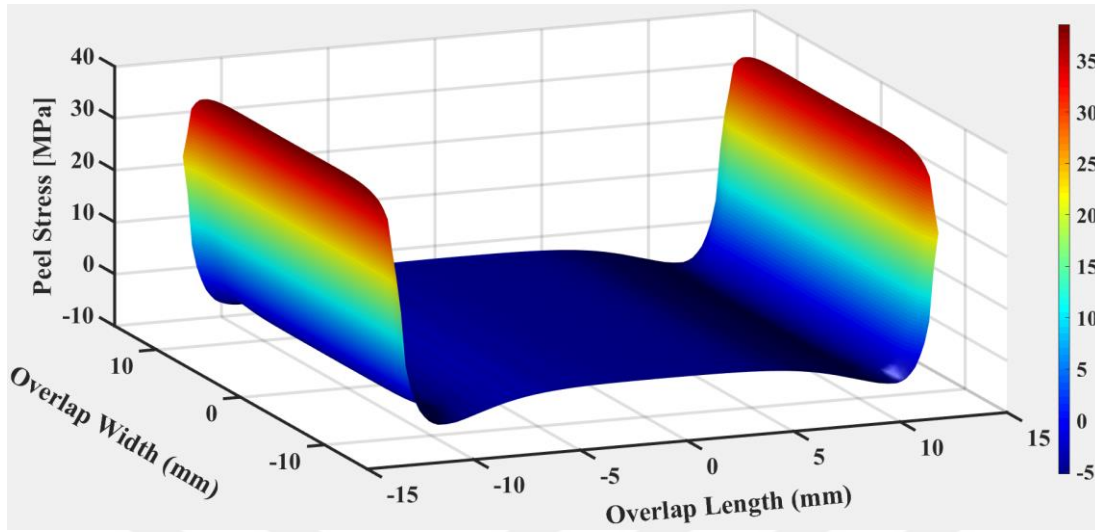


Figure 5.39 : 3D peel stress distribution on the adhesive for tapered SLJ model.

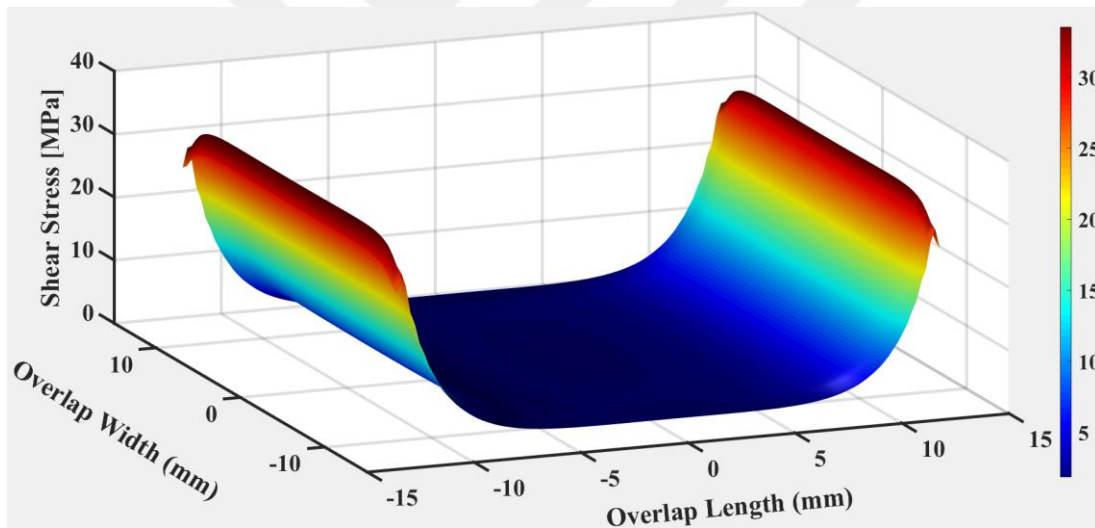


Figure 5.40 : 3D shear stress distribution on the adhesive for tapered SLJ model.

The stress distribution of the adhesive in spew fillet SLJ model is investigated. The von-Mises stress, peel stress and shear stress, are shown in Figure 5.41 together. Furthermore, the 3D peel stress and the 3D shear stress distributions on the adhesive for spew fillet SLJ model are given in Figure 5.42 and Figure 5.43, respectively.

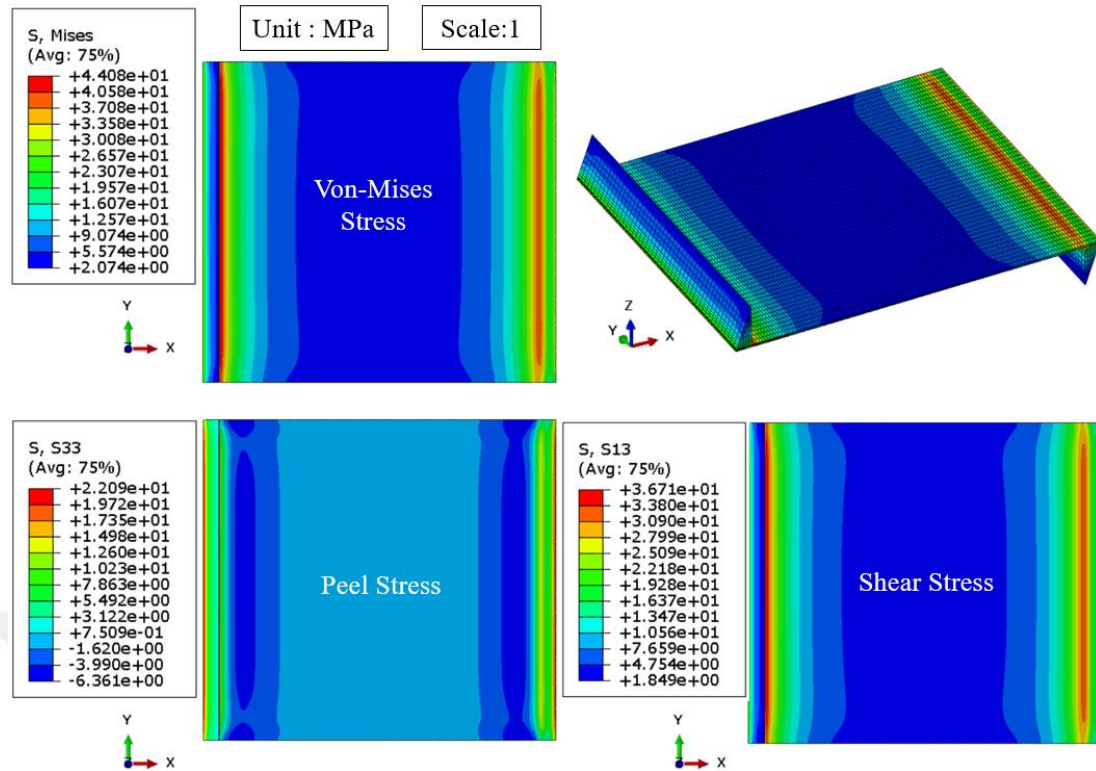


Figure 5.41 : Stress results and distributions of the adhesive in spew fillet SLJ.

It can be realized that all stress values of the spew fillet SLJ are below the tapered SLJ model and master SLJ model stress results which were given in Figure 5.3. The adhesive fillet provides a smoother load transfer and reduces the peak stresses. Likewise in the tapered model, the angle and fillet shape of the adhesive effects stress levels considerably. The maximum stresses of the von-Mises, peel and shear stresses are observed around 44 MPa, 22 MPa, and 37 MPa, respectively.

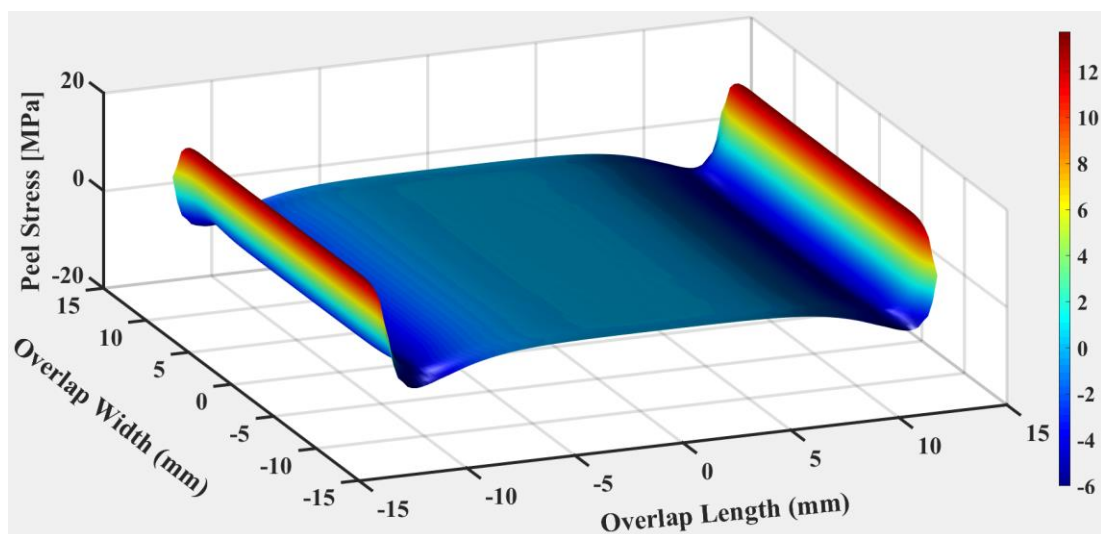


Figure 5.42 : 3D peel stress distribution on the adhesive for spew fillet SLJ model.

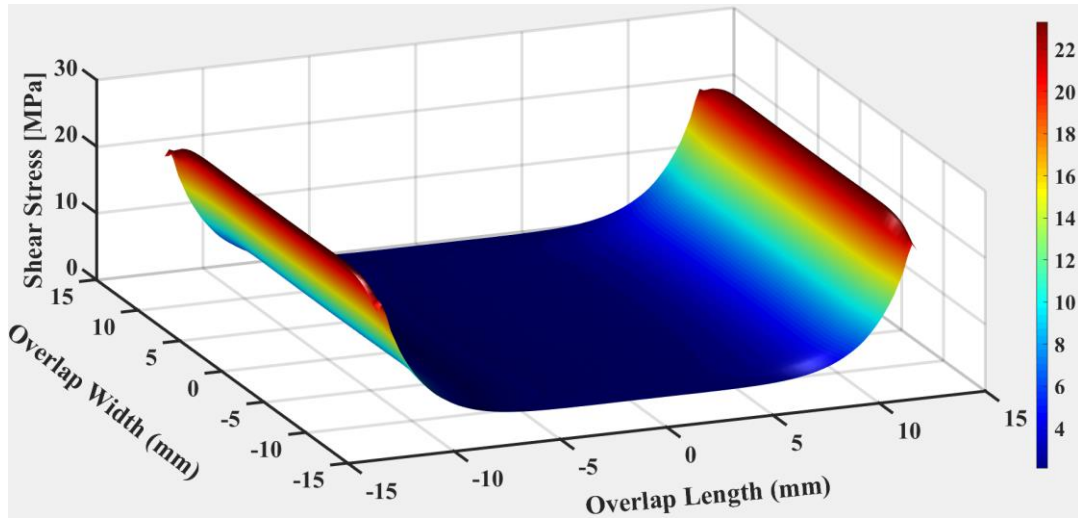


Figure 5.43 : 3D shear stress distribution on the adhesive for spew fillet SLJ model. In order to obtain a better understanding, the stress levels on the adhesive can be given as graph. To avoid FE based misdirection due to contact region, the middle mesh layer is considered and hence, upper two mesh layers of the adhesive are omitted in order to obtain the middle of the adhesive layer stresses. Then, center line of the middle mesh layer's elements are selected through the adhesive and plotted as peel and shear stresses via ABAQUS. All processes steps were similar to master adhesive SLJ model which were shown in Figure 5.4 and Figure 5.5, to obtain centerline of middle region's stress of the adhesive through the overlap region. The whole models' centerline of middle region are indicated in Figure 5.44.

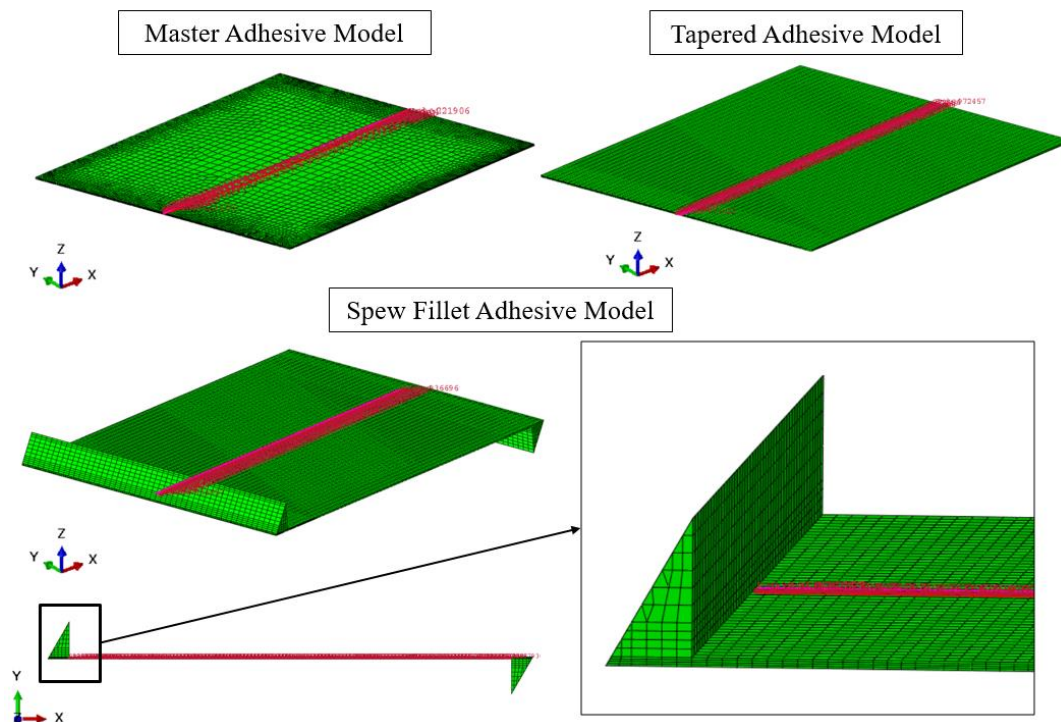


Figure 5.44 : The reading stress path of the whole models.

All the models' peel and shear stresses are plotted as graphs in Figure 5.45 and 5.46, respectively. It can be easily realized that the spew fillet SLJ for both peel and shear stresses are well below the master and tapered SLJs' stress levels in the stress concentration regions, which are the both ends of the overlap region. On the other hand, tapered SLJ is also decreased according to the master SLJ in the stress concentration regions.

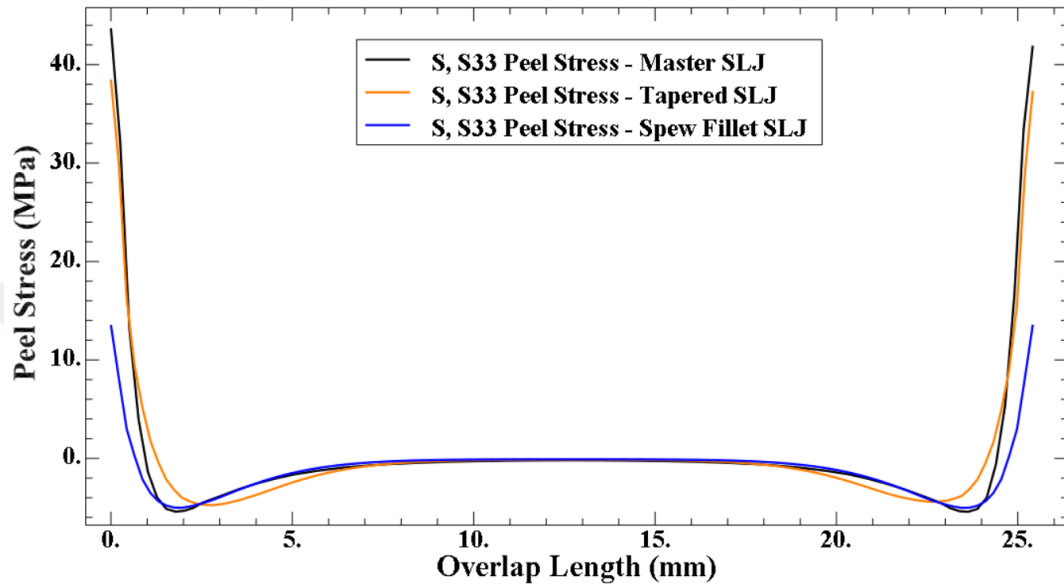


Figure 5.45 : Peel stress along the centerline of the adhesive for all models.

The peel stress levels in the stress concentration region are approximately 45 MPa for master SLJ, 38 MPa for tapered SLJ, and 14 MPa for spew fillet SLJ. The shear stress levels in the stress concentration region are approximately 36 MPa for master SLJ, 33 MPa for tapered SLJ, and 23 MPa for spew fillet SLJ.

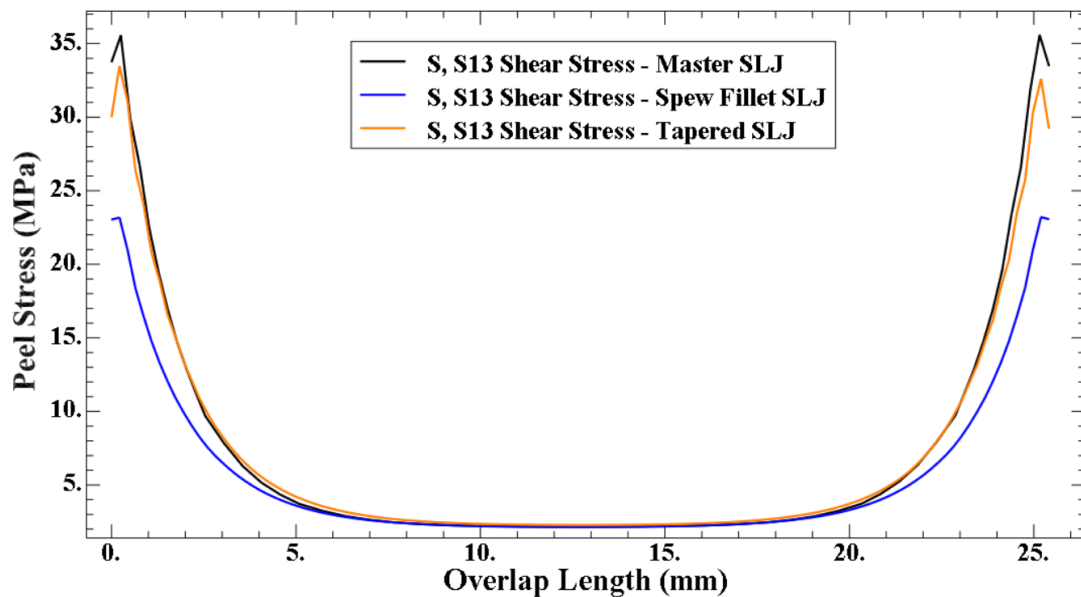


Figure 5.46 : Peel stress along the centerline of the adhesive for all models.

Although there is no significant reduction of the both peel and shear stress for tapered SLJ, it is important to note that, the tapered model is effected significantly taper angles, likewise spew fillet model.

In the center region of the adhesive stresses for all models have similar stress levels, which are almost zero for both peel and shear stresses.

5.4 The Fibers Direction Angle Effects on the Stress Concentration of the Adhesive for SLJ

The purpose of this study is to how the stress distributions react when the angles of fiber directions are changed. For this purpose, as it is stated before, the adherends are unidirectional composite materials with a lay-up of $[0/45/45/0]_s$, which denotes the laminate above consisting of eight plies. The plies above the midplane are of the same orientation, material, and thickness as the plies below the midplane, and hence this is a symmetric laminate. Due to using symmetric composite lay-up, angles possibilities of the eight plies angles reduce four plies. Therefore, just four plies angles are changed with the possibility of 0° , 15° , 30° , 45° , 60° , 75° , and 90° angles. There are seven variables and 4 plies, totally 7^4 cases, that is 2401 cases, are occurred. All cases are generated automatically via MATLAB and Python codes. Then, all cases are performed stress analysis automatically and again all cases stress distribution through the centerline of the adhesive are obtained and plotted corresponding graphs automatically. As it is expected, the lowest peel and shear stresses values are observed the adherends with a lay-up of $[0/0/0/0]_s$ model, due to the fibers alignment with the loadings. On the other hand, the highest peel and shear stresses values are observed the adherends with a lay-up of $[90/90/90/90]_s$ model, due to exposing off-axis loads. The stress concentration rises for peel and shear stresses can be seen in Figure 5.47 and 5.48, respectively. There are eight cases, which are specified in Figure 5.47 and 5.48. Both A region of Figure 5.47 and 5.48, the more fibers angles increase, the more stress rises proportionally.

Furthermore, both B region of Figure 5.47 and 5.48, stress fluctuation can be seen more, as long as fibers angles rise. In briefly, more stress concentration can be observed in the critical regions.

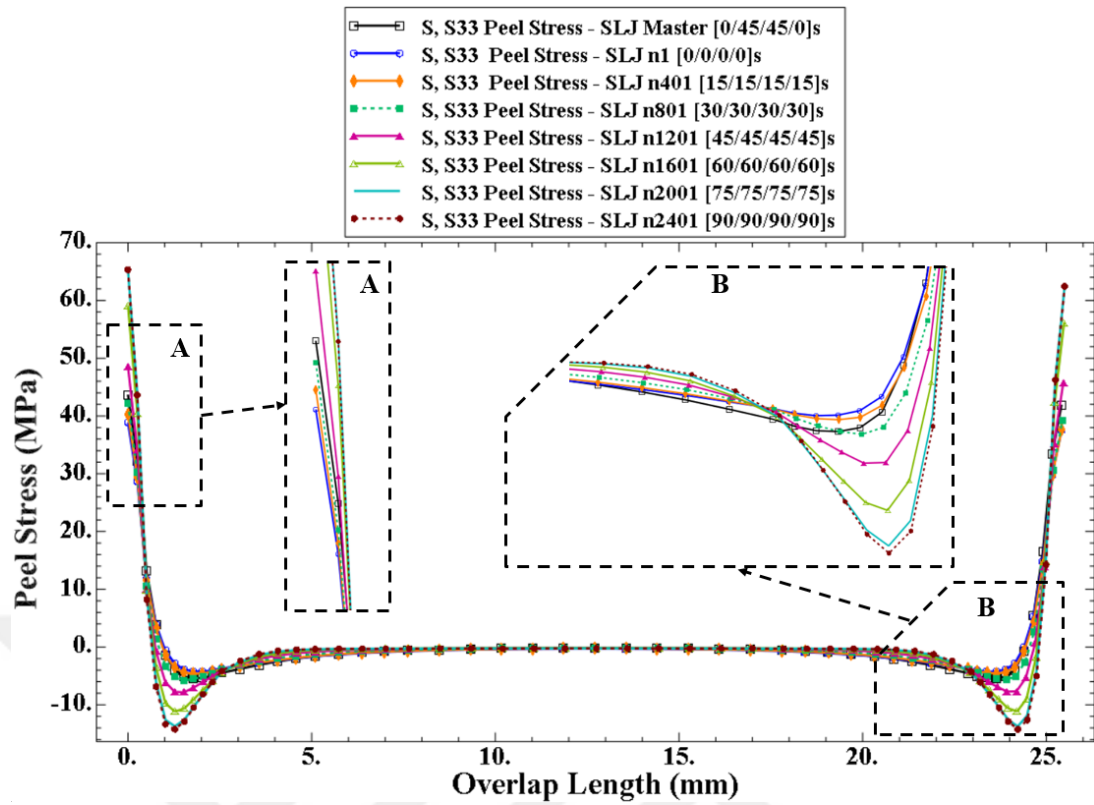


Figure 5.47 : The peel stress changing with the angles of fibers.

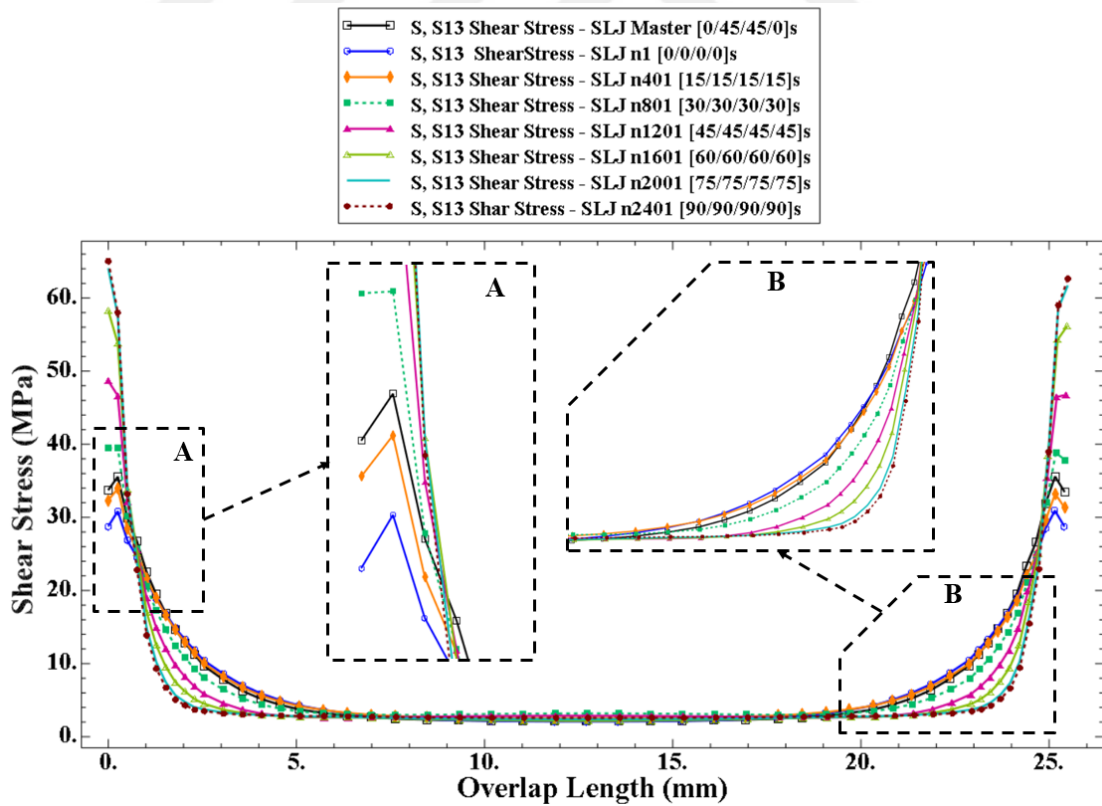


Figure 5.48 : The shear stress changing with the angles of fibers.

It can be seen easily in Figure 5.49 and 5.50, the angle of the fibers effects the peel stress and shear stress levels significantly.

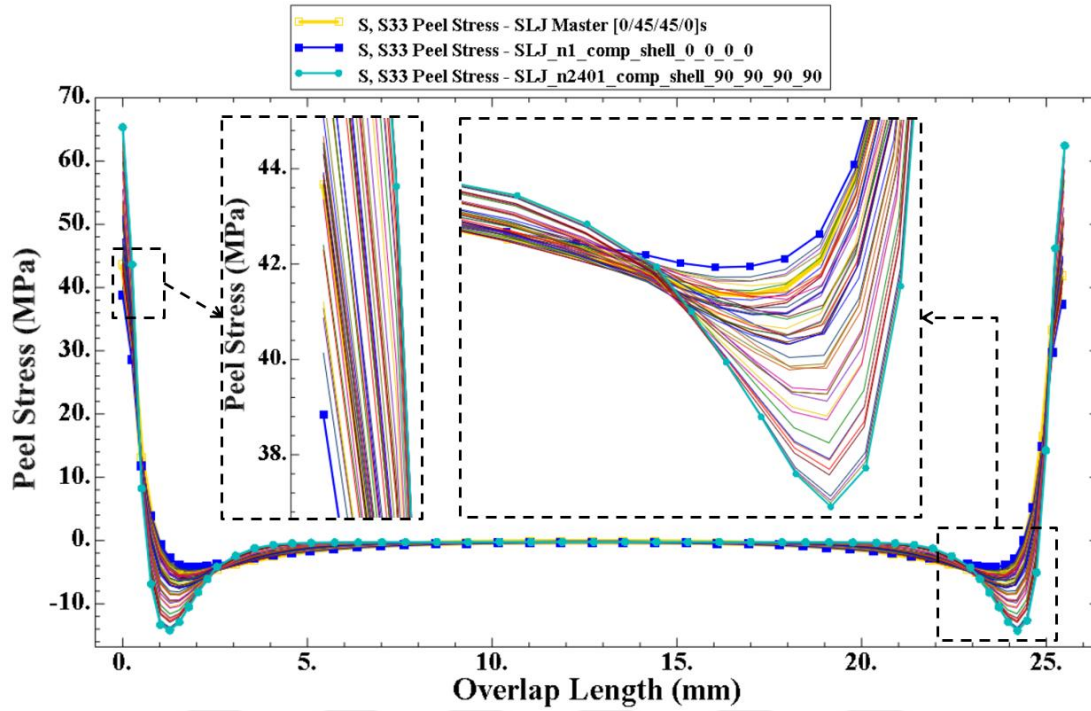


Figure 5.49 : The changing of peel stress according to fibers angles.

The peel and shear stresses of the adherends with a ply-up orientation of $[0/0/0/0]_s$ model are around 39 MPa and 31 MPa in stress concentration region, respectively. The model with $[0/45/45/0]_s$ fiber orientation, which is called master SLJ, the peel and shear stresses are observed around 45 MPa and 36 MPa in stress concentration region, respectively. The model with $[90/90/90/90]_s$ fiber orientation, which is the last one of the 2401 models, the peel and shear stresses are around 66 MPa and 65 MPa, respectively.

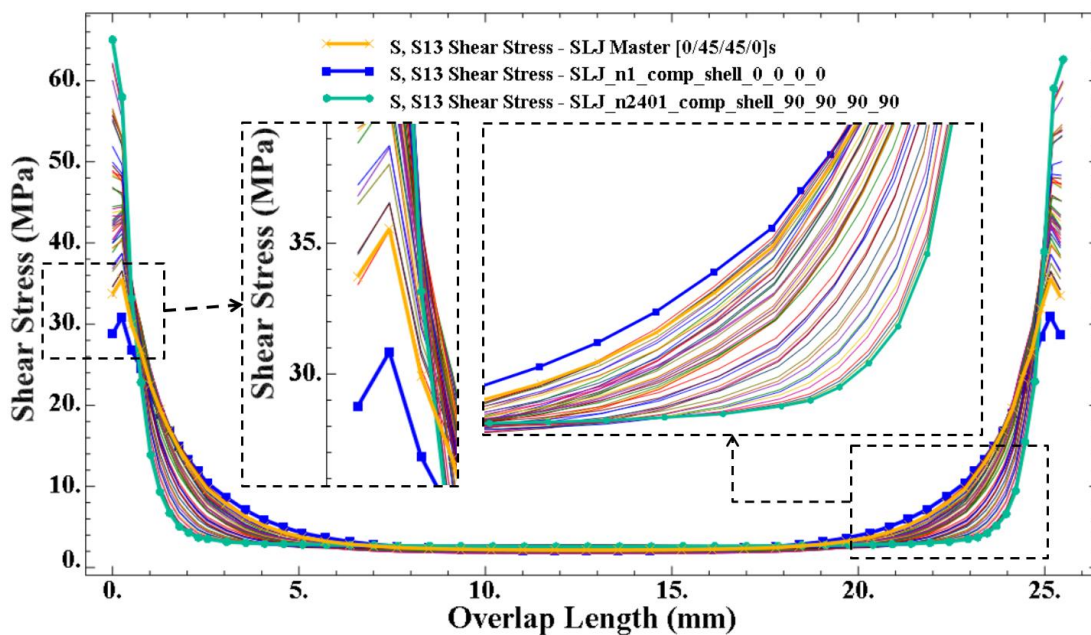


Figure 5.50 : The changing of shear stress according to fibers angles.

There are 51 cases, which are generated and plotted by scripts, are performed by MATLAB, Python, and ABAQUS. All 51 cases are listed in Figure 5.51. Except master SLJ with a lay-up [0/45/45/0]_s adherends model and SLJ n1 with a lay-up [0/0/0/0] adherends model, the other 49 models are selected 49 increment. For instance, the first chosen model is SLJ n49 and the second is SLJ n98. Due to same chosen cases of peel and shear stresses, only the legend of peel stress is given in Figure 5.51.

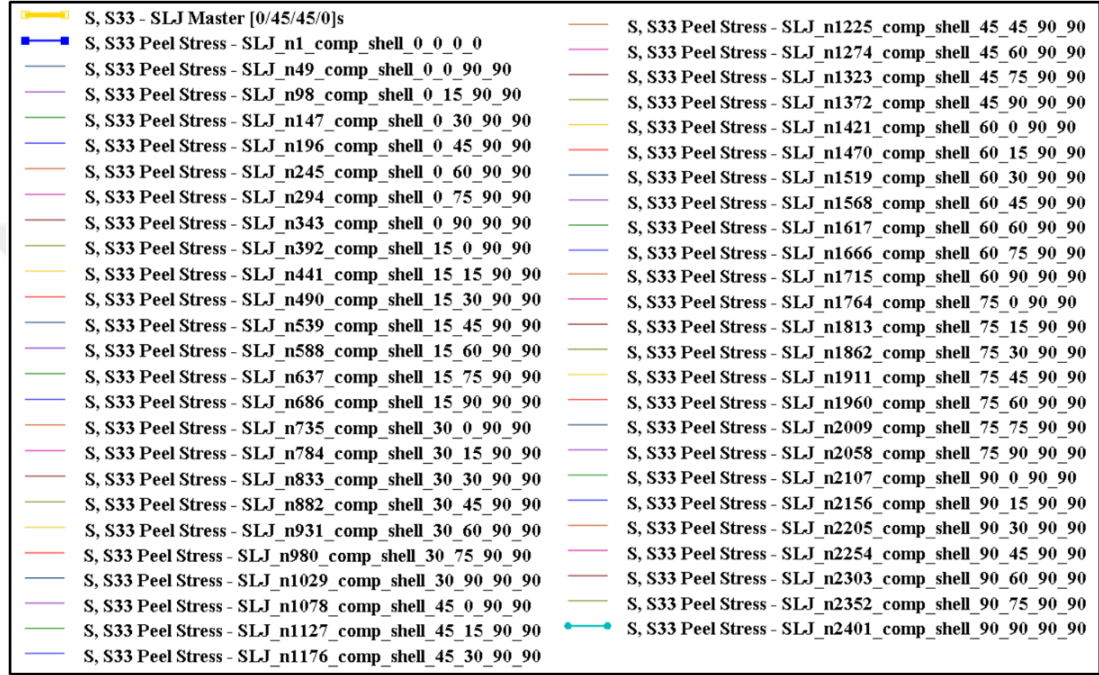


Figure 5.51 : The list of 51 chosen cases and legend chart of peel and shear stress.



6. CONCLUSION AND RECOMMENDATIONS

In this research, SLJ and DLJ configurations are examined and compared each other, in order to realized unbalanced and balanced structures of the configurations. Due to observing the higher stress and non-uniform and unbalanced stresses on the adhesives in SLJ, the study is focused on the SLJ configuration. In order to minimize peak stresses at critical regions such as both ends of the overlap region, and provide better uniform and balanced stresses on the adhesive, the study is investigated pattern and rigidity effects of adhesive on the stress distribution of the adhesively bonded composite joints at first, then the effects of tapered adherends, and spew fillet on the adhesives, and the effects of fibers angles on the adhesives stress distributions finally. The finite element models are created using ABAQUS finite element software program. Furthermore, MATLAB and Python is also used for scripting to generate 2401 cases and performing analysis and finally, plotted the corresponding results of the analyses automatically.

The peel stress and shear stress of SLJ and DLJ were reacted differently under the tensile load, due to the unbalanced and balanced characteristic, respectively. It is revealed that the DLJ is more reliable than the SLJ. The maximum deformation of the SLJ for both upper and lower adherends were observed ends of the adherends, which is seen easily in 10 scale factor deformation figures.

The key part of the adhesively bonded joint is adhesive obviously. The study reveals that the adhesive's material properties and pattern combinations play a key role to strength of the joints. The ductile adhesives, whose elastic modulus are lower than the general adhesives, are located the peak stress region with the specific pattern and geometric dimension. Due to observing the maximum deformation at the ends of the adherends in SLJ, in order to prevent peak stress levels, the deformations of the adherends ends can be allowed to by using ductile adhesives in the limited regions, which are the peak stress regions. On the other hand, the brittle adhesives, whose elastic modulus are higher than the general adhesives, are placed the middle region of the adhesive. As a result, mixed adhesive and specific pattern shape and geometry models are caused more uniform stresses and less peak stresses on the adhesives.

Another way to reduce the peak stresses on the adhesive, is tapered adherends models and spew fillet model. The key parameter in these types of the joints are angle of the tapered adherends, which is also called bevelling angle. The taper angle, effects directly the stress distribution and stress concentration significantly for the model. There are many different types of tapered joints such as inside taper, outside taper, and inside taper and adhesive fillet, which are shown in Figure 5.35, respectively. The field of the taper model can be examined in details for future studies.

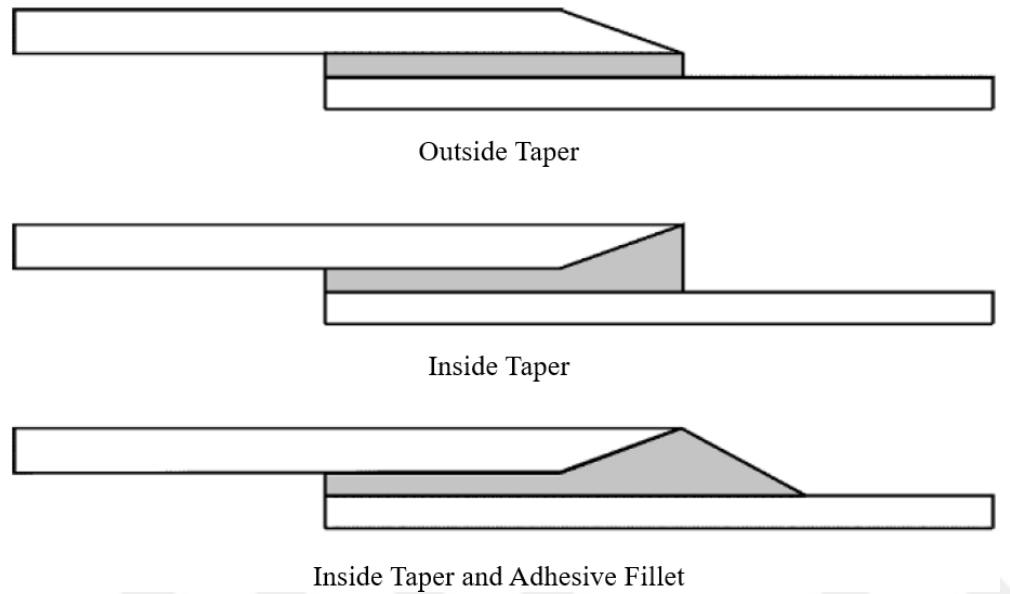


Figure 6.1 : The different types of taper SLJs.

In this research, it is realized that the taper adherends model's stress levels are just below the master SLJ model. Unlike the taper adherends model, very low stress levels are observed significantly in the spew fillet model. The angle of the spew fillet also plays a key role to reduce the peak stress levels. The field of the spew fillet model can be investigated more as the contact between adhesive and adherends in the vertical region and shape of the spew fillets and adherends.

The last issue of the study is the fiber angles effects on the stress distribution and stress levels of the adhesives. The wide range of stress levels of the joints change in the models, especially in the stress concentration region. It can be focused on that the upper and lower adherends have opposite fiber orientations in the future studies. Unlike this study, the peel and shear stresses on the both ends can be observed differently each other and it is possible to obtain non-symmetrical stress results on the ends of the overlap regions.

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APPENDICES

APPENDIX A: The macro of ABAQUS model generating via seqrep function

APPENDIX B: The seqrep function

APPENDIX C: The macro of running analysis of ABAQUS model

APPENDIX D: The macro of capturing and plotting the analysis results of ABAQUS model

APPENDIX E: The macro of capturing and plotting the analysis results of ABAQUS model in same chart.



APPENDIX A

```
clear all;clc;close all;
NameArray={};
temppy='SLJ_Composite_Shell_template_v1.inp';
phi_all=[0 15 30 45 60 75 90];
phi_list=seqrep(phi_all,4);
len=size(phi_list,1);
A_matrixes=[];

for k=1:1:4

    phi=phi_list(k,:);
    name=strcat('SLJ_n', int2str(k),'_comp_shell_', int2str(phi(1)), '_',
int2str(phi(2)), '_', int2str(phi(3)), '_', int2str(phi(4)), '.inp')

    firstply=strcat('0.25, 5, Adherent_Lamina,', int2str(phi(1)), ', Ply-1')
    secondply=strcat('0.25, 5, Adherent_Lamina,', int2str(phi(2)), ', Ply-2')
    thirdply=strcat('0.25, 5, Adherent_Lamina,', int2str(phi(3)), ', Ply-3')
    forthply=strcat('0.25, 5, Adherent_Lamina,', int2str(phi(4)), ', Ply-4')
    data=textread(temppy,'%s','delimiter','\n','whitespace','');
    data(214987)={firstply};
    data(149656)={firstply};
    data(214988)={secondply};
    data(149657)={secondply};
    data(214989)={thirdply};
    data(149658)={thirdply};
    data(214990)={forthply};
    data(149659)={forthply};
    f_out=fopen(name,'w')
    for p=1:1:size(data,1)
        fprintf(f_out, '%s\n',char(data(p)))
    end
    fclose('all');

end
```

APPENDIX B

```
function result=seqrep(V,N)
l=length(V)
a=zeros(l^N,N)
for i=1:N
    a(:,i)=reshape(reshape(repmat((1:l),1,l^(N-1)),l^i,l^(N-i)),l^N,1);
end
result=V(a);
```

APPENDIX C

```
clear all;clc;close all;
NameArray={};
phi_all=[0 15 30 45 60 75 90];
phi_list=seqrep(phi_all,4);
len=size(phi_list,1);

f_out=fopen('abaqus_runner_1-2401.py','w')

fprintf(f_out, 'from abaqus import *\n' );
fprintf(f_out, 'from abaqusConstants import *\n' );
fprintf(f_out, 'from caeModules import *\n' );
fprintf(f_out, 'import job\n' );

path='C:/Users/Ferhat
Demir/Desktop/Master_Thesis/Master_Thesis/CAE/04_Fibers_Direction_Effects/M
acro/'

for k=1:1:4
    phi=phi_list(k,:)
    nameinp=strcat('SLJ_n', int2str(k),'_comp_shell_', int2str(phi(1)), '_ ',
int2str(phi(2)),'_', int2str(phi(3)),'_', int2str(phi(4)),'.inp');
    name=strcat('SLJ_n', int2str(k),'_comp_shell_', int2str(phi(1)), '_ ',
int2str(phi(2)),'_', int2str(phi(3)),'_', int2str(phi(4)));
    p1=strcat('mdb.ModelFromInputFile(name="",name,"',
inputFileName="",path,nameinp,"")\n');
    p2=strcat('a = mdb.models["',name,""].rootAssembly\n');
    p3=strcat('mdb.Job(name="",name,"', model="",name,"', numDomains=8,
numCpus=8)\n');
```

```

p4=strcat('mdb.jobs["',name,'"'].submit()\n');
p5=strcat('mdb.jobs["',name,'"'].waitForCompletion()\n');
p6=strcat('del mdb.jobs["',name,'"']\n');
p7=strcat('del mdb.models["',name,'"']\n');
fprintf(f_out, p1 )
fprintf(f_out, p2 );
fprintf(f_out, p3 );
fprintf(f_out, p4 );
fprintf(f_out, p5 );
fprintf(f_out, p6 );
fprintf(f_out, p7 );

end

```

APPENDIX D

```

from abaqus import *
from abaqusConstants import *
from caeModules import *
import visualization

phi=[0,15,30,45,60,75,90]
counter=0
phi_list=[]
for phi1 in phi:
    for phi2 in phi:
        for phi3 in phi:
            for phi4 in phi:
                counter=counter+1
                phi_list.append([counter,phi1,phi2,phi3,phi4])

begin=1
end=2439
path='C:/Users/Ferhat
Demir/Desktop/Master_Thesis/Master_Thesis/CAE/04_Fibers_Direction_Effects/M
acro/'
countt=begin-1

```

```
while countt<end :
```

```
    name='SLJ_n'+'%i'%phi_list[countt][0]+'_comp_shell_'+'%i'%phi_list[countt][1]+'_'+'%i'%phi_list[countt][2]+'_'+'%i'%phi_list[countt][3]+'_'+'%i'%phi_list[countt][4]
```

```
    session.Viewport(name='Viewport: 1', origin=(0.0, 0.0), width=200, height=150)
```

```
    session.viewports['Viewport: 1'].makeCurrent()
```

```
    o1name=path + name +'.odb'
```

```
    o1 = session.openOdb(name=o1name)
```

```
    session.viewports['Viewport: 1'].setValues(displayedObject=o1)
```

```
    odb = session.odbs[o1name]
```

```
    session.viewports['Viewport: 1'].odbDisplay.display.setValues(plotState=(  
        CONTOURS_ON_DEF, ))
```

```
    xyList = visualization.XYDataFromShellThickness(odb=odb,  
        outputPosition=INTEGRATION_POINT, variable=('S',  
INTEGRATION_POINT, ((  
        COMPONENT, 'S11'), )), ), elementPick=('LOWER_ADHERENT-  
1', 1, (  
        ['#0:106 #800000 ], ), ), )
```

```
    curveList = session.curveSet(xyData=xyList)
```

```
    x = session.xyPlots['XYPlot-1']
```

```
    session.viewports['Viewport: 1'].setValues(displayedObject=x)
```

```
    xyp = session.xyPlots['XYPlot-1']
```

```
    chartName = xyp.charts.keys()[0]
```

```
    chart = xyp.charts[chartName]
```

```
    chart.setValues(curvesToPlot=curveList)
```

```
    yaxis = chart.axes2[0]
```

```
    session.viewports['Viewport: 1'].setValues(displayedObject=xyp)
```

```
    titlename=name + '_Thickness_S11'
```

```
    pngname=name + '_Thickness_S11.png'
```

```
    session.viewports['Viewport:  
1'].setValues(titleStyle=CUSTOM,customTitleString=(titlename))
```

```
    session.printToFile(fileName=pngname, format=PNG,  
canvasObjects=(session.viewports['Viewport: 1'], ))
```

```
    session.charts['Chart-  
1'].legend.area.setValues(positionMethod=AUTOMATIC,  
alignment=BOTTOM_CENTER)
```

```

xyList2 = visualization.XYDataFromShellThickness(odb=odb,
          outputPosition=INTEGRATION_POINT, variable=((('LE',
INTEGRATION_POINT, ((
          COMPONENT, 'LE11'), )), ),
elementPick=((('LOWER_ADHERENT-1', 1, (
          ['#0:106 #800000'], )), ), )
curveList = session.curveSet(xyData=xyList2)
xyp = session.xyPlots['XYPlot-1']
chartName = xyp.charts.keys()[0]
chart = xyp.charts[chartName]
chart.setValues(curvesToPlot=curveList)
yaxis = chart.axes2[0]
session.viewports['Viewport: 1'].setValues(displayedObject=xyp)
titlename=name + '_Thickness_LE11'
pngname=name + '_Thickness_LE11.png'
session.viewports['Viewport:
1'].setValues(titleStyle=CUSTOM,customTitleString=(titlename))
session.printToFile(fileName=pngname, format=PNG,
canvasObjects=(session.viewports['Viewport: 1'], ))
session.charts['Chart-
1'].legend.area.setValues(positionMethod=AUTOMATIC,
alignment=BOTTOM_CENTER)

```

```

xyList3 = visualization.XYDataFromShellThickness(odb=odb,
          outputPosition=INTEGRATION_POINT, variable=((('LE',
INTEGRATION_POINT, ((
          INVARIANT, 'Max. In-Plane Principal'), )), ), elementPick=((
          'LOWER_ADHERENT-1', 1, (['#0:106 #800000'], )), ), )
curveList = session.curveSet(xyData=xyList3)
xyp = session.xyPlots['XYPlot-1']
chartName = xyp.charts.keys()[0]
chart = xyp.charts[chartName]
chart.setValues(curvesToPlot=curveList)
yaxis = chart.axes2[0]
session.viewports['Viewport: 1'].setValues(displayedObject=xyp)
titlename=name + '_Thickness_LEmax'

```

```

    pngname=name + '_Thickness_LEmax.png'
    session.viewports['Viewport:
1'].setValues(titleStyle=CUSTOM,customTitleString=(titlename))
    session.printToFile(fileName=pngname, format=PNG,
canvasObjects=(session.viewports['Viewport: 1'], ))
    session.charts['Chart-
1'].legend.area.setValues(positionMethod=AUTOMATIC,
alignment=BOTTOM_CENTER)

    xyList4 = visualization.XYDataFromShellThickness(odb=odb,
outputPosition=INTEGRATION_POINT,
        variable=((('TSHR13', INTEGRATION_POINT), ),
elementPick=((('LOWER_ADHERENT-1', 1, (
        ['#0:106 #800000'], )), ), )
    curveList = session.curveSet(xyData=xyList4)
    xyp = session.xyPlots['XYPlot-1']
    chartName = xyp.charts.keys()[0]
    chart = xyp.charts[chartName]
    chart.setValues(curvesToPlot=curveList)
    yaxis = chart.axes2[0]
    session.viewports['Viewport: 1'].setValues(displayedObject=xyp)
    titlename=name + '_Thickness_THSR13'
    pngname=name + '_Thickness_THSR13.png'
    session.viewports['Viewport:
1'].setValues(titleStyle=CUSTOM,customTitleString=(titlename))
    session.printToFile(fileName=pngname, format=PNG,
canvasObjects=(session.viewports['Viewport: 1'], ))
    session.charts['Chart-
1'].legend.area.setValues(positionMethod=AUTOMATIC,
alignment=BOTTOM_CENTER)

    session.Path(name='Adhesive_Path', type=EDGE_LIST,
expression=((('ADHESIVE-1', (
        (183535, 6, 1, 1), (183517, 6, 1, 1), (183510, 3, 3, 1), (183502, 5, 2, 1),
        (183494, 5, 2, 1), (182623, 5, 2, 1), (182618, 5, 2, 1), (182613, 3, 3, 1),
        (182609, 5, 2, 1), (182605, 5, 2, 1), (182240, 5, 2, 1), (182233, 5, 2, 1),
        (182227, 5, 2, 1), (182222, 5, 2, 1), (182217, 5, 2, 1), (182017, 4, 3, 1),
        (182011, 4, 3, 1), (182006, 4, 3, 1), (182002, 6, 1, 1), (181886, 5, 2, 1),

```

```

(181885, 5, 2, 1), (181888, 3, 3, 1), (181877, 5, 2, 1), (181876, 5, 2, 1),
(181879, 3, 3, 1), (181922, 5, 2, 1), (181921, 5, 2, 1), (181924, 3, 3, 1),
(181913, 5, 2, 1), (181912, 5, 2, 1), (181915, 3, 3, 1), (181935, 6, 1, 1),
(181932, 6, 1, 1), (181929, 6, 1, 1), (181928, 6, 1, 1), (182092, 6, 1, 1),
(182088, 6, 1, 1), (182085, 4, 3, 1), (182082, 4, 3, 1), (182080, 4, 3, 1),
(182382, 6, 1, 1), (182378, 6, 1, 1), (182375, 6, 1, 1), (182373, 6, 1, 1),
(182372, 6, 1, 1), (182856, 6, 1, 1), (182849, 3, 3, 1), (182844, 4, 3, 1),
(182841, 4, 3, 1), (182839, 4, 3, 1), ), )

session.viewports['Viewport: 1'].odbDisplay.setPrimaryVariable(
    variableLabel='S', outputPosition=INTEGRATION_POINT,
refinement=(COMPONENT,
    'S11'))

xyp = session.xyPlots['XYPlot-1']
chartName = xyp.charts.keys()[0]
chart = xyp.charts[chartName]
pth = session.paths['Adhesive_Path']
xy1 = xyPlot.XYDataFromPath(path=pth, includeIntersections=False,
    projectOntoMesh=False, pathStyle=PATH_POINTS,
numIntervals=10,
    projectionTolerance=0, shape=DEFORMED,
labelType=TRUE_DISTANCE_X)

c1 = session.Curve(xyData=xy1)
chart.setValues(curvesToPlot=(c1, ), )
session.viewports['Viewport: 1'].setValues(displayedObject=xyp)
titlename=name + '_Adhesive_Path_S11'
pngname=name + '_Adhesive_Path_S11.png'
session.viewports['Viewport:
1'].setValues(titleStyle=CUSTOM,customTitleString=(titlename))

session.printToFile(fileName=pngname, format=PNG,
canvasObjects=(session.viewports['Viewport: 1'], ))

session.charts['Chart-
1'].legend.area.setValues(positionMethod=AUTOMATIC,
alignment=BOTTOM_CENTER)

session.viewports['Viewport: 1'].odbDisplay.setPrimaryVariable(
    variableLabel='S', outputPosition=INTEGRATION_POINT,
refinement=(COMPONENT,
    'S13'))

```

```

xyp = session.xyPlots['XYPlot-1']
chartName = xyp.charts.keys()[0]
chart = xyp.charts[chartName]
pth = session.paths['Adhesive_Path']
xy1 = xyPlot.XYDataFromPath(path=pth, includeIntersections=False,
                             projectOntoMesh=False, pathStyle=PATH_POINTS,
numIntervals=10,
                             projectionTolerance=0, shape=DEFORMED,
labelType=TRUE_DISTANCE_X)
c1 = session.Curve(xyData=xy1)
chart.setValues(curvesToPlot=(c1, ), )
session.viewports['Viewport: 1'].setValues(displayedObject=xyp)
titlename=name + '_Adhesive_Path_S13'
pngname=name + '_Adhesive_Path_S13.png'
session.viewports['Viewport:
1'].setValues(titleStyle=CUSTOM,customTitleString=(titlename))
session.printToFile(fileName=pngname, format=PNG,
canvasObjects=(session.viewports['Viewport: 1'], ))
session.charts['Chart-
1'].legend.area.setValues(positionMethod=AUTOMATIC,
alignment=BOTTOM_CENTER)

```

```

session.viewports['Viewport: 1'].odbDisplay.setPrimaryVariable(
    variableLabel='S', outputPosition=INTEGRATION_POINT,
refinement=(COMPONENT,
'S33'))
xyp = session.xyPlots['XYPlot-1']
chartName = xyp.charts.keys()[0]
chart = xyp.charts[chartName]
pth = session.paths['Adhesive_Path']
xy1 = xyPlot.XYDataFromPath(path=pth, includeIntersections=False,
                             projectOntoMesh=False, pathStyle=PATH_POINTS,
numIntervals=10,
                             projectionTolerance=0, shape=DEFORMED,
labelType=TRUE_DISTANCE_X)
c1 = session.Curve(xyData=xy1)
chart.setValues(curvesToPlot=(c1, ), )
session.viewports['Viewport: 1'].setValues(displayedObject=xyp)

```

```

titlename=name + '_Adhesive_Path_S33'
pngname=name + '_Adhesive_Path_S33.png'
session.viewports['Viewport:
1'].setValues(titleStyle=CUSTOM,customTitleString=(titlename))
session.printToFile(fileName=pngname, format=PNG,
canvasObjects=(session.viewports['Viewport: 1'], ))
session.charts['Chart-
1'].legend.area.setValues(positionMethod=AUTOMATIC,
alignment=BOTTOM_CENTER)
session.odbs[o1name].close()
countt=countt+1

```

APPENDIX E

```

from abaqus import *
from abaqusConstants import *
from caeModules import *
import visualization

phi=[0,15,30,45,60,75,90]
counter=0
phi_list=[]
for phi1 in phi:
    for phi2 in phi:
        for phi3 in phi:
            for phi4 in phi:
                counter=counter+1
                phi_list.append([counter,phi1,phi2,phi3,phi4])

begin=1
end=2401
path='E:/Master/Matlab_Workstation/Runner/Serhan/'
countt=begin-1

name='SLJ_n'+'%i'%phi_list[countt][0]+'_comp_shell_'+'%i'%phi_list[countt][1]+'_'
+ '%i'%phi_list[countt][2]+'_'+ '%i'%phi_list[countt][3]+'_'+ '%i'%phi_list[countt][4]
session.Viewport(name='Viewport: 1', origin=(0.0, 0.0), width=200, height=150)

```

```

session.viewports['Viewport: 1'].makeCurrent()
o1name=path + name +'.odb'
o1 = session.openOdb(name=o1name)
session.viewports['Viewport: 1'].setValues(displayedObject=o1)
odb = session.odbs[o1name]
session.viewports['Viewport:
1'].odbDisplay.display.setValues(plotState=(CONTOURS_ON_DEF, ))

session.Path(name='Adhesive_Path', type=EDGE_LIST, expression=((('ADHESIVE-
1', (
    (183535, 6, 1, 1), (183517, 6, 1, 1), (183510, 3, 3, 1), (183502, 5, 2, 1),
    (183494, 5, 2, 1), (182623, 5, 2, 1), (182618, 5, 2, 1), (182613, 3, 3, 1),
    (182609, 5, 2, 1), (182605, 5, 2, 1), (182240, 5, 2, 1), (182233, 5, 2, 1),
    (182227, 5, 2, 1), (182222, 5, 2, 1), (182217, 5, 2, 1), (182017, 4, 3, 1),
    (182011, 4, 3, 1), (182006, 4, 3, 1), (182002, 6, 1, 1), (181886, 5, 2, 1),
    (181885, 5, 2, 1), (181888, 3, 3, 1), (181877, 5, 2, 1), (181876, 5, 2, 1),
    (181879, 3, 3, 1), (181922, 5, 2, 1), (181921, 5, 2, 1), (181924, 3, 3, 1),
    (181913, 5, 2, 1), (181912, 5, 2, 1), (181915, 3, 3, 1), (181935, 6, 1, 1),
    (181932, 6, 1, 1), (181929, 6, 1, 1), (181928, 6, 1, 1), (182092, 6, 1, 1),
    (182088, 6, 1, 1), (182085, 4, 3, 1), (182082, 4, 3, 1), (182080, 4, 3, 1),
    (182382, 6, 1, 1), (182378, 6, 1, 1), (182375, 6, 1, 1), (182373, 6, 1, 1),
    (182372, 6, 1, 1), (182856, 6, 1, 1), (182849, 3, 3, 1), (182844, 4, 3, 1),
    (182841, 4, 3, 1), (182839, 4, 3, 1), )), ))

session.viewports['Viewport: 1'].odbDisplay.setPrimaryVariable(
    variableLabel='S', outputPosition=INTEGRATION_POINT,
    refinement=(COMPONENT,
        'S13'))
xyp = session.xyPlots['XYPlot-1']
chartName = xyp.charts.keys()[0]
chart = xyp.charts[chartName]
pth = session.paths['Adhesive_Path']
xy1 = xyPlot.XYDataFromPath(path=pth, includeIntersections=False,
    projectOntoMesh=False, pathStyle=PATH_POINTS, numIntervals=10,
    projectionTolerance=0, shape=DEFORMED,
    labelType=TRUE_DISTANCE_X)

```

```

c1 = session.Curve(xyData=xy1)
chart.setValues(curvesToPlot=(c1, ), )
session.viewports['Viewport: 1'].setValues(displayedObject=xyp)
titlename=name + '_Adhesive_Path_S13'
pngname=name + '_Adhesive_Path_S13.png'
session.viewports['Viewport:
1'].setValues(titleStyle=CUSTOM,customTitleString=(titlename))
session.printToFile(fileName=pngname, format=PNG,
canvasObjects=(session.viewports['Viewport: 1'], ))
session.charts['Chart-1'].legend.area.setValues(positionMethod=AUTOMATIC,
alignment=BOTTOM_CENTER)
#session.odbs[o1name].close()
LABEL="S, S13 (Avg: 75%): X Dist. along 'Adhesive_Path' - " + name
session.curves['_temp_1'].setValues(useDefault=False, legendLabel=LABEL)
tempdata_counter=2
while countt<end-49 :
    #2nci
    countt_2=countt+49
    name='SLJ_n'+'%i'%phi_list[countt_2][0]+'_comp_shell_'+'%i'%phi_list[cou
ntt_2][1]+'_'+'%i'%phi_list[countt_2][2]+'_'+'%i'%phi_list[countt_2][3]+'_'+'%i'%ph
i_list[countt_2][4]
    session.Viewport(name='Viewport: 1', origin=(0.0, 0.0), width=200,
height=150)
    session.viewports['Viewport: 1'].makeCurrent()
    o1name=path + name +'.odb'
    o1 = session.openOdb(name=o1name)
    session.viewports['Viewport: 1'].setValues(displayedObject=o1)
    odb = session.odbs[o1name]
    session.viewports['Viewport: 1'].odbDisplay.display.setValues(plotState=(
        CONTOURS_ON_DEF, ))

    session.viewports['Viewport: 1'].odbDisplay.setPrimaryVariable(
        variableLabel='S', outputPosition=INTEGRATION_POINT,
refinement=(COMPONENT,'S13'))
    xyp = session.xyPlots['XYPlot-1']
    chartName = xyp.charts.keys()[0]
    chart = xyp.charts[chartName]

```

```

pth = session.paths['Adhesive_Path']
xy1 = xyPlot.XYDataFromPath(path=pth, includeIntersections=False,
                             projectOntoMesh=False, pathStyle=PATH_POINTS,
numIntervals=10,
                             projectionTolerance=0, shape=DEFORMED,
labelType=TRUE_DISTANCE_X)
c1= session.Curve(xyData=xy1)
chart.setValues(curvesToPlot=(c1, ), appendMode=True)
session.viewports['Viewport: 1'].setValues(displayedObject=xyp)
#session.odbs[o1name].close()
LABEL="S, S13 (Avg: 75%): X Dist. along 'Adhesive_Path' - " + name
templabel='_temp_' + '%i'%tempdata_counter
session.curves[templabel].setValues(useDefault=False,
legendLabel=LABEL)

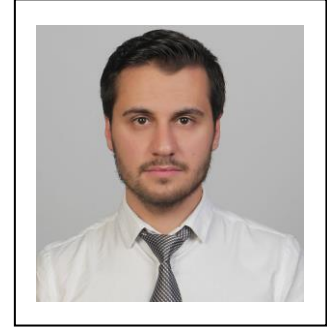
tempdata_counter=tempdata_counter+1
countt=countt+49

titlename= 'From49_Adhesive_Path_S13'
pngname='From49__Adhesive_Path_S13.png'
session.viewports['Viewport:
1'].setValues(titleStyle=CUSTOM,customTitleString=(titlename))
session.printToFile(fileName=pngname, format=PNG,
canvasObjects=(session.viewports['Viewport: 1'], ))
session.charts['Chart-1'].legend.area.setValues(positionMethod=AUTOMATIC,
alignment=BOTTOM_CENTER)

```



CURRICULUM VITAE



Name Surname : Ferhat Demir

Place and Date of Birth : 13/09/1990, Torul

E-Mail : ferhatdmir@gmail.com

EDUCATION :

- **B.Sc.** : 2012, Yildiz Technical University, Faculty of Mechanical Engineering , Mechanical Engineering

EXPERIENCE :

- **Turkish Aerospace Industries, INC.** Start Date (11/17) - Present
Design and Analysis Engineer, Design and Structural Department
- **SDM Research and Engineering** Start Date (06/14) - End Date (11/17)
Senior Mechanical Design Engineer, Design and Structural Department
 - Verifies structural integrity by using analytical methods, finite element models/simulations throughout the product lifecycle to develop the structural environment, characteristics and performance.
 - Construct meshed FEA models with Hypermesh from CAD, providing strength, thermal, sequential thermal-mechanical, modal, fatigue analyses with Abaqus and recommended improvements.
 - Support R&D projects with CAE support as necessary, participate in manufacturing process improvement and cost reduction initiatives as needed.

Projects

- General Electric / Thermal-Mechanical Analysis of Rotating Blade Assembly/ 2015

- Ford Otosan / Single Cylinder Research Engine /2016
 - It took approximately 2 years. It is the first domestic production in Turkey. All keys parts such as metal liner, drive shaft, conrod are designed and generated necessary structural and thermal analyses.
- Ford Otosan / Optical Single Cylinder Research Engine / 2016
 - It took approximately 2 years. It is the first domestic production in Turkey. All keys parts such as metal liner, drive shaft, conrod and elongated piston which is especially key part for optical research engine are designed and generated necessary structural and thermal analyses.
- Siemens / Transition Side Seal Project / 2017
- **Assan Hanil Otomotiv Sanayi A.Ş.** Start Date (09/13) - End Date (06/14)
 Research and Development Engineer, Design and Structural Department
Project
 - Driver seat tilt mechanism / Mechanical design and analysis/ 2014

SKILLS :

- **Language:** English (YDS – 88.75), German (Beginner)
- **Computer:** Catia, Hypermesh, Abaqus, MS Office Programs