

ANKARA YILDIRIM BEYAZIT UNIVERSITY
GRADUATE SCHOOL OF NATURAL AND APPLIED SCIENCES



**BUCKLING OF METALLIC SKIN-STRINGER AND
SANDWICH PANEL STRUCTURES:
A COMPARATIVE NUMERICAL STUDY**

M.Sc. Thesis by

Emre Apa

Department of Aerospace Engineering

February, 2024

ANKARA

**BUCKLING OF METALLIC SKIN-STRINGER AND
SANDWICH PANEL STRUCTURES:
A COMPARATIVE NUMERICAL STUDY**

A Thesis Submitted to

The Graduate School of Natural and Applied Sciences of

Ankara Yıldırım Beyazıt University

**In Partial Fulfilment of the Requirements for the Degree of Master of Science in
Aerospace Engineering, Department of Aerospace Engineering**

by

Emre Apa

February, 2024

ANKARA

M.Sc. THESIS EXAMINATION RESULT FORM

We have read the thesis entitled “**BUCKLING OF METALLIC SKIN-STRINGER AND SANDWICH PANEL STRUCTURES: A COMPARATIVE NUMERICAL STUDY**” completed by **EMRE APA** under the supervision of **ASST. PROF. DR. ORHAN ÖZÇELİK** and we certify that in our opinion it is fully adequate, in scope and in quality, as a thesis for the degree of Master of Science.

Asst. Prof. Dr. Orhan Özçelik

Supervisor

Prof. Dr. Ferhat Kadioğlu

Jury Member

Prof. Dr. Zeki Kırıl

Jury Member

Prof. Dr. Sadettin Orhan

Director

Graduate School of Natural and Applied Sciences

ETHICAL DECLARATION

I hereby declare that, in this thesis which has been prepared in accordance with the Thesis Writing Manual of Graduate School of Natural and Applied Sciences,

- All data, information and documents are obtained in the framework of academic and ethical rules,
- All information, documents and assessments are presented in accordance with scientific ethics and morals,
- All the materials that have been utilized are fully cited and referenced,
- No change has been made on the utilized materials,
- All the works presented are original,

and in any contrary case of above statements, I accept to renounce all my legal rights.

Date: 05.02.2024

Signature:

Name & Surname: Emre Apa

ACKNOWLEDGMENTS

I would like to express my sincere gratitude to my supervisor, Asst. Prof. Dr. Orhan Özçelik for his invaluable guidance, enormous support, and insightful feedback throughout the entire journey of my master's thesis. His precious knowledge and recommendations played an important role in shaping the direction and quality of this work.

I am also profoundly thankful to my family for their understanding, patience, and support. Their belief in me has been a constant source of motivation for my academic life.

This achievement is not solely mine; it is a collective effort of the exceptional individuals who have been part of my academic and personal journey. I want to express my gratitude to all people who give motivation and support me for being the pillars of strength in my pursuit of this milestone.

2024, 05 February

Emre Apa

BUCKLING OF METALLIC SKIN-STRINGER AND SANDWICH PANEL STRUCTURES: A COMPARATIVE NUMERICAL STUDY

ABSTRACT

In this study, global buckling behaviour of the metallic skin-stringer and honeycomb sandwich panels, which are used on the helicopter fuselage sections, is analysed. The metallic skin-stringer having different stringer cross sectional areas (z-, c-, and omega-section) and the metallic honeycomb sandwich panels in three variants (panels with longitudinal doublers, with transverse doublers, and without doublers) are examined with finite element simulations. The linear eigenvalue buckling analysis is used to calculate the critical global buckling load for simply supported boundary conditions.

The most durable skin-stringer panel against the global buckling failure is determined to be the panel reinforced with the omega stringers. The panel with z stringers is found to be the least durable configuration against global buckling. The buckling strength of the panel with omega stringer is 13.1 and 15.6 percent higher than the panels with c and z stringers, respectively. The most durable honeycomb sandwich panel configuration against global buckling is determined to be the panel with transverse doublers. The panel without any doublers is found to be the least durable configuration against global buckling. The buckling strength of the panel with transverse doublers is 11.8 and 12.6 percent higher than the panels with longitudinal doublers and without any doublers, respectively.

Honeycomb sandwich panels exhibit higher buckling strength compared to the skin-stringer panels. Critical buckling loads of the sandwich panels per kilogram are greater than those of the skin-stringer panels. Considering the structural, manufacturing, and environmental requirements, metallic honeycomb sandwich panels can be utilized as side panels of a helicopter fuselage and they can give better buckling performance per unit weight. Two panel concepts also can be used on different sections of a helicopter fuselage as long as they meet the structural requirements.

Keywords: Metallic stiffened panel, metallic sandwich panel, global buckling phenomena, linear eigenvalue buckling analysis, stringer, honeycomb, aerostructures.

METALİK KİRİŞ-DIŞ PANEL VE BAL PETEĞİ SANDVIÇ PANEL YAPILARININ BURKULMASI: KARŞILAŞTIRMALI NÜMERİK BİR ÇALIŞMA ÖZ

Bu çalışmada, helikopter gövde kısımlarında kullanılan metalik kiriş-dış panel ve bal peteği sandviç panellerinin genel burkulma davranışı analiz edilmiştir. Farklı kiriş kesit alanlarına (z-, c- ve omega-kesiti) sahip metalik kiriş-dış ve üç varyanta sahip metalik bal peteği sandviç paneller (boyuna güçlendiricili, enine güçlendiricili ve güçlendiricisiz paneller) sonlu eleman simülasyonlarıyla incelenmiştir. Doğrusal özdeğer burkulma analizi, basit mesnetli sınır koşulları için kritik genel burkulma yükünü hesaplamak amacıyla kullanılmıştır.

Genel burkulmaya karşı en dayanıklı kiriş-dış panelin omega kirişlerle güçlendirilmiş panel olduğu belirlenmiştir. Z kirişli panelin genel burkulmaya karşı en az dayanıklı konfigürasyon olduğu bulunmuştur. Omega kirişli panelin burkulma mukavemeti c ve z kirişli panellere göre sırasıyla yüzde 13,1 ve 15,6 daha yüksektir. Genel burkulmaya karşı en dayanıklı bal peteği sandviç panel konfigürasyonu enine güçlendiricili panel olarak belirlenmiştir. Güçlendirici içermeyen panelin genel burkulmaya karşı en az dayanıklı konfigürasyon olduğu bulunmuştur. Enine güçlendiricili panelin burkulma mukavemeti, boyuna güçlendiricili ve güçlendiricisiz panellerden sırasıyla yüzde 11,8 ve 12,6 daha yüksektir.

Bal peteği sandviç paneller, kiriş-dış panellere kıyasla daha yüksek burkulma mukavemeti sergiler. Bal peteği sandviç panellerin kilogram başına kritik burkulma yükleri, kiriş-dış panellerinkinden daha fazladır. Metalik bal peteği sandviç paneller; yapısal, imalat ve çevresel gereksinimler göz önüne alındığında, helikopter gövdesinin yan panelleri olarak kullanılabilir ve birim ağırlık başına daha iyi burkulma performansı sağlayabilir. Yapısal gereklilikleri karşıladığı sürece helikopter gövdesinin farklı bölümlerinde iki panel konsepti de kullanılabilir.

Anahtar Kelimeler: Metalik güçlendirmeli panel, metalik sandviç panel, genel burkulma davranışı, doğrusal özdeğer burkulma analizi, kiriş, bal peteği, hava yapıları.

CONTENTS

M.Sc. THESIS EXAMINATION RESULT FORM	ii
ETHICAL DECLARATION	iii
ACKNOWLEDGMENTS	iv
ABSTRACT	v
ÖZ	vi
CONTENTS	vii
NOMENCLATURE	x
LIST OF TABLES	xiii
LIST OF FIGURES	xiv
 CHAPTER 1 - INTRODUCTION	 1
1.1 Motivation of the Study	1
1.2 Literature Review	3
1.2.1 Stiffened Panel Structure Literature Review	3
1.2.2 Sandwich Panel Structure Literature Review	9
1.3 Scope of the Study	13
1.4 Content of the Study	14
 CHAPTER 2 – THEORY OF BUCKLING	 16
2.1 Buckling Phenomena and Column Buckling Theory	16
2.1.1 Meaning of Buckling	16
2.1.2 Types of Buckling Analysis	19
2.1.2.1 Linear Buckling Analysis	19
2.1.2.2 Nonlinear Buckling Analysis	23
2.1.3 General Types of Buckling Failure Mode	24
2.1.3.1 Local Buckling	24
2.1.3.2 Global Buckling	26
2.1.4 Buckling and Bending Phenomena	26
2.1.5 Column Buckling Theory	27
2.2 Fundamental Theory of Flat Thin Sheet Buckling	28
2.3 Fundamental Theory of Stiffened Panel Buckling	38
2.3.1 Buckling Behaviour of Stiffened Panel	40
2.3.2 Buckling Behaviour of Stiffeners	43

2.4 Fundamental Theory of Sandwich Panel Buckling.....	46
2.4.1 Global or Overall Buckling for Sandwich Panels.....	47
2.4.2 Local Buckling for Sandwich Panels.....	50
CHAPTER 3 – METALLIC PANEL DESIGN	53
3.1 Metallic Stiffened Panel Design.....	53
3.1.1 General Overview of Loads on Metallic Panels	57
3.1.2 Material Choice for Metallic Panels	59
3.1.3 Manufacturing Methods for Metallic Panels	60
3.1.4 Metallic Stringer Design.....	61
3.1.4.1 Different Cross Sections of Stringers	62
3.1.5 Metallic Frame Design	63
3.1.5.1 Different Cross Sections of Frames	65
3.2 Metallic Sandwich Panel Design.....	66
3.2.1 Material and Manufacturing Methods for Sandwich Structures.....	69
3.3 Metallic Panel Designs of the Study	70
3.3.1 Z Cross Section Stringer-Skin Metallic Panel	70
3.3.2 Omega Cross Section Stringer-Skin Metallic Panel.....	71
3.3.3 C Cross Section Stringer-Skin Metallic Panel.....	73
3.3.4 Metallic Honeycomb Sandwich Panel.....	74
3.3.5 Metallic Honeycomb Sandwich Panel with Longitudinal Doublers	75
3.3.6 Metallic Honeycomb Sandwich Panel with Transverse Doublers	76
3.3.7 Summary of the Geometrical Properties and Material Properties	77
CHAPTER 4 – NUMERICAL ANALYSIS.....	80
4.1 General Numerical Analysis Phenomena.....	80
4.2 Linear Eigenvalue Buckling Analysis Method	81
4.2.1 Extraction Methods for Eigenvalues	84
4.2.1.1 The Lanczos Method.....	84
4.2.1.2 The Inversed Power Method	84
4.2.1.3 The Enhanced Inversed Power Method	84
4.2.1.4 Summary of the Extraction Methods of Eigenvalues	85
4.3 Finite Element Analysis on Buckling.....	85
4.3.1 Model Description and Mesh	86
4.3.1.1 Metallic Skin-Stringer Panel Model Description and Mesh.....	87

4.3.1.2 Metallic Honeycomb Sandwich Panel Model Description and Mesh.....	90
4.3.2 Boundary Conditions and Loads	94
CHAPTER 5 – RESULTS AND DISCUSSION.....	98
5.1 General Approach to Linear Eigenvalue Buckling Analysis	98
5.2 Buckling Analysis of Metallic Flat and Curved Skin Panels	99
5.2.1 Buckling Analysis of Metallic Flat Skin Panel	99
5.2.1 Buckling Analysis of Metallic Curved Skin Panel.....	101
5.3 Buckling Analysis of Metallic Skin-Stringer Panels.....	103
5.3.1 Buckling Analysis of Metallic Z Cross Section Stringer-Skin Panel....	103
5.3.2 Buckling Analysis of Metallic Omega Cross Section Stringer-Skin Panel.....	105
5.3.3 Buckling Analysis of Metallic C Cross Section Stringer-Skin Panel....	107
5.4 Buckling Analysis of Metallic Honeycomb Sandwich Panels.....	109
5.4.1 Buckling Analysis of Metallic Honeycomb Sandwich Panel without Doublers.....	109
5.4.2 Buckling Analysis of Metallic Honeycomb Sandwich Panel with Longitudinal Doublers	110
5.4.3 Buckling Analysis of Metallic Honeycomb Sandwich Panel with Transverse Doublers	112
CHAPTER 6 – CONCLUSION AND FUTURE WORK.....	115
6.1 Comparison of Buckling Behaviour of Metallic Skin-Stringer Panel Structures.....	115
6.2 Comparison of Buckling Behaviour of Metallic Honeycomb Sandwich Panel Structures.....	116
6.3 Overall Summary and Future Work	118
REFERENCES.....	121
CURRICULUM VITAE.....	128

NOMENCLATURE

Roman Letter Symbols

$P_{cr,e}$	Critical Euler elastic buckling load, N
P	Compressive force, N
E	Modulus of elasticity, N/mm ²
I_c	Moment of inertia of column, kg·m ²
L_c	Column length, mm
k	Column buckling constant
$k_{cr,x}$	Buckling coefficient for thin plates/panels
w	Lateral deflection, mm
N_x	Resultant force in x direction, N
$N_{x,cr}$	Critical resultant force in x direction, N
N_y	Resultant force in y direction, N
N_{xy}	Resultant force in x-y plane, N
N_{yx}	Resultant force in x-y plane, N
D	Flexural rigidity of a plate/panel, Pa·m ³
W	Virtual displacement, mm
m	Number of buckling half wave in x direction
n	Number of buckling half wave in y direction
δV	Increment of strain energy
a	Long dimension of panel/plate, mm
b	Short dimension of panel/plate or loaded edge, mm
t	Sheet thickness for a thin panel/plate, mm
k_y	Buckling coefficient for uniform transverse compressive loading
c_l	Loading ratio
w_{long}	Local buckling mode in longitudinal direction for a stiffened panel/plate
w_{trans}	Local buckling mode in transverse direction for a stiffened panel/plate
$w_{overall}$	Overall buckling mode for a stiffened panel/plate
A	Cross section area, mm ²
I_e	Moment of inertia of the panel, kg·m ²
L	Length of a panel/plate under combined biaxial loading, mm
B	Width of a panel/plate under combined biaxial loading, mm

H	Effective torsional rigidity, $N \cdot m^2$
w_{stiff}	Stiffener deflection mode in transverse direction
y	Lateral deflection of stiffener web, mm
D_w	Bending rigidity of stiffener web, $N \cdot m^2$
t_w	Thickness of stiffener web, mm
k_w	Buckling strength coefficient of stiffener web
h_w	Short dimension of stiffener web, mm
w_{web}	Local buckling mod of a stiffened panel/plate in terms of stiffener web
h_{web}	Height of a stiffened web, mm
d_{web}	Deflection of a stiffened web, mm
V_i	Deflection components of a stiffened web, mm
C	Panel Coefficient for sandwich plates/panels
D_{ij}	Elements of flexural stiffness matrix
K_0	Buckling coefficient for sandwich plates/panels
t_c	Equivalent thickness for sandwich plates/panels, mm
t_b	Thickness of bottom face plate for sandwich plates/panels, mm
t_t	Thickness of top face plate for sandwich plates/panels, mm
I_1	Moment of inertia along longitudinal axis for sandwich plates/panels, $kg \cdot m^2$
I_2	Moment of inertia along transverse axis for sandwich plates/panels, $kg \cdot m^2$
b_c	Width of plates between two core elements, mm
h	Core height, mm
E_c	Young's modulus of homogenized core, N/mm^2
K^{AB}	Tangent stiffness matrix
v^A	Nontrivial displacement solution vector
K_0^{BA}	Stiffness matrix
i	Related buckling number of the buckling modes
K_v^{BA}	Initial differential stress and load stiffness matrix
Q^B	Applied force for linear buckling phenomena, N
P^B	Critical buckling load for linear buckling phenomena, N

Greek Letter Symbols

ν	Elastic Poisson's ratio
σ_x	Stress in x direction, N/mm ²
σ_y	Stress in y direction, N/mm ²
σ_E^W	Elastic buckling strength of stiffener web, N/mm ²
ξ_w	Torsional rigidity of the plating, N·m ²
σ_0	Plate material yield stress for sandwich plates/panels, N/mm ²
λ_i	Eigenvalues for linear buckling phenomena
v_i^A	Eigenvectors or the buckling mode shapes

Acronyms

FE	Finite Elements
C	Clamped or fixed boundary edge condition
F	Free boundary edge condition
SS	Simply supported or hinged boundary edge condition
MPC	Multi point constraints
RBE	Rigid body elements
3D	Three dimensional
EIGRL	Lanczos method
INV	Inverse power method
SINV	Enhanced inversed power method
U	Translation boundary condition
R _{ot}	Rotation boundary condition
N	Newton
kg	Kilogram
pcf	Pounds per cubic feet

LIST OF TABLES

Table 3.1 Geometrical and material properties of metallic panels	78
Table 4.1 Number of elements for each skin-stringer panel	88
Table 4.2 Number of elements for each honeycomb sandwich panel	93
Table 4.3 Boundary Conditions.....	95
Table 5.1 Comparison between analytical and numerical solution for metallic flat panel	100
Table 5.2 Comparison critical buckling loads between flat skin panel and curved skin panel	102
Table 6.1 Critical global buckling loads for each stringer-skin panel.....	115
Table 6.2 Critical global buckling loads for each metallic honeycomb sandwich panel.....	116
Table 6.3 Critical global buckling loads per kg for each panel.....	118

LIST OF FIGURES

Figure 1.1 Metallic skin-stringer panel structure	1
Figure 1.2 Metallic honeycomb sandwich panel structure.....	2
Figure 2.1 Stages of equilibrium	16
Figure 2.2 Buckling strength vs slenderness curve	18
Figure 2.3 Elastic and plastic regions of buckling	18
Figure 2.4 Linear buckling cases.....	20
Figure 2.5 Bifurcation points for linear buckling analysis.....	21
Figure 2.6 Plate deflection	22
Figure 2.7 Nonlinear buckling behaviour	23
Figure 2.8 General buckling failure modes.....	24
Figure 2.9 Local skin buckling.....	25
Figure 2.10 Local stringer buckling	25
Figure 2.11 Difference between bending and buckling	27
Figure 2.12 Moment equilibrium for thin plates 1	30
Figure 2.13 Moment equilibrium for thin plates 2	30
Figure 2.14 A plate under in-plane uniform compression	32
Figure 2.15 Critical load factor for buckling.....	34
Figure 2.16 General boundary conditions of buckling situation.....	35
Figure 2.17 Buckling coefficient with aspect ratios and boundary conditions	36
Figure 2.18 A plate under in-plane biaxial compression	37
Figure 2.19 Stiffened panel under combined in-plane loads and lateral pressure.....	39
Figure 2.20 Buckling modes of stiffened plate under compression.....	41
Figure 2.21 Buckling modes of stiffener in stiffened plate.....	43
Figure 2.22 Stiffener web local buckling	44
Figure 2.23 Sandwich plate subjected to uniaxial compression.....	47
Figure 2.24 Buckling failure modes for a sandwich panel.....	47
Figure 2.25 Buckling coefficient for an orthotropic plate.....	50
Figure 3.1 Example of a fuselage metallic panel	54
Figure 3.2 A stiffened panel	54
Figure 3.3 Airbus Beluga single and double curve panels.....	55
Figure 3.4 One-way stiffened panel	56
Figure 3.5 Two-way stiffened discontinues stringers panel.....	56

Figure 3.6 Two-way stiffened continues stringers panel	57
Figure 3.7 Skin wrinkling	58
Figure 3.8 Compression buckling	59
Figure 3.9 Typical stringer profile	61
Figure 3.10 Different kinds of stringer types	63
Figure 3.11 Typical fuselage section with frames and stringers	64
Figure 3.12 Different frame section profiles	65
Figure 3.13 Definition of honeycomb sandwich structure panel	66
Figure 3.14 Honeycomb structure	67
Figure 3.15 Effect of core thickness on sandwich panel stiffness	68
Figure 3.16 Z cross section stringer-skin panel.....	71
Figure 3.17 General dimensions of z stringer	71
Figure 3.18 Omega cross section stringer-skin panel	72
Figure 3.19 General dimensions of omega stringer	72
Figure 3.20 C cross section stringer-skin panel	73
Figure 3.21 General dimensions of c stringer	74
Figure 3.22 Metallic honeycomb sandwich panel.....	75
Figure 3.23 Metallic honeycomb sandwich panel with longitudinal doublers	76
Figure 3.24 Metallic honeycomb sandwich panel with transverse doublers	77
Figure 3.25 Material specifications of Al 2024 T3	78
Figure 3.26 Material specifications of aluminium honeycomb core 5052.....	79
Figure 4.1 Shell elements with different nodes.....	86
Figure 4.2 Z cross section stringer-skin assembly mesh view	88
Figure 4.3 Omega cross section stringer-skin assembly mesh view	88
Figure 4.4 C cross section stringer-skin assembly mesh view.....	89
Figure 4.5 TIE contact property between Z cross section stringer and skin	90
Figure 4.6 Metallic honeycomb sandwich panel without doublers.....	93
Figure 4.7 Metallic honeycomb sandwich panel with longitudinal doublers	94
Figure 4.8 Metallic honeycomb sandwich panel with transverse doublers.....	94
Figure 4.9 Boundary conditions	96
Figure 4.10 Applied loads on panel structure	97
Figure 5.1 Linear static analysis results for flat skin panel.....	99
Figure 5.2 Eigenvalues of flat skin panel.....	100
Figure 5.3 Linear static analysis results for curved skin panel	101

Figure 5.4 Eigenvalues of curved skin panel	102
Figure 5.5 Linear static analysis results for z cross section stringer-skin panel	103
Figure 5.6 Eigenvalues of z cross section stringer-skin panel	104
Figure 5.7 Linear static analysis results for omega cross section stringer-skin panel.....	106
Figure 5.8 Eigenvalues of omega cross section stringer-skin panel	106
Figure 5.9 Linear static analysis results for c cross section stringer-skin panel	108
Figure 5.10 Eigenvalues of c cross section stringer-skin panel	108
Figure 5.11 Linear static analysis results for honeycomb sandwich panel without doublers	109
Figure 5.12 Eigenvalues of honeycomb sandwich panel without doublers	110
Figure 5.13 Linear static analysis results for honeycomb sandwich panel with longitudinal doublers.....	111
Figure 5.14 Eigenvalues of honeycomb sandwich panel with longitudinal doublers.....	112
Figure 5.15 Linear static analysis results for honeycomb sandwich panel with transverse doublers.....	113
Figure 5.16 Eigenvalues of honeycomb sandwich panel with transverse doublers	113

CHAPTER 1

INTRODUCTION

1.1 Motivation of the Study

Aluminium alloys have consistently served as the predominant material for structural components in aircraft. These alloys are applied in diverse sections of both commercial and military aircraft, encompassing the fuselage, wings, and supporting structures. The preference for aluminium is rooted in considerations such as performance characteristics, cost-effectiveness, adherence to design specifications, and feasibility in manufacturing. These factors play a pivotal role in the utilization of aluminium for vital aircraft elements like frames, longerons, stringers, panels, and skins, tasked with bearing specific loads and meeting design criteria.

This study primarily concentrates on the buckling behavior observed in two key types of aircraft panel structures: metallic skin-stringer panels and metallic honeycomb sandwich panels. These panels serve as essential structural components in various aircraft platforms, offering significant advantages in terms of both weight and stiffness. They are found predominantly in fuselage and wing structures and these panel configurations are widely prevalent in the aviation industry. Illustrated in Figure 1.1, metallic skin-stringer panels are constructed with thin face sheets supported by various types of stringers.



Figure 1.1 Metallic skin-stringer panel structure [1]

As depicted in Figure 1.2, metallic honeycomb sandwich panels exhibit noteworthy lightweight characteristics in contrast to alternative structures like metallic alloys, which are frequently employed in aerospace engineering. These sandwich panel structures are extensively utilized in aircraft platforms where elevated mechanical properties are essential.

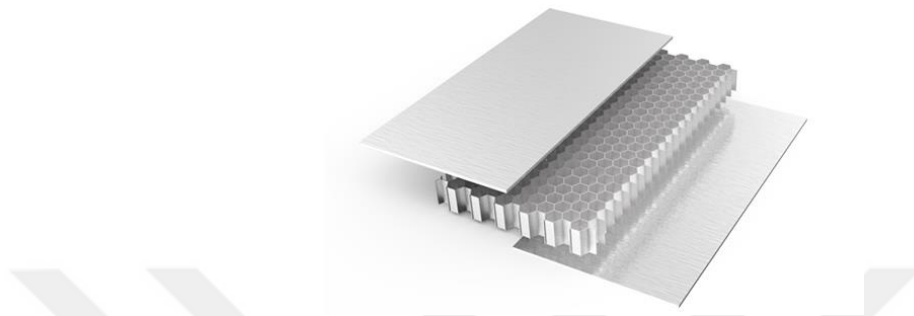


Figure 1.2 Metallic honeycomb sandwich panel structure [2]

Studying both structures is crucial, as the overall structural capacity and integrity hinge on their interplay. Furthermore, the interconnected behaviours of these structures exert a mutual influence, shaping the dynamics of the surrounding framework.

Both metallic honeycomb sandwich panels and metallic skin-stringer panels, exhibit a heightened susceptibility to buckling owing to predominantly in-plane loads and thin plating. Consequently, the primary focus is on identifying the critical buckling loads for these structures to ensure their safe and efficient performance.

Buckling, stemming from compression, shear, or their interplay, represents the usual load scenarios for fuselage or wing structures. In aviation, there is a defined threshold of acceptable buckling that structures can undergo before reaching ultimate load conditions. Buckling predominantly occurs in structures subjected to compressive forces, posing a potential compromise to the integrity of structural elements under such loads.

Examining the stringer sections in metallic skin-stringer panel structures is crucial for understanding the buckling load conditions offered by the stringers to the skin face

sheet. Similarly, optional doublers in metallic honeycomb sandwich panel structures play a role in determining the critical buckling loads for the entire assemblies.

Although numerous studies on stiffened panel structures exist in the literature, this study is managed by myself due to the deprivation of research studies that specify the most efficient types of stringer cross sections concerning linear buckling behaviours. Additionally, there is a gap in studies addressing the linear buckling behavior of metallic honeycomb sandwich panels, prompting a focused investigation in this area.

In addition, another aim of this study is to inspect the buckling behavior of aerospace panels which are made from aluminium alloys since one of the biggest aerospace companies in Turkey would like to standardize its own panel design processes.

Furthermore, research in the literature reveals numerous boundary conditions that may not be sufficiently advantageous for aerospace structures. To address this gap, this study applies the closest boundary conditions to the panel structures in question.

The outcomes of this study have the potential to identify optimal design solutions for both panel structures, considering factors such as mechanical performance, weight savings, manufacturability, and installation processes. As is widely recognized, these four criteria, along with others like maintainability, materials, and processes, collectively contribute to enhancing the overall performance of the aircraft platform.

1.2 Literature Review

Two distinct literature surveys that are related to buckling of stiffened and sandwich panels are given under this subtitle.

1.2.1 Stiffened Panel Structure Literature Review

Aluminium alloys have emerged as the primary material of preference for the construction of structural components within aircraft. These alloys find extensive application in the fuselage, wings, and support framework of both commercial and military aircraft. The utilization of aluminium is rooted in its advantageous performance attributes, cost-effective manufacturing, alignment with design

requirements, and compatibility with various manufacturing methodologies. This makes aluminium the material of choice for essential aircraft components, encompassing elements like frames, longeron, stringers, panels, and skin. These components are pivotal in bearing specific loads and fulfilling stringent design criteria, thus underlining the significance of aluminium in airframe construction.

Aluminium alloys have secured their position in pivotal locations and components of aircraft ever since the aviation industry diverged from other engineering sectors. Among these applications, the utilization of aluminium alloys is notably evident in the construction of skin-stringer panel structures. In such configurations, a stringer serves as a longitudinal structural element within the framework of the aircraft, while the skin forms the outer surface enveloping the wings and fuselage. These fuselage structures are effectively fortified through the integration of longitudinal and transverse aluminium structural elements, thereby reinforcing the overall structural integrity of the aircraft.

In conventional aircraft design practices, the incorporation of stringers aligned in the longitudinal direction and frames arranged in the circumferential direction constitutes a fundamental strategy for reinforcing the skin of the aircraft. These stringers are commonly affixed to the skin through techniques like riveting, bonding, and other fastening methods. Stringers serve as pivotal components in preventing various failure modes, particularly the onset of buckling. Buckling, a phenomenon that can manifest due to compression, shear, or a combination of these forces, represents typical load scenarios encountered by fuselage structures. When it comes to aircraft design, a certain level of controlled buckling is deemed acceptable within the framework of ultimate load considerations. Consequently, precise analyses become imperative for accurately characterizing the behavior of stiffened panels within even the post-buckling regime.

Research on plate buckling commenced in the early 19th century, with Claude-Louis Navier being credited for deriving the stability equation for a rectangular thin plate. This derivation, dating back to 1822, is rooted in the assumptions laid out by Gustav Robert Kirchhoff [3].

Bryan, in 1891, formulated the critical buckling stress equation for a rectangular thin plate subjected to uniaxial compression load and featuring a simply supported edge condition [3].

Timoshenko and Gere addressed the buckling problem using an alternative method. Their assumption involved the plate buckling into multiple sinusoidal half waves in the direction of compression. Additionally, they explored the buckling behavior of uniformly compressed rectangular plates, featuring simple support along the edges perpendicular to the direction of the applied load. Simultaneously, the other two edges were subjected to various end conditions [4].

Timoshenko and Gere's research highlights that alternative stringer configurations, such as diagonal stringers, exhibit superior buckling behavior compared to a conventional rectangular stringer stiffened structure. Timoshenko and Gere provide a solution for an infinitely long isotropic simply supported plate, with the result being approximately 6.5% higher than the exact solution. Consequently, employing this approach may entail an expected error of this magnitude [4].

Previous works by Stein have demonstrated that diagonal stiffeners possess distinct advantages over conventional rectangular stringers, particularly in terms of buckling resilience [5]. Additionally, Kromm has conducted similar research that explains critical shear stress of rectangular plates with diagonal steel bracing in this field [6]. Lundquist and Stowell utilized this approach for infinitely long isotropic flat rectangular plates with elastically restrained longitudinal edges [7]. Bedair and Sherbourne adopted the same technique in their approach [8]. Thielemann presented approximate solutions for boundary conditions involving free, clamped, and simply supported edges specifically for orthotropic materials and he added to the special reference in his study [9].

Klinzmann and Horst conducted a buckling analysis of cylindrical panels stiffened in a triangular pattern, exploring their application in aircraft structures. They also investigated the integral stiffening concept, identifying its potential for reducing weight [10]. They adopted an integrated approach by employing a triangular skin panel configuration with integral stringers for their study. They subsequently contrasted this

design with a traditional fuselage structure featuring a conventional stringer-frame arrangement, with the stringers being characterized by an I cross-section. The design framework was established based on a rectangular reference structure, ensuring that both fuselage configurations maintained equivalent cross-sectional dimensions and volume. In the case of Klinzmann and Horst, Aluminium Alloy 2024 was employed for each stringer component. Assessing the buckling resistance of stiffened plates using intricate numerical models can yield highly accurate outcomes. However, for the initial design stages of aircraft structures, this approach might prove impractical due to the necessity for numerous iterations or sensitivity analyses. Despite being relatively advanced, two distinct types of buckling analyses can be carried out using computer software's numerical methods: a linear elastic stability analysis and a nonlinear stability analysis. Differences exist between these two types of buckling analysis. Linear elastic stability analysis revolves around the theoretical buckling strength of an ideal linear elastic structure. However, real structures are hindered from attaining this theoretical strength due to imperfections, geometrical deviations, and physical nonlinearities. These factors contribute to the need for a nonlinear analysis to describe such effects accurately. Klinzmann and Horst's study revealed that triangular structures exhibit distinct advantages over rectangular stiffened fuselage structures in terms of tensile, shear, and compression stress distribution. They noted, however, that advancements in manufacturing technologies could potentially lead to more refined conclusions [10].

Pflüger provides an overview of analytical solutions for specific structures, considering diverse boundary conditions and load combinations [11]. Pflüger highlights the impact of boundary conditions on the buckling factor, emphasizing that the critical buckling stress in compression or shear can be calculated using Timoshenko's equation. [11].

In Paul et al.'s study, they focused on a standard transport aircraft wing and conducted a buckling analysis. The initial design showed susceptibility to buckling, prompting several design modifications to enhance its resilience. The study employed both finite element analysis and theoretical approaches to ensure realistic results in the wing buckling analysis [3].

Wickens, Odermann, and Kling conducted a study about benchmarking the buckling and post buckling behaviour of the finite element analysis results against experiments for j-stringer stiffened curved panels under in-plane loading conditions [12]. They have used carbon fiber reinforced polymer panels to carry out their studies and their study has shown that the results of numerical and experimental solution matches. Their study has verified usage of the modified test rig under compression and shear loading. Additionally, Bai et al. explored j-stiffened panels through both experimental and numerical investigations [13]. In the context of shear buckling, Seydel enhanced the approach for isotropic simply supported plates [14].

Budiansky, Connor, and Stein utilized the Lagrange multiplier method to integrate robust adaptor supports for isotropic materials [15]. Additionally, Rockey and Cook contributed to this domain by developing various models for differently stiffened isotropic panels [16].

Paik and Thayamballi have shown a buckling behaviour for isotropic stiffened plate. The problem involving differential equations is analogous to the buckling problem of a plate with elastic restraints. This issue is resolved using the Kantorovich method, and a precise determination of the critical local buckling load is obtained through exact calculations [17].

Muameleci's research explores both linear and nonlinear buckling analyses of plates, with and without a 9 cut-out, employing the finite element method. Different classical boundary conditions and loading scenarios are applied to model shear web beam structures. This study centres on exploring the buckling behavior of plates and evaluating the capacities of the MSC Nastran and ABAQUS finite element tools in performing both linear and nonlinear buckling analyses [18].

Bisagni and Vescovini enhanced a formulation for the buckling behavior of isotropic materials, specifically addressing the issue of shear-loaded omega-stringer stiffened panels [19]. A parallel effort was undertaken by Stamatelos, Labeas, and Tserpes, employing the Ritz displacement function [20].

Shama conducted investigations on stiffened panels experiencing in-plane combined forces, using isotropic materials [21].

Mantzaroudis and Stamatelos emphasized the significant importance of determining rotational stiffness, as it signifies the impact of stringers on the skin. They further highlighted a closed-form solution derived from the Rayleigh-Ritz approach [22]. Therefore, two different mathematical modelling approaches might be used. First mathematical model was used by Kollar and Springer and they have shown that a single skin-stringer assembly is isolated from the multi-stringer assembly in order to solve the local buckling problem for a hollow box beam and for a beam with a double-T cross section [23]. Another method is explained by Mittelstedt and Beerhost. This method concludes a unit-length frame of the cross section of the stringer [24].

Ullah et al. introduced alternative analytical approaches employing a two-dimensional generalized finite integral transform method for shear buckling analysis in fully clamped isotropic plates subjected to shear loading [25].

Yu conducted a study on the buckling behavior of rectangular plates under mid and end loads, encompassing considerations for both elastic and plastic buckling behavior. The study also examines the influence of different plate aspect ratios, and boundary types on buckling. The considered plate is simply supported along two opposite edges parallel to the direction of applied loads while the remaining two edges can assume any combination of clamped, simply supported, and free edge boundary conditions [26].

Britt and Simmons have examined failure analysis of large stiffened metallic fuselage panels subjected to compression loading [27]. They successfully projected both the buckling load and the ultimate failure load of a metallic stiffened panel for the Gulfstream V business jet fuselage, employing both theoretical and experimental methods. The panel configuration encompasses four frames and five stringers, along with a single skin. These structural elements, namely frames and stringers, are interconnected using rivets. Nevertheless, distinct panel configurations were studied, characterized by variations in material thickness and the configuration of skin, frame, and stringer components in terms of shape, orientation, and attachment method. In the

case of each panel, both the skin and doublers are constructed from 2024-T3 clad aluminium, while the stringers, frames, and shear ties that link the frame to the skin are fashioned from 7075-T62 clad aluminium. A comprehensive evaluation of each panel is conducted with a focus on diverse failure modes. Notably, experimental observations indicate that the failure load generally escalates as the cross-sectional area increases. This trend aligns with the findings obtained through finite element methods, which incorporate both linear elastic stability analysis and nonlinear stability analysis, carried out concurrently.

Mateus conducted a study on the influence of a multi-load scenario on various geometric thin-walled structures, evaluating their ultimate strength capacity. The structural behavior of thin-walled panels under identical geometric cases and loading conditions was assessed, and Mateus performed a comparison of the results [28].

Georgiou performed to assess existing buckling models of stiffened plate as proposed by ship design criteria and constituted a combination of analytical work and numerical analyses using non-linear FE tool. Comparison of eigenvalues and the second, on ultimate capacity over a relevant parameter span such as plate thickness, stiffener dimensions etc. as typical for ship designs were evaluated by Georgiou [29].

1.2.2 Sandwich Panel Structure Literature Review

Sandwich structures stand out as notably lightweight components when contrasted with other structures like metallic alloys in aerospace engineering. The utilization of metallic sandwich panel structures is prevalent in aircraft platforms where elevated mechanical properties are necessary. Within this category, a diverse range of configurations, materials, and mechanical characteristics exist. The selection of appropriate sandwich materials hinges on multiple factors including the intended purpose of the structure, the anticipated loading over its operational lifespan, material availability, and cost considerations.

Broadly speaking, sandwich structures consist of five distinct components. These components encompass an upper sheet situated on the outer surface of the sandwich structure, a lower sheet positioned on the inner surface of the structure, a metallic core

positioned between the upper and lower sheets, epoxy glass that envelops the entirety of the sandwich structure to provide protection against corrosion, and a doubler element that imparts additional strength to the structure, a particularly relevant consideration within the aviation industry.

During the design phase of an aircraft platform, paramount consideration must be given to ensuring the platform boasts robust mechanical properties that can withstand the demanding conditions under which it will operate. Achieving an optimal balance between the weight of the vehicle and its payload is equally crucial. Moreover, in the event of a component encountering a failure mode, it is imperative that this does not cascade into a failure of the entire aircraft.

Concerning the potential of buckling failure mode, this specific subassembly assumes a pivotal role. Notably, metallic sandwich panels exhibit considerable resilience against the threats of both buckling and bending failure modes, which could otherwise lead to severe consequences for the aircraft. The subassembly possesses the flexibility to be oriented along both the global Z and global Y axis. In scenarios involving axial compression, it is anticipated that these panels might experience buckling. As a result, the ability of the metallic honeycomb sandwich panel to bear buckling loads becomes a critical consideration in the design of aircraft structures.

Every individual component is intricately bonded together using a specialized adhesive with remarkable specific attributes. There exist well-defined standard procedures for aircraft designers to formulate the design of a metallic sandwich panel. Consequently, a thorough consideration of constraints related to distance, material selection, corrosion resistance, and mechanical properties is imperative. In this regard, both the upper and lower sheets, along with the doubler, are crafted from aluminium alloy. Notably, the metallic core consists of an aluminium honeycomb core.

Luqman and Farzana have studied on comparison of buckling strength between metallic and composite panel experimentally [30]. This experiment was conducted with specific parameters, and the subsequent results were analyzed in accordance with these inputs. Upon evaluating the outcomes, it became evident that the aluminium carbon fiber reinforced polymer sandwich panel exhibited superior buckling strength

when compared to the pure aluminium sandwich panel of the same dimensions. Furthermore, a noticeable trend was observed wherein the buckling load carrying capacity of the aluminium carbon fibre reinforced polymer sandwich panel decreased as its thickness was reduced.

Sayyad and Ghugal have shown bending, buckling and free vibration of laminated composite and sandwich beams, but mostly it has been about to civil engineering [31]. They have presented that there are many theories such as equivalent single layer, layerwise/zigzag, finite element method, Carrera unified formulation etc. to design a sandwich panel. Similarly, Abrate and Di Sciuva have presented equivalent single layer theories for composite and sandwich structures including classical, first, second and third order formulations, polynomial and non-polynomial displacement methods [32]. In other study, Brischetto has published exact solutions for multi-layered simply supported sandwich structures mathematically in terms of buckling behaviour [34].

Carlsson and Kardomateas discussed the structural and failure mechanics of sandwich composites, employing modern methods of analysis for sandwich structures [33].

Ingrole, Hao, and Liang presented the design and modelling of auxetic and hybrid honeycomb structures to enhance in-plane properties, specifically compression strength, and potential buckling endurance [35]. They conducted comparisons and obtained results between regular honeycomb, re-entrant auxetic honeycomb, locally reinforced auxetic-strut structure, and a hybrid structure combining regular honeycomb and auxetic-strut structure.

Rose et al. [36] investigated the finite element modelling of various instability responses exhibited by a sandwich panel. They explored different modelling approaches to anticipate both the global and local instability responses of these panels. In this research, the sandwich panel comprises an anisotropic face-sheet and a highly thick core. The outcomes from general analysis solution are compared with the results which are came from various numerical methods [36].

Hua et al. [37] introduced a novel finite element model for determining the global and local buckling behaviours of sandwich beams. The suggested one-dimensional finite

element model was formulated upon a comprehensive kinematics model tailored for investigating both global and local instability modes in a sandwich structure. They observed that the critical load determined through the one-dimensional numerical element model compared to the critical loads derived from the two-dimensional numerical element model, demonstrated satisfactory agreement [37].

D'Ottavio, Polit, Ji, Waas have pointed out benchmark solutions and assessment of variable kinematics models for global and local buckling of sandwich struts that includes solutions of buckling issues for sandwich plates with using analytical models [38].

Shah and Kapania have performed a parametric study by changing honeycomb core angles that are shown to cores as hexagonal and triangular [39]. The study encompassed the examination of failure modes in alternate honeycomb core sandwich panels. Subsequent to the investigation, it was determined that sandwich panels employing triangular cores demonstrated superior torsional and bending stiffness when contrasted with those utilizing hexagonal cores. In this comprehensive study, various failure phenomena such as buckling, intra-cellular buckling, skin wrinkling, shear crimping, or a blend of these factors, which are influenced by cell size, relative density, and core shape, were thoroughly explored under in-plane loading conditions, specifically edgewise compressive loading.

According to Shah and Kapania study, the failure mode distinction becomes evident: shear crimping predominates at lower densities, while panel buckling becomes prominent at higher densities. The mechanical response and failure tendencies of sandwich panels are intricately linked to the mechanical and geometric attributes of the core, as well as the face sheets such as the upper and lower layers. Within this context, crucial failure modes under in-plane loading encompass panel buckling, shear crimping, skin wrinkling, intra-cellular buckling, and local compression [39]. These failure modes exhibit nonlinearity and involve intricate interplays and interactions between the face sheets and core structure. The emergence of these failure modes can be attributed to various factors, including the material composition of components, dimensional parameters of these components, core characteristics such as shape, size, type, density, as well as core thickness, among others.

Petras and Sutcliffe have experimented failure modes loading under three-point bending [40]. Paik has also investigated experimentally and theoretically strength behaviour of aluminium honeycomb sandwich panels under three-point bending, axial compression such as buckling behaviour, and lateral crushing loads as well [41]. Magnucki and Jasion have carried out particular studies about strength and buckling of sandwich beams with corrugated core experimentally, numerically and analytically [42].

Vinson introduced specific models to analyse buckling behavior modes and post-buckling failure modes under in-plane compressive loads. Vinson's objective was to demonstrate the optimal design of composite honeycomb sandwich panels under uniaxial compression [43].

Coburn and Weaver [44] showed the analysis, design, and optimization of variable-stiffness sandwich panels, including buckling considerations. The study conducted by Coburn and Weaver also involved subjecting the panels to in-plane compressive loads and examining phenomena such as panel buckling, shear crimping, skin wrinkling, and more [44].

Jeyakrishnan et al. investigated a study that includes buckling behaviour of hexagonal honeycomb sandwich panel under simply supported boundary conditions and static loading. Their studies contain theoretical, experimental and numerical solutions and each method is compared to others. They have found that the critical buckling load for each method is the same. Three failure modes of buckling have been observed in their study for experimental and numerical analysis [45].

1.3 Scope of the Study

The primary emphasis of this study lies in benchmarking the critical buckling loads of metallic skin-stringer panel structures with varied stringer sections and metallic honeycomb sandwich panel structures that one of both has a doubler. These panels which exhibits the highest resilience against buckling in aerospace applications are widely used on both movable wing and fixed wing platforms. Both metallic panels are designed as per design guides that take into account aerostructure principles such as,

mechanical properties, manufacturability, installation requirements, cost-effectiveness, and maintainability. Computer aided design software is employed to facilitate the design stages.

For the purpose of simulating the mechanical responses of these two differing panel structures subjected to in-plane compression loads, the numerical finite element method such as linear eigenvalue buckling analysis is utilized and critical buckling loads are obtained for each panel. The benchmarking of metallic panels is segmented into three key parts. Initially, the study illustrates the impact of various stringer sections on the critical buckling load, comparing their respective performances. Similarly, a comparison is made between metallic honeycomb sandwich panel with a doubler reinforcement and pure metallic honeycomb core sandwich panel in terms of critical buckling load. The third segment of this benchmarking assesses these two distinct panel concepts for their overall panel buckling performance. It is essential to note that the only loading condition under consideration for this study is compression loading.

1.4 Content of the Study

Chapter 1 is a general introduction and gives an overview that includes the motivation and scope of the study as well as literature review which includes stiffened and sandwich panel structures.

Chapter 2 gives buckling phenomena and buckling theories behind the column, flat thin sheet plates, stiffened panel and sandwich panel structures. Theories in this chapter are mentioned with analytical terms, pictures, graphics, and equations.

Chapter 3 presents fundamental of design of metallic skin-stringer and metallic honeycomb sandwich panels as well as overall metallic panels. This chapter contains aerospace structure design knowledge and experiences. Chapter 3 ends up with design solutions that are created for this study.

Chapter 4 explains linear buckling analysis method and covers the finite element analysis on buckling behaviour of metallic skin-stringer and metallic honeycomb sandwich panel structures. Finite element analysis on buckling behaviour compasses

description of numerical models such as boundary conditions, mesh density and accuracy etc.

Chapter 5 demonstrates the results of linear buckling analysis that is performed in previous chapter. The numerical outputs are analysed and they are compared to each other in terms of critical buckling load capability. Finding of this study are defined in detail.

Chapter 6 encapsulates a summary of the conducted study, the thesis conclusion, and recommendations for future endeavours.



CHAPTER 2

THEORY OF BUCKLING

2.1 Buckling Phenomena and Column Buckling Theory

2.1.1 Meaning of Buckling

Buckling is an instability phenomenon that commonly occurs in compressed slender elements, such as airframe stringers and skin. It can be identified by abrupt displacements in the transverse direction, perpendicular to the direction of compression. This behaviour is critical to understand in the design and analysis of structures to prevent potential failures and ensure the overall structural integrity of the components [46]. Figure 2.1 shows the stages of equilibrium.

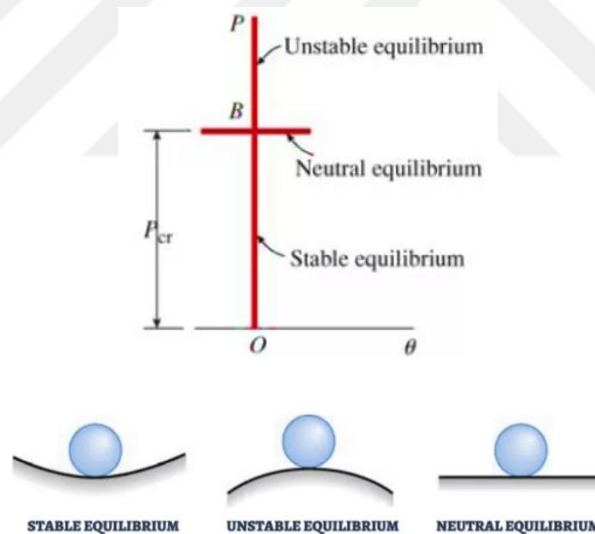


Figure 2.1 Stages of equilibrium [47]

According to the Figure 2.1, if the sphere is situated within a large concave curve, its equilibrium statement is referred as stable equilibrium. That is, when the stable equilibrium is disrupted by a slight displacement, the sphere, following some oscillations, eventually comes back to its initial position. When the sphere is placed on a flat surface, the relevant equilibrium state is considered as neutral. In other words, a

minor displacement does not alter either the forces equilibrium or the potential of the sphere. If the sphere is situated within a large convex curve, its equilibrium statement is referred as unstable equilibrium. In this scenario, even the slightest disturbance can trigger to a total loss of equilibrium, and then resulting in failure. It is noteworthy that in classical buckling theory, the transition from a stable to an unstable equilibrium always goes through a neutral state of equilibrium. This crucial statement establishes the main basis for all mathematical formulations in the linear-elastic stability analysis of plates [48].

Buckling exclusively takes place in structures when they encounter compressive loads [46]. In contrast, tension loads offer the advantage of inducing a structural element to straighten, preventing any occurrence of buckling in the structure. When structures are subjected to applied compression loads, the strength of their members can be compromised due to buckling. Buckling in structures can be triggered either by direct pure compression loads or by shear loads. In both cases, the stability of the structure may be affected, potentially leading to a loss of load-carrying capacity and structural integrity. Shear buckling can be analysed by resolving it into a tension component and a compression component [46]. It is the compression force component that can lead to shear buckling, occurring at a 45-degree angle.

The critical load for buckling depends on numerous factors such as loading characteristics, initial imperfections, boundary conditions, geometry, and material properties. Among these factors, slenderness stands out as one of the most influential [46]. It plays a significant role in determining the failure mechanism by which the structure will ultimately fail. As demonstrated Figure 2.2, the more components have slenderness more, the more they are susceptible to possess buckling phenomena. The graphic in Figure 2.2 has slenderness ration on x axis and y axis of the graphic shows the ratio between the critical buckling strength and yielding strength.

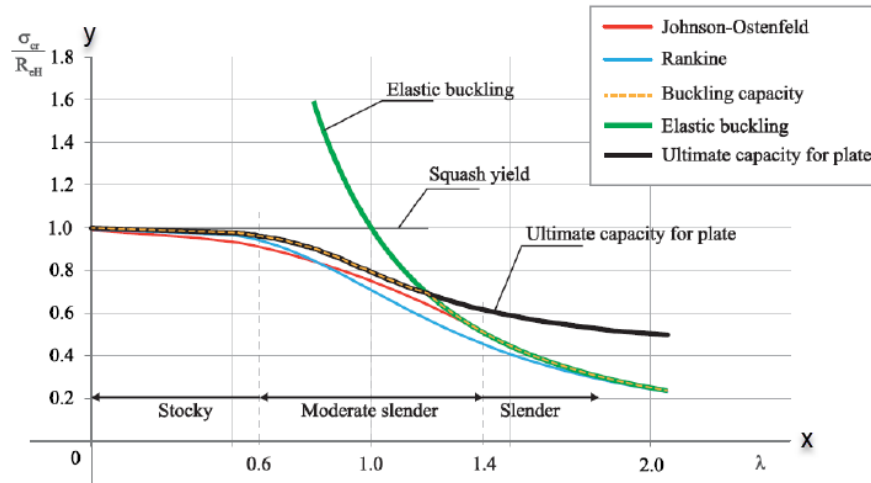


Figure 2.2 Buckling strength vs slenderness curve [49]

The occurrence of buckling is typically classified based on plasticity in three regions that are elastic buckling, elastic-plastic buckling, plastic buckling, as depicted in Figure 2.3.

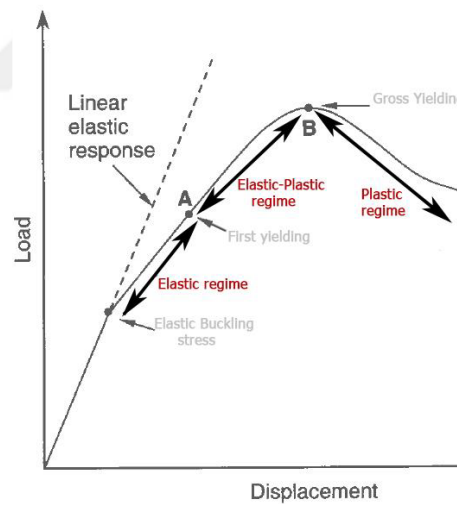


Figure 2.3 Elastic and plastic regions of buckling [50]

Elastic buckling is characterized by the point at which a structure loses its stability, leading to the emergence of significant out-of-plane deflections [46]. The elastic buckling load is the beginning point of this behavior, typically linked to the structure's minimum eigenvalue. Within this range, the stresses applied remain below the yielding

point. Consequently, when the structural element is unloaded, this element returns to its initial shape.

The elastic buckling limit, as illustrated in Figure 2.1, can be categorized as stable, unstable, or neutral based on the load-bearing capacity of the element. Within a stable limit, the structure can support higher loads than the eigenvalue, even in the presence of significant deflection [46]. Conversely, an unstable limit hinders the structure from bearing additional loads, causing its capacity to fall below the eigenvalue with rapid deflection growth. In the case of a neutral limit, the load-carrying capacity remains unchanged.

Elastic-plastic buckling occurs when a specific region in the structural element undergoes yielding, leading to plastic deformation. Subsequently, yielding progresses across the element until gross yielding occurs.

Plastic buckling is defined by a non-linear material behavior and, owing to significant deformations, is also associated with non-linear geometric behavior [46].

2.1.2 Types of Buckling Analysis

2.1.2.1 Linear Buckling Analysis

Linear buckling is an engineering concept using linear theory. The structure can only maintain a certain load level in compression. Linear buckling analysis can be applicable and effective for column and plate structures under certain conditions, especially when dealing with slight manufacturing imperfections or slightly eccentric loading.

The process of linear buckling analysis isolates the buckling modes of a structure based on distinct loading and boundary conditions, assuming that stiffness remains constant throughout the analysis [46]. Additionally, it is assumed that the structure in the pre-buckled configuration is perfectly straight and elastic. Nevertheless, in real-world scenarios, once the initial buckling mode emerges, the stiffness within the buckled region undergoes changes, thereby causing significant shifts in the internal load distribution within the structure. Linear buckling analysis fails to account for this

phenomenon, potentially introducing inaccuracies when calculating additional buckling modes beyond the initiation of buckling, particularly in areas proximate to the buckled region within the examined structure [46]. To ensure reliable outcomes for linear buckling, if required, it becomes essential to incorporate the engineering judgments by a failure assessment [48].

Figure 2.4 depicts linear buckling scenarios, specifically column buckling, aimed at illustrating the underlying physics of buckling. The outcome of this phenomenon is the structure's loss of load-carrying capacity when subjected to compression loading beyond the critical compressive buckling load [46].

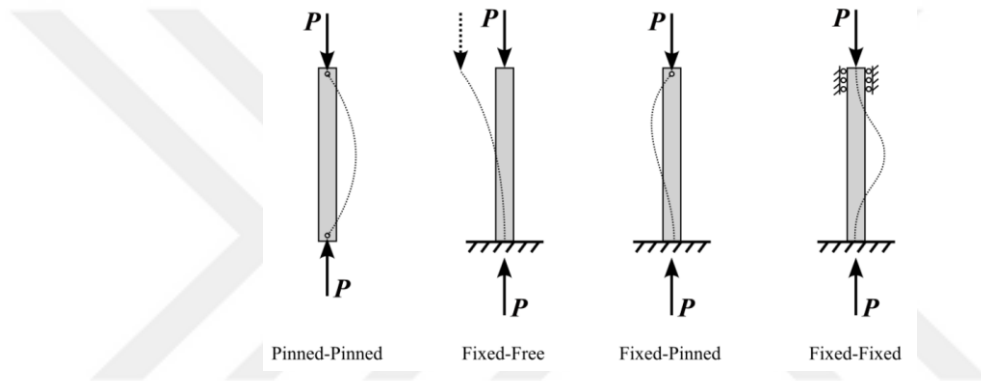


Figure 2.4 Linear buckling cases [51]

In the linear buckling scenario, a critical load corresponds to an equilibrium position, causing the structure to abruptly buckle, as depicted in Figure 2.5. This critical load is commonly referred to as the bifurcation point, that is why linear buckling analysis can be called bifurcation buckling analysis [46]. Bifurcation buckling analysis is focused on determining the load level at which the structural system transitions into an unstable state, while also providing insights into the anticipated shape that the structure will assume following the onset of instability.

Bifurcation buckling analysis does not inherently indicate whether buckling leads to complete structural failure. Certain structures, like flat sheets, retain finite positive stiffness in the post-buckling regime, while others, like thin cylinders, do not exhibit this behavior. When the pre-buckling behavior is linear, or if the impact of pre-buckling nonlinearity is disregarded, the bifurcation buckling analysis results in a

linear eigenvalue problem [52]. In summary, bifurcation buckling analysis calculates critical loads that may not be conservative indicators of complete structural failure. The distinguishing feature of bifurcation buckling analysis is that the direction of the load is perpendicular to the direction of displacement [46].

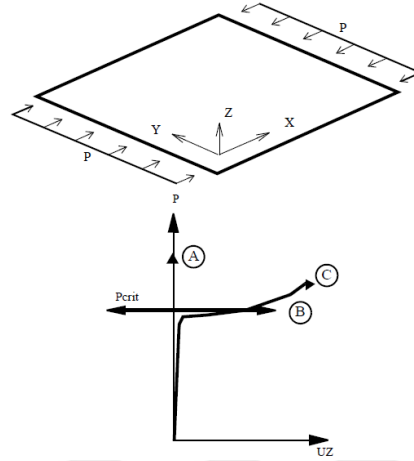


Figure 2.5 Bifurcation points for linear buckling analysis [53]

In Figure 2.5, “*balloon A*” signifies the primary equilibrium path which is determined by linear elastic static analysis, “*balloon B*” indicates the secondary equilibrium path is determined by bifurcation buckling analysis, “*balloon C*” stands the actual load deflection path which includes initial imperfections and geometrical nonlinear effects, and “ P_{crit} ” shows the elastic buckling load. As shown in Figure 2.5, bifurcation point which is placed on the primary equilibrium path in the load-displacement relationship is intersected by a secondary equilibrium path. Equilibrium on the primary path starts unstable at a point, and the buckling behavior beyond the bifurcation point is determined by the secondary path. A linearized analysis identifies the load level for a new deformation pattern that initiates to emerge and it determines the first mode of this pattern [52]. Nevertheless, it does not provide information about the specific shape of the secondary path [52].

Theoretical concept of linear buckling can be described with six steps [46]:

- A compressive force P is applied to a perfectly straight column.
- A lateral force is introduced to create a small lateral deflection.
- When the lateral force is eliminated, and the lateral deflection returns to its original position, the straight form of equilibrium is considered stable.
- As P is gradually increased, a point is reached where the straight form of equilibrium becomes unstable. At this stage, even a small lateral force will cause a deflection that does not disappear when the lateral force is removed. This instability in the equilibrium indicates that the structure is susceptible to buckling under the applied loads.
- The critical load, denoted as P_{crit} , is the axial force necessary to maintain the column in a slightly bent form. When the column is subjected to this critical load, it is considered to be buckled due to the presence of the deformation. This critical point marks the onset of buckling behaviour in the column. Figure 2.6 demonstrates the plate deflection.
- The critical loads for the buckling of plates and shells are developed using the same concept.

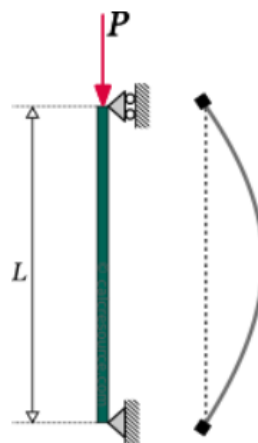


Figure 2.6 Plate deflection [54]

2.1.2.2 Nonlinear Buckling Analysis

Nonlinear buckling analysis considers pre-buckled deformations, geometric nonlinearity, and material nonlinearity effects [46]. This type of analysis is generally more accurate compared to linear analysis. However, it is not automatically more accurate; rather, its advantage lies in its ability to account for additional physical effects and calculate the true physical behaviour of structures.

Nonlinear analysis methods do not yield a specific buckling load value directly [46]. Instead, the onset of buckling needs to be determined from the analysis results. This can be achieved by gradually increasing the applied load on the model until a deviation in the linear proportionality of the load-deformation relationship occurs. Beyond the critical loading point, the structure exhibits non-linear behaviour, and the effects of buckling become evident, as shown Figure 2.7. As the applied load surpasses the critical load, the structure undergoes deformations that deviate from the linear response observed under smaller loads.

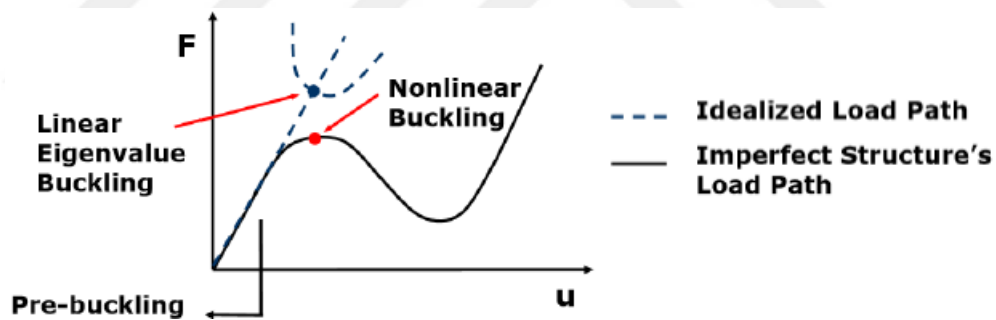


Figure 2.7 Nonlinear buckling behaviour [55]

In nonlinear analysis, the stiffness matrix is updated at each step of the analysis as a result of the deformations occurring under loading increments [46]. This dynamic updating of the stiffness matrix accounts for the changes in the structure's geometry and material behaviour as the applied loads increase. Additionally, it can take into account force following, which means that the loading pattern can change during the analysis based on the deformed configuration. Small deltas in the real structure due to material and/or geometric defects can affect the internal load distribution within the

structure and hence affect the actual achievable critical buckling strength of the real structure [46].

While nonlinear analysis provides a more comprehensive representation of structural behaviour, it still has its own limitations [46]. In many cases, due to the inherent complexity of real structures and the uncertainty associated with material properties, boundary conditions, and imperfections, it is challenging to definitively predict the exact behaviour that will occur between the theoretical model and reality. In addition, this method can be computationally expensive. In addition, calculating the critical loads of structures by using analytical methods for imperfection structures are not currently available in terms of buckling behaviour [52].

2.1.3 General Types of Buckling Failure Mode

Airframe or other structures can experience two primary types of buckling failure phenomena: Global buckling and local buckling. Figure 2.8 provides an illustration that highlights the distinction between these two buckling modes in a compressed beam.

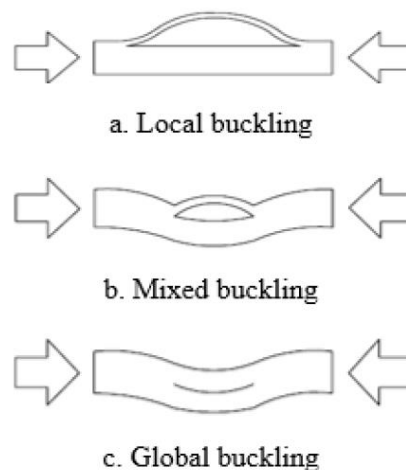


Figure 2.8 General buckling failure modes [56]

2.1.3.1 Local Buckling

Local buckling is commonly associated with detailed components of the structure, such as skin buckling occurring between stringers or elements of stringers and frames.

These localized buckling phenomena can arise due to the interaction of various structural elements, and they play a crucial role in the overall structural integrity and performance of the aircraft [46]. For instance, an example of local skin buckling is shown in Figure 2.9. Local skin buckling takes place when the skin between stringers undergoes elastic buckling, and the stiffeners remain unaffected by the deformation of the skin. On the other hand, when local buckling occurs in a structure, it leads to a change in the stiffness of the affected elements.

Local buckling does not cause distortion in the axis of the entire member, but it jeopardizes the strength of the cross-section due to buckling of an element within the stringer cross-section [46]. Local buckling may or may not result in catastrophic failure, depending on the severity of the deformation and the structural redundancy. Figure 2.10 demonstrates stringer local buckling on a fuselage panel structure.



Figure 2.9 Local skin buckling [57]

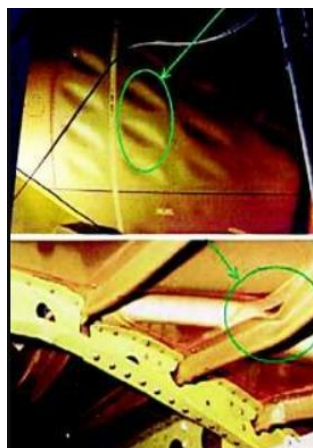


Figure 2.10 Local stringer buckling [58]

2.1.3.2 Global Buckling

Global buckling is the most critical type of buckling as it results in the overall structure's inability to carry any additional load, often leading to catastrophic failure [46]. In many cases, local buckling tends to occur in the structure before global buckling takes place.

Global buckling is characterized by a significant distortion or buckling of the longitudinal axis of the member, which can lead to catastrophic failure of the structure. Euler-column buckling is the term used to describe the global buckling of a skin-stringer section. For instance, an example of global buckling is shown in Figure 2.8.

2.1.4 Buckling and Bending Phenomena

Bending is a phenomenon that occurs in beams or plates when they are subjected to transverse loads or bending moments. This phenomenon allows for deformations as long as they do not exceed the permissible limits for tension or compression stress [46]. Bending is not an instantaneous occurrence; rather, it develops gradually over time. It only leads to failure if the allowable stress values are surpassed.

On the other hand, the buckling phenomenon arises in elements subjected to axial compression forces. Buckling can manifest in various modes, such as flexural and torsional buckling, depending on the type of loads or torque applied and the direction in which the elements buckle. Unlike bending, buckling is an instantaneous instability phenomenon. If buckling occurs globally, it invariably leads to catastrophic failure of the structure.

After understanding the concepts of bending and buckling, it can be concluded that when a body is subjected to forces that attempt to induce curvatures, as seen in beams supporting slabs or other beams and plates, it undergoes bending [46]. On the other hand, buckling occurs when forces are applied to a long and slender body with a narrow cross-section, resulting in longitudinal compression that leads to sudden bending or flexing of the structure. Figure 2.11 shows the difference between buckling and bending load conditions.

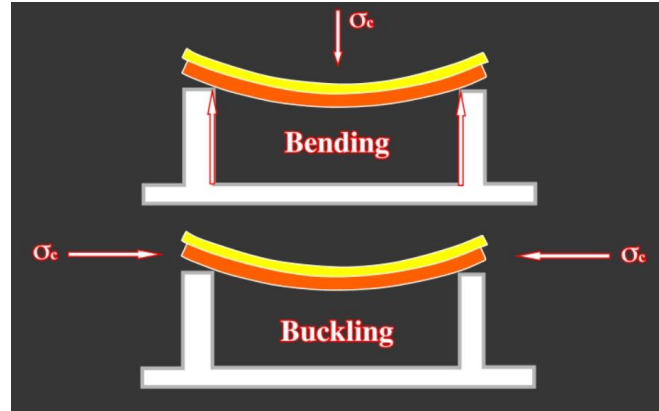


Figure 2.11 Difference between bending and buckling [59]

2.1.5 Column Buckling Theory

Column buckling, also known as the Euler column buckling mode, represents a global failure mode for the panel. The theoretical critical compressive load in Euler column buckling leads to an infinite deviation of the stiffener, giving as the ultimate strength criterion for a column under compression [46]. Euler column buckling theory is given in Eq. 2.1:

$$P_{cr} = \frac{k\pi^2 EI_c}{L_c^2} \quad (2.1)$$

where:

P_{cr} : Critical Euler buckling load.

E : Modulus of elasticity.

I_c : Moment of inertia of the column.

L_c : Column length.

k : Constant that depends on column end conditions.

Prior to reaching critical Euler buckling load, there is often some yielding in the material, resulting in an actual maximum panel load that is less than critical Euler buckling load [46]. In cases where the column is not perfectly straight and regular, the

buckling failure load will be less than the calculated Euler load. Nonetheless, the Euler-buckling load still can be regarded as the maximum axial load that the stiffener can carry. Figure 2.4 illustrates the shape associated with the Euler buckling failure mode.

The distinction between plate buckling, which will be discussed in the next subtitle, and column buckling lies in the boundary conditions. Plate buckling requires three or more supported edges for buckling behaviour, whereas column buckling involves only two supported edges. In addition, the plate keeps to withstand additional loads, post-buckling loads, due to its geometry and boundary conditions in any case. As a result, the critical buckling load for the plates does not signify its ultimate strength, whereas the critical buckling load for the columns shows the exact value [52].

2.2 Fundamental Theory of Flat Thin Sheet Buckling

Plates or sheets are flat, planar, two-dimensional structural elements, with one dimension known as thickness which is significantly smaller than the other dimensions [48]. Geometrically, they can have either straight or curved lines. Similar to beams, plates or sheets do not demonstrate only function as structural components but can also cover entire structures, as seen in examples like aircraft panels. They can possess different boundary conditions such as simply supported, free, and fixed and they contain elastic supports, elastic restraints, and occasionally point supports in certain cases [48].

Flat sheets or plates play a crucial role as essential structural elements and find widespread use within the aerospace industry thanks to their mechanic and economic advantages [48]. This prominence stems from their application in covering key structural components like skin-stringer structures or metallic sandwich panels, which are prevalent in aircraft construction. Slightly curved plates which has been undergone both in-plane compressive forces and lateral loads can display a distinct form of instability known as “*snap-through*” buckling. This behavior is described by reversed deflections arising from the nonlinear correlation between the buckling load and deflections [48]. Given the presence of compression loads acting on these structural

elements, a comprehensive understanding of the stress intensities that may lead to the buckling of thin sheet panels becomes imperative [60].

The aim of thin-plate or sheet theory is to simplify a three-dimensional problem into an approximate two-dimensional representation [52]. The thin-plate theory can be explained based on the following simplifying approximations [48]:

- The material is isotropic, linearly elastic, and homogeneous, adhering to Hooke's law.
- The plate is initially in a flat state.
- The middle surface of the plate is not affected by bending, it maintains unstrained situation.
- The thickness of the plate is assumed too small, such as at least 10 times less than its smallest lateral dimension.
- Transverse deflections are significantly smaller compared to the plate thickness.
- The slopes of the deflected middle surface are considerably smaller than unity.
- Sections that are perpendicular to the middle surface before deformation maintain flat and perpendicular to the deflected middle surface.
- The normal stress in the transverse direction to the plate surface can be disregarded.

When the equilibrium of a thin sheet is disrupted by edge loads that surpass a critical threshold, the plate undergoes buckling. These critical values are determined by assuming a minor initial curvature or a slight transverse load applied to the plate. At the point where the lateral deflection " w " corresponding to these edge load conditions becomes infinite, the critical values are revealed [60].

Another approach to investigating instability involves the buckling of a plate due to a specific external load perturbation, resulting in the calculation of edge loads

corresponding to a buckled configuration (with non-zero deflections). In cases where deflections become extremely large, it was found that this latter solution (known as Euler's solution) approaches the first solution. However, even for relatively small deflections, the calculated edge load closely approaches the critical value determined by Euler.

The second technique is more mathematically advantageous and provides relatively accurate estimates of the compressive strength of the plates. This approach is particularly practical as it focuses on edge loads without considering transverse loads [60].

Figure 2.12 and Figure 2.13 show moment equilibrium for two thin plates from moment equilibrium of thin plates;

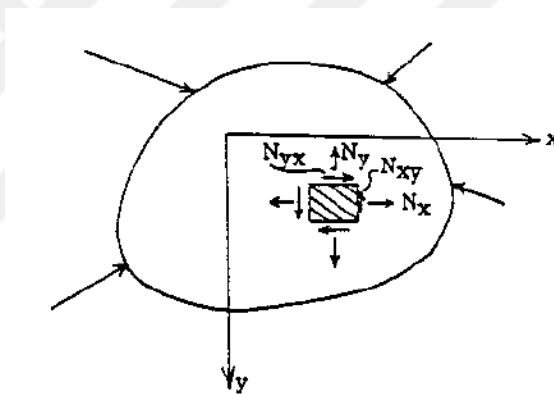


Figure 2.12 Moment equilibrium for thin plates 1 [60]

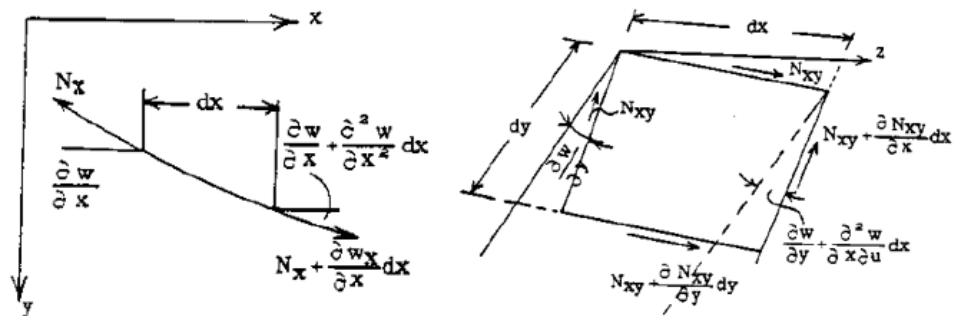


Figure 2.13 Moment equilibrium for thin plates 2 [60]

Eq. 2.2 indicates the governing differential equation of plate buckling and it points out that assumed buckling occurs when “w” is not equal to zero, and this is possible for specific values of the edge loads. In fact, the critical load is determined by the smallest of these edge loads [48].

$$\frac{\partial^4 w}{\partial x^4} + 2 \frac{\partial^4 w}{\partial x^2 \partial y^2} + \frac{\partial^4 w}{\partial y^4} = \frac{1}{D} \left(N_x \frac{\partial^2 w}{\partial x^2} + N_y \frac{\partial^2 w}{\partial y^2} + 2N_{xy} \frac{\partial^2 w}{\partial x \partial y} \right) \quad (2.2)$$

“ N_x ” signifies resultant force in x direction, “ N_y ” points out resultant force in y direction and “ N_{xy} ” and “ N_{yx} ” are resultant forces in xy plane.

The energy method can also be employed to study buckling problems. In this approach, the assumption is made that the plate is initially subjected to plane stress conditions due to edge loads, and the stress distribution is already known. In this scenario, “w” represents a virtual displacement.

When the increment of work is less than the increment of strain energy for every potential mode of buckling, the equilibrium remains stable. Conversely, if the opposite situation arises for a specific configuration of buckling modes, the flat arrangement becomes unstable, and subsequently, the plate will buckle when subjected to any load exceeding the critical load [60].

If the increment of work is equal to the increment of strain energy, the critical load from this equilibrium that is neutral can be found. Eq. 2.3 can be shown below, “w” is a certain assumed deflection that accounts for the boundary conditions, “D” and “ ν ” are flexural rigidity and Poisson’s ratio, respectively.

$$\begin{aligned} & -\frac{1}{2} \iint \left[N_x \left(\frac{\partial w}{\partial x} \right)^2 + N_y \left(\frac{\partial w}{\partial y} \right)^2 + 2N_{xy} \frac{\partial w}{\partial x} \frac{\partial w}{\partial y} \right] dx dy \\ & = \frac{D}{2} \iint \left\{ \left(\frac{\partial^2 w}{\partial x^2} + \frac{\partial^2 w}{\partial y^2} \right)^2 \right. \\ & \quad \left. - 2(1 - \nu) \left[\frac{\partial^2 w}{\partial x^2} \frac{\partial^2 w}{\partial y^2} - \left(\frac{\partial^2 w}{\partial x \partial y} \right)^2 \right] \right\} dx dy \end{aligned} \quad (2.3)$$

A plate of sides a and b as shown Figure 2.14, simply supported under in-plane uniform compression load condition “ N_x ”, “ $x=0$ and $x=a$ ”.

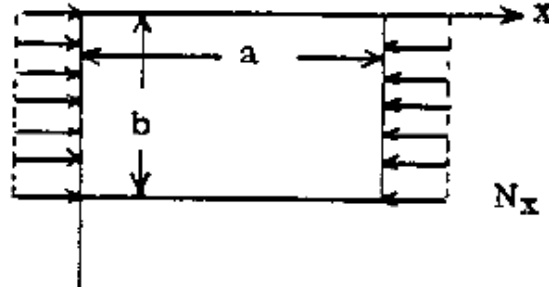


Figure 2.14 A plate under in-plane uniform compression [60]

General expression of the deflection surface of a simply supported plate can be written in Double series form in Eq. 2.4. The wave number for the plate in x direction is demonstrated “ m (buckling half-wave)” and “ n (buckling half-wave)” is used in y direction. When it comes to “ c_{mn} ”, “ a ”, and “ b ”, they are an arbitrary variable, long dimension of panel, and short dimension of panel, respectively.

$$w = \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} c_{mn} \sin \frac{m\pi x}{a} \sin \frac{n\pi y}{b} \quad (2.4)$$

The increment of strain energy found by substituting Eq. 2.4 in the right-hand side of Eq. 2.3, is written in Eq. 2.5:

$$\delta V_1 = \frac{\pi^4 ab}{8} D \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} c_{mn}^2 \left(\frac{m^2}{a^2} + \frac{n^2}{b^2} \right)^2 \quad (2.5)$$

The increment of work done by the external forces is found by substituting Eq. 2.4 in the left-hand side of Eq. 2.3 and “ $N_x = \text{constant}$ ” and “ $N_x = N_{xy} = 0$ ” and Eq. 2.6 is derived.

$$\delta w = \frac{1}{2} N_x \int_0^a \int_0^b \left(\frac{\partial w}{\partial x} \right)^2 dx dy = \frac{\pi^2 b}{8a} N_x \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} m^2 c_{mn}^2 \quad (2.6)$$

When the increment of work is equal to the increment of strain energy, the critical load which is placed in the Eq. 2.6 can be found. Therefore, “ N_x ” is defined in Eq. 2.7:

$$N_x = \frac{\pi^2 a^2 D \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} c_{mn}^2 \left(\frac{m^2}{a^2} + \frac{n^2}{b^2} \right)^2}{\sum_{m=1}^{\infty} \sum_{n=1}^{\infty} m^2 c_{mn}^2} \quad (2.7)$$

Eq. 2.7 has “ c_{mn} ” which is arbitrary. Whereas “ N_x ” is maximum value, “ c_{mn} ” will be minimum value. The minimum expression for “ c_{mn} ” obtained by dropping all terms except “ c_{mn} ” becomes in Eq. 2.8:

$$N_x = \frac{\pi^2 a^2 D}{m^2} \left(\frac{m^2}{a^2} + \frac{n^2}{b^2} \right)^2 \quad (2.8)$$

It is evident that the smallest value of “ N_x ” is achieved when “ n ” is equal to 1. This implies that the plate buckles at this point, as there can be multiple half-waves “ m ” in the direction of compression, but only a single half-wave in the perpendicular direction. When “ n ” is equal to 1, Eq. 2.7 transforms to Eq. 2.9:

$$(N_x)_{cr} = \frac{\pi^2 D}{a^2} \left(m + \frac{1}{m} \frac{a^2}{b^2} \right)^2 \quad (2.9)$$

The value of the number of half-waves “ m ” depends on “ a/b ” and this value has a potential to make this critical “ N_x ” value the smallest possible and “ k ” shown in Eq. 2.10 is a numerical factor depending on “ a/b ”.

$$(N_x)_{cr} = k \frac{\pi^2 D}{b^2} \quad (2.10)$$

From Eq. 2.9 and Eq. 2.10, Eq. 2.11 can be obtained.

$$k = \frac{b^2}{a^2} \left(m + \frac{1}{m} \frac{a^2}{b^2} \right)^2 \text{ or} \quad (2.11)$$

$$k_x = \left[\frac{a}{(mb)} + \frac{(mb)}{a} \right]^2 \quad (2.11)$$

By varying integer values of “ m ”, a graph can be plotted with “ k ” against “ a/b ”, as shown in Figure 2.15 which helps determine the critical load factor “ k ”. The transition from “ m ” to “ $m+1$ ” half-waves take place at the point where the two corresponding lines intersect [60]. Eq. 2.12 characterize the smallest buckling “ m (half-wave)” integer number.

$$\frac{a}{b} = \sqrt{m(m+1)} \quad (2.12)$$

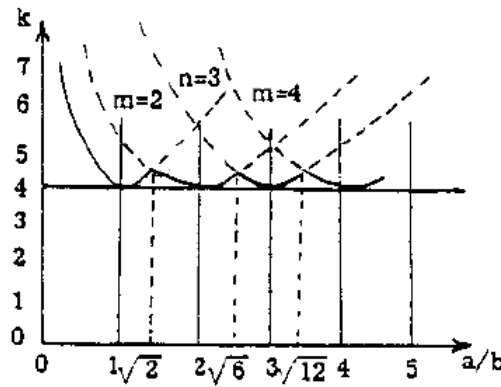


Figure 2.15 Critical load factor for buckling [60]

The number of half-waves increases as the ratio “ a/b ” increases. This indicates that a very elongated plate undergoes buckling with half-waves whose lengths approach the width of the plate.

The buckled plate is subdivided into squares. The critical value of the compression stress can be found with Eq. 2.13:

$$\sigma_{xv} = \frac{(N_x)_{cr}}{h} = \frac{k_c \pi^2 E}{12(1 - \nu_e^2)} \frac{t^2}{b^2} \quad (2.13)$$

where:

k_c : Buckling coefficient which depends on edge boundary conditions and sheet aspect ratio (a/b).

E : Modulus of elasticity.

ν_e : Elastic Poisson's ratio.

b : Short dimension of plate or loaded edge.

t : Sheet thickness.

Buckling situations involve specific boundary conditions as the unloaded edges transition from being free to restrained [60]. In Figure 2.16 top illustration, where the sides are free, the sheet behaves akin to a column. In Figure 2.16 mid illustration, one side is restrained while the other remains free, resembling a flange. Figure 2.16 bottom illustration, both sides are restrained, rendering this restrained element as a plate.

Additionally, the buckling coefficient is given for uniform transverse compressive loading with Eq. 2.14:

$$k_y = \left[1 + \left(\frac{b}{a} \right)^2 \right]^2 \quad (2.14)$$

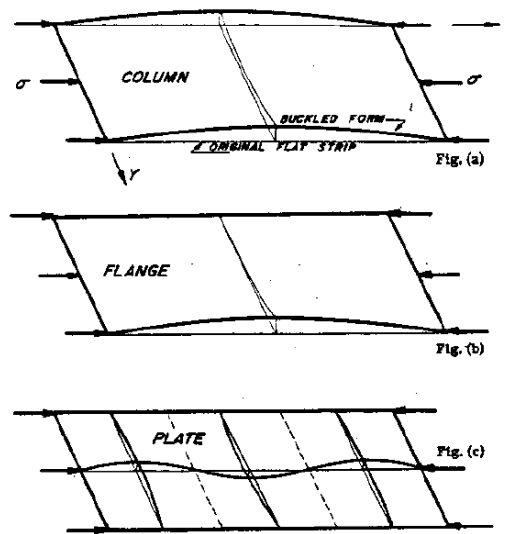


Figure 2.16 General boundary conditions of buckling situation [60]

The Figure 2.17 depicted illustrates curves designed to determine the buckling coefficient across different boundary or edge conditions, in relation to the " a/b " ratio of the thin sheet. The letter "C" at the edge signifies a clamped or fixed condition against rotation, while the letter "F" indicates a free edge. "SS" stands for simply

supported or hinged. According to Figure 2.17, the smallest critical loads frankly can be obtained at the free edges. In this case, the plate behaves like a column and is referred to as a wide column [52].

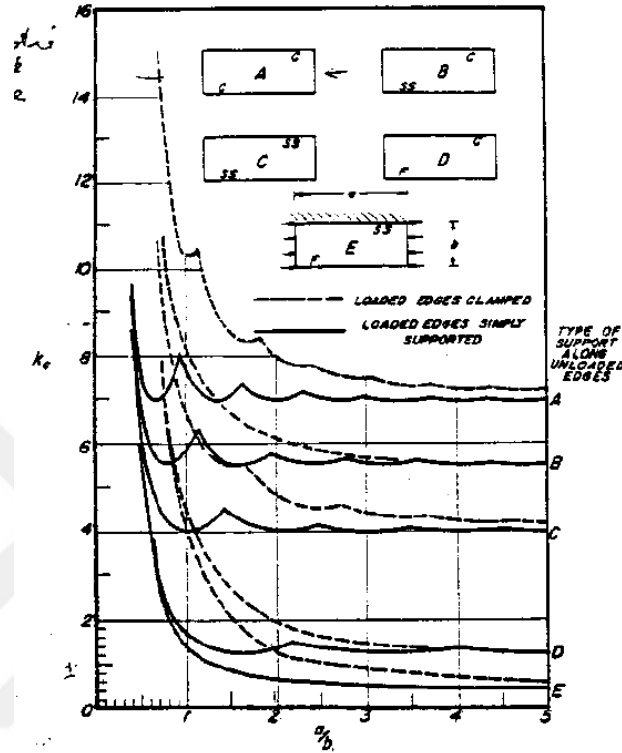


Figure 2.17 Buckling coefficient with aspect ratios and boundary conditions [60]

Eq. 2.15 provides an analytical solution for the elastic buckling of a plate subjected to biaxial loading as shown in Figure 2.18 and having with simply supported boundary conditions. “ D ” in Eq. 2.15 is called plate flexural rigidity and it is expressed in Eq. 2.16. “ σ_x ” and “ σ_y ” indicate stresses in x and y direction, respectively.

$$\frac{m^2}{a^2} \sigma_x + \frac{n^2}{b^2} \sigma_y - \frac{\pi^2 D}{t} \left(\frac{m^2}{a^2} + \frac{n^2}{b^2} \right)^2 \quad (2.15)$$

where:

$$D = \frac{Et^3}{12(1 - \nu^2)} \quad (2.16)$$

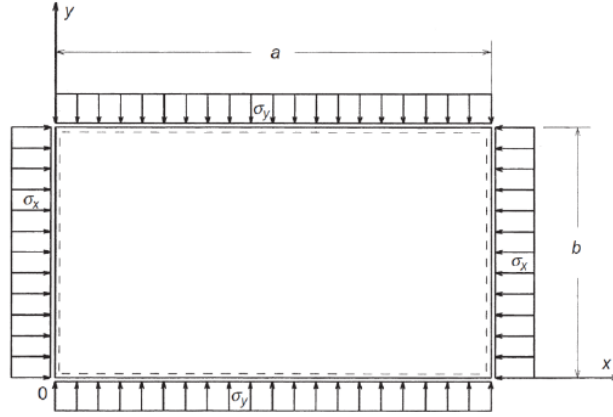


Figure 2.18 A plate under in-plane biaxial compression [50]

Typically, one-half of the wave number is considered either along the transverse edge or in the longitudinal direction where the axial load is overpowering [50]. Generally, “ a/b ” is equal and bigger than 1, “ n ” is equal to 1. Since implemented load ratio “ $c_l = \sigma_y/\sigma_x$ ” is constant, Eq. 2.15 converts to Eq. 2.17 which gives buckling. Therefore, the elastic buckling or longitudinal elastic buckling stress is given in Eq. 2.18.

$$\sigma_x = \left(\frac{m^2}{a^2} + \frac{c_l}{b^2} \right) - \frac{\pi^2 D}{t} \left(\frac{m^2}{a^2} + \frac{1}{b^2} \right)^2 = 0 \quad (2.17)$$

$$\sigma_{xE} = \frac{\pi^2 D \left(\frac{m^2}{a^2} + \frac{1}{b^2} \right)^2}{\left(\frac{m^2}{a^2} + \frac{c_l}{b^2} \right)} \quad (2.18)$$

Eq. 2.19 characterize the smallest buckling “ m (half-wave)” integer number in x direction for biaxial loading condition.

$$\frac{\left(\frac{m^2}{a^2} + \frac{1}{b^2} \right)^2}{\left(\frac{m^2}{a^2} + \frac{c_l}{b^2} \right)} \leq \frac{\left(\frac{(m+1)^2}{a^2} + \frac{1}{b^2} \right)^2}{\left(\frac{(m+1)^2}{a^2} + \frac{c_l}{b^2} \right)} \quad (2.19)$$

According to the Eq. 2.19, The relationship between the half-wave number and the applied loading ratio “ c_l ” and plate aspect ratio is noticeable.

With the provided definition of " c_l ," the elastic transverse buckling stress is expressed in Eq. 2.20:

$$\sigma_{yE} = c\sigma_{xE} \quad (2.20)$$

2.3 Fundamental Theory of Stiffened Panel Buckling

Adding stiffeners on a panel or plate can significantly improve the in-plane load capacity of a plate, and it reduces the risk of buckling [48]. Nonetheless, enhancing the critical load for the buckling phenomena also escalates the challenges faced during analysis. Specifically, the complexity of the buckling problem rises as the number of necessary variables expands, and so the buckling modes tend to become sophisticated more. As a result, in engineering practice, simplified formulas, like Eq. 2.25, or charts are predominantly employed for designing plates [48].

The primary load-carrying elements in a stiffened panel are longitudinal and transverse stiffeners, which serve as supporting members. Longitudinal stiffeners which are parallel to the in-plane load, resist a share of the applied force, whereas transverse stiffeners are primarily utilized to separate the plate into smaller zones, as these types of stiffeners carry a relatively minor portion of the load [48]. The panel generally has a rectangular or slightly curved shape. When the panel undergoes buckling and lateral deflection occurs slowly, the stiffeners which are placed on the panel counteract the rotation of the panels. In other words, the panel is supported elastically along all four of its sides while influence of the stiffeners is generally regarded as weak constraints.

In contrast to column buckling, this stiffened panel structure has the ability to withstand additional loads even after buckling takes place, either in the elastic or inelastic range, until it reaches its ultimate capacity [61].

Special stiffened panel structures for aerospace design will be discussed in Chapter 3. However, general overview of the stiffened panel can be seen in Figure 2.19.

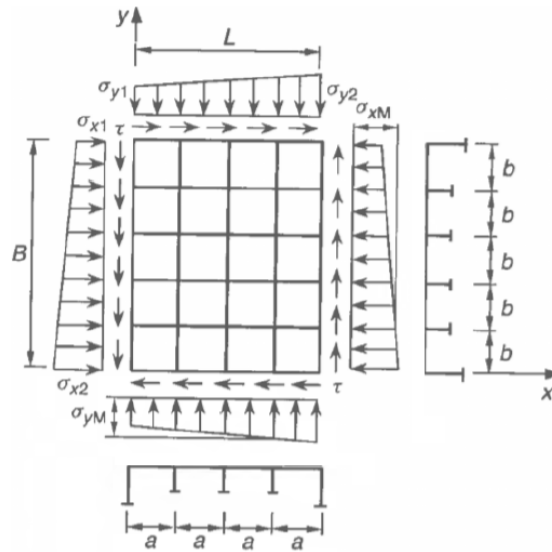


Figure 2.19 Stiffened panel under combined in-plane loads and lateral pressure [61]

In the buckling analysis of stiffened plates, there are typically two forms of buckling that are taken into account [48]. One likely mode is the local buckling of the plate or panels between the stiffeners. In this case, it is assumed that the panel is reinforced with robust stiffeners as much as possible. When it comes to the second scenario, an overall or global buckling can occur on the stiffened plate assembly and this phenomenon is referred to as primary buckling [48].

For certain structures, it may be needed that the local buckling of the plate firstly happens without any causing significant distortions on the stiffeners [48]. In such instances, since there is a structural interaction between the panel and stiffeners, the stiffeners simply push the nodal lines of the buckling modes of the panel. As a result, the problem is simplified to calculating the critical loads of the unstiffened panels that have curves or are flat and identifying the minimum flexural rigidity of the stiffeners necessary to ensure early panel buckling [48].

Achieving a significantly more cost-effective design can be feasible by allowing synchronous local and primary buckling at approximately the same stress level [48]. In order to do this, it is essential to consider the structural interaction between the panel and stiffeners for the linear buckling stability analysis of stiffened panels. When the panel is fortified with many parallel uniform stiffeners and they provide equally

spaced, this configuration can be potentially converged to orthotropic plate theory [48]. Apart from this general condition, the stiffened panels that have three or fewer stiffeners should be taken into account differently. At that point, Rayleigh-Ritz method can be used for the interaction between stiffeners and panels or plates. In other words, there is no significant distinction in the critical load whether the number of stiffeners is infinite or exceeds this limit [48].

2.3.1 Buckling Behaviour of Stiffened Panel

The distance between stiffeners on a stiffened plate is considered equal and compressive load is applied in longitudinal direction or stiffener direction. If the stiffeners possess sufficient stiffness and exhibit minimal deflection when subjected to compression loads, the plate experiences localized buckling, while the stiffeners remain unaffected by buckling [50]. This situation can be seen in Figure 2.20-A and the local buckling mode of the plate is defined in Eq. 2.21:

$$w_{long} = c_{mn} \sin \frac{m\pi x}{a} \sin \frac{n\pi y}{b} \quad (2.21)$$

Secondly, when a compression load is applied to the stiffened panel in a transverse direction, and the stiffeners exhibit sufficient stiffness, a distinct mode of local buckling takes place [50]. This situation can be seen in Figure 2.20-B. The local buckling mode corresponds to a one-half wave mode between stiffeners and is described in Eq. 2.22:

$$w_{transv} = c_{1n} \sin \frac{\pi x}{a} \sin \frac{n\pi y}{b} \quad (2.22)$$

Thirdly, if the stiffeners lack sufficient stiffness, overall or global buckling occurs, as illustrated in Figure 2.20-C and the buckling mode points out a one-half wave mode, irrespective of the direction of compression [50]. Global buckling frequently occurs as a secondary buckling phenomenon following the primary occurrence of local buckling and it is described in Eq. 2.23 and it can be seen in Figure 2.20-D.

$$w_{overall} = c_{11} \sin \frac{\pi x}{a} \sin \frac{\pi y}{b} \quad (2.23)$$

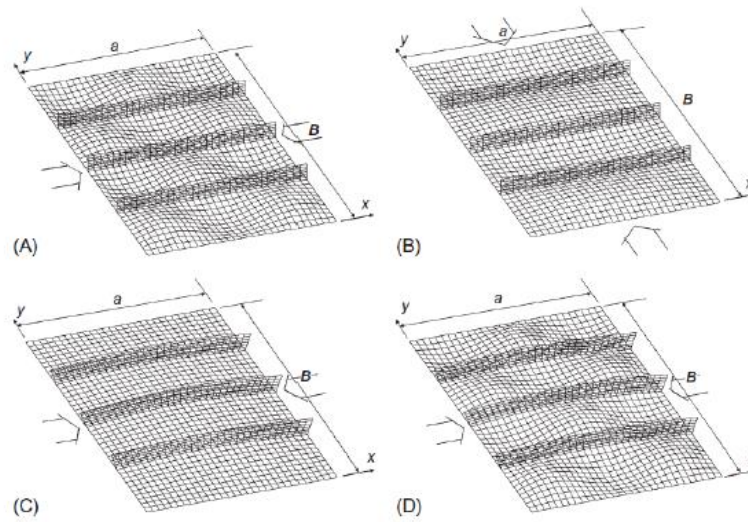


Figure 2.20 Buckling modes of stiffened plate under compression:(A) local buckling, (B) local buckling, (C) overall buckling, (D) overall buckling after local buckling [49]

In addition, overall or global buckling mode behaves like a single plate buckling mode. The presence of global buckling is regarded as a notable structural instability and that's why the panel is designed in order to prevent this failure mode.

In the case of longitudinal loading, the panel's global buckling can be estimated through the column buckling of a plate and a stiffener combination under the simply supported boundaries [61]. The elastic global buckling stress can be described in Eq. 2.24 which is line with Euler buckling theory.

$$\sigma_{xEO} = \frac{P_E}{A} \quad (2.24)$$

where:

P_e : Euler buckling load in Eq. 2.25, the same in Eq. 2.1.

A : Cross section area of the panel (including stiffener combination).

$$P_E = \frac{\pi^2 EI_e}{L^2} \quad (2.25)$$

I_e : Moment of inertia of the panel (including stiffener combination).

L : Length of the plate (including stiffener combination).

When it comes to transverse loading condition, the global buckling strength of the panel can be anticipated using the Eq. 2.13 by ignoring the stiffeners.

As for a combined biaxial loading, the equations for thin plates or sheets can be extended to encompass orthotropic constructions as well [52]. The global buckling strength of the panel can be calculated analytically by using non-linear differential equations which calls large-deflection orthotropic plate theory that means the lateral deflection of the panel is equal to zero before buckling starts and this situation can be described in Eq. 2.26 and baseline demonstration is Figure 2.19 [61] .

$$\frac{m^2 B}{L} \sigma_x + \frac{m^2 L}{B} \sigma_y + \frac{\pi^2}{t} \left(D_x \frac{m^4 B}{L^3} + 2H \frac{m^2 n^2}{LB} + D_y \frac{n^4 L}{B^3} \right) = 0 \quad (2.26)$$

In Eq. 2.26, “ B ” signifies width of the panel, “ L ” indicates length of the panel, and “ H ” implies effective torsional rigidity of the panel under combined biaxial loading.

The elastic global buckling stress under combined biaxial loading can be provided in Eq. 2.27:

$$\sigma_{xEO} = - \frac{\pi^2}{t \left(m^2 \frac{B}{L} + c_l n^2 \frac{L}{B} \right)} \left(D_x \frac{m^4 B^2}{n^2 L^4} + 2H \frac{m^2}{L^2} + D_y \frac{n^2}{B^2} \right) \quad (2.27)$$

where:

D_x and D_y are flexural rigidities of the plate in x and y direction. If feasible, it is advisable to find the flexural and torsional rigidities via test methods [48].

$c_l = \sigma_y / \sigma_x$: Implemented load ratio.

H : Effective torsional rigidity.

2.3.2 Buckling Behaviour of Stiffeners

Various buckling modes of stiffeners, primarily under compression, are depicted in Figure 2.21. Lateral torsional buckling involves the rotation of stiffeners around a panel-stiffener assembly projection and it can be demonstrated in Figure 2.21-A. In the case of more slim stiffeners like a flat-bar stiffener, there is a tendency to buckle locally more, known as tripping which is shown in Figure 2.21-B. Tripping is prone to happening and frequently interacts with local panel buckling. When a slender web with a comparatively rigid flange is under compression loading, there is a possibility of local web buckling occurring and this frequently affects the plate due to the interaction between components [50]. This failure can be seen in Figure 2.21-C and in Figure 2.22 individually. Typically, local panel buckling occurs before these kinds of stiffener buckling failures which can lead to global buckling.

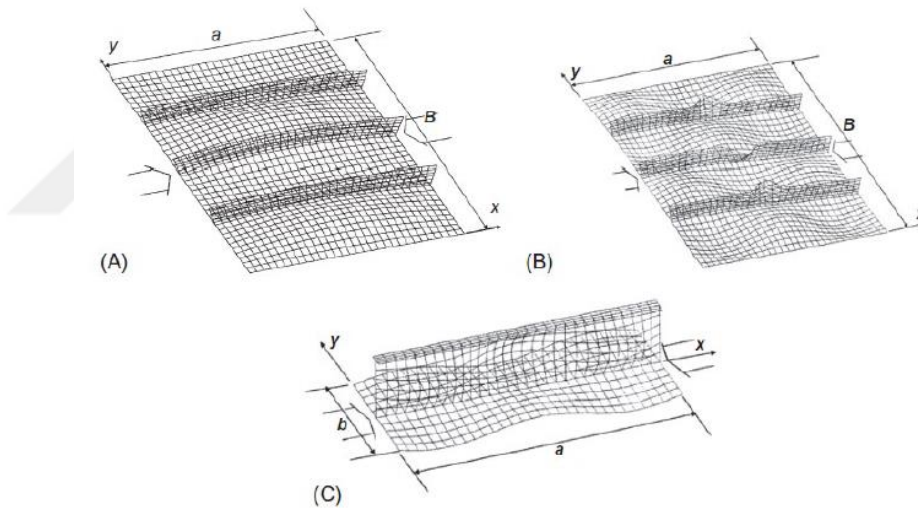


Figure 2.21 Buckling modes of stiffener in stiffened plate. (A) lateral torsional buckling. (B) tripping. (C) web buckling [50]

In the occurrence of overall buckling, as depicted in Figure 2.20-C, the deflection of a stiffener placed at “ $y = y_i$ ” in transverse direction is expressed in Eq. 2.28 [50]:

$$w_{stiff} = c_{11} \sin \frac{\pi x}{a} \sin \frac{\pi y_i}{b} \quad (2.28)$$

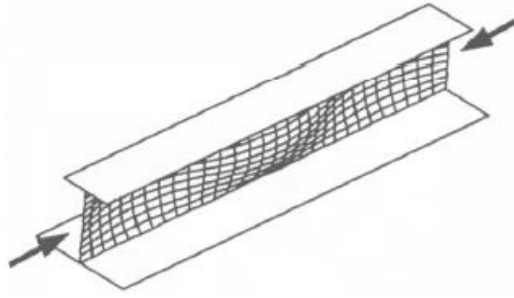


Figure 2.22 Stiffener web local buckling [61]

The local buckling of the stiffener web is primarily affected by the torsional rigidities of mating components. This buckling mode can manifest suddenly, leading to the strength loss of the stiffened panel.

The stiffener web buckling strength can be explained with differential equations which calls out of plane deflection. According to these equations, there is a zero deflection at the beginning and boundary conditions are simply supported [61]. This situation can be shown in Eq. 2.29:

$$D_w = \left(\frac{\partial^4 y}{\partial x^4} + 2 \frac{\partial^4 y}{\partial^2 x \partial^2 z} + \frac{\partial^4 y}{\partial z^4} \right) + t_w \sigma_x \frac{\partial^4 y}{\partial z^4} = 0 \quad (2.29)$$

where:

y : Lateral deflection of stiffener web.

D_w : Bending rigidity of stiffener web.

t_w : Thickness of stiffener web.

Eq. 2.30 is derived from Eq. 2.29.

$$\sigma_E^W = -k_w \frac{\pi^2 E}{12(1 - \nu^2)} \left(\frac{t_w}{h_w} \right)^2 \quad (2.30)$$

where:

σ_E^W : Elastic buckling strength of stiffener web.

k_w : Buckling strength coefficient of stiffener web.

h_w : Short dimension of stiffener web.

Values of buckling strength coefficient of stiffener web for flat bar stiffeners can be seen in Eq. 2.31 [61]:

$$\begin{aligned}
 k_w &= 0.303\xi_p + 0.427 \quad \text{for } 0 \leq \xi_p \leq 1 \\
 k_w &= 1.227 - \frac{1}{(1.40\xi_p + 0.428)} \quad \text{for } 1 < \xi_p \leq 6 \\
 k_w &= 1.2652 \quad \text{for } 60 < \xi_p
 \end{aligned} \tag{2.31}$$

ξ_p : Torsional rigidity of the plating.

The buckling coefficient for a stiffened panel is generally considered as 4. Nevertheless, the local buckling strength of a stiffened panel is influenced by the correlation between the panel and the stiffener web so increasing of this local buckling strength is inevitable for a stiffened panel [50]. Stiffener web criteria should be thought as a plate in order to handle the correlation between the panel and stiffener web properly. This can be expressed in Eq. 2.32 and Eq. 2.33, respectively for panel and stiffened panel as well as “ d_{web} ”, “ w_{web} ”, and “ h_{web} ” signifies local buckling mod of a stiffened panel in terms of stiffener web, deflection of a stiffened web, and height of a stiffened web, respectively. “ V_i ” also implies deflection components of a stiffened web.

$$w_{web} = \sin \frac{m\pi x}{a} \left\{ c_{mn} \sin \frac{\pi y}{b} + \frac{1}{2} c_{mn} (1 - \cos \frac{2\pi y}{b}) \right\} \tag{2.32}$$

In Eq. 2.32, the first term in the brackets specifies the buckling mode for a simply supported plate, while the second term represents for a clamped plate. Second term generates a bending moment that flows to stiffener web.

$$d_{web} = \sin \frac{m\pi x}{a} \left\{ V_1 \frac{z}{h_{web}} + V_2 \left(1 - \cos \frac{\pi z}{2h_{web}} \right) + V_3 \sin \frac{\pi z}{2h_{web}} \right\} \quad (2.33)$$

In Eq. 2.33, the first term in the brackets specifies a rigid body rotation which is related to intersection of stiffener web and the panel. Second term generates a bending moment that flows to the panel while third term shows another bending moment that is transferred from stiffener web to stiffener flange.

2.4 Fundamental Theory of Sandwich Panel Buckling

Sandwich panels are formed by constituting very thin three or more materials which are bonded together by using powerful adhesives. Therefore, they have anisotropy behaviour in their nature. In contrast to the isotropic materials, anisotropic materials demonstrate complex mechanical behaviours that are not readily compliant with exact solutions [48]. These bonded material layers can have isotropy or anisotropy behaviours. When formulating the governing differential equation, the assumption is made that each independent layer is isotropic, and any likelihood of sliding between them is prohibited and this refers to classical laminated plate theory that represents a special case of the Kirchhoff's theory for thin plates where the influence of transverse shear and strain components in the plate is not taken into account. [48].

Buckling may not harm a sandwich panel structure but this occurrence should be blocked since the buckling can trigger negative load carrying capacity and loss of stability. When thinking on the buckling phenomena of the sandwich panel structures, global or overall buckling and various kinds of local buckling modes are taken into account. The buckling strength of a sandwich panel is based on elastic beam column and plate theory.

The estimation of the global buckling strength of a sandwich panel can be approached either by simulating it as a beam column which has its own length and cross-sectional characteristics as variables, or by solving it as an orthotropic plate with equivalent shear stiffnesses and bending. In sandwich panels, there is a widespread supposition that the bending stiffness of the core and of the individual faces, with respect to their own centres, can be ignored [52].

In order to estimate the local buckling strength of a sandwich panel, each component that defines faceplates and core is analysed separately with considering local boundary conditions and interactions between the core and faceplates [62].

Figure 2.23 shows a typical uniaxial loading on a sandwich panel structure, while Figure 2.24 illustrates the most frequently occurring buckling failure modes for a sandwich panel structure

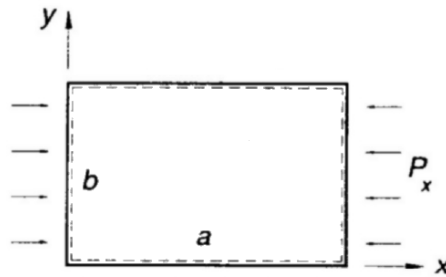


Figure 2.23 Sandwich plate subjected to uniaxial compression [62]

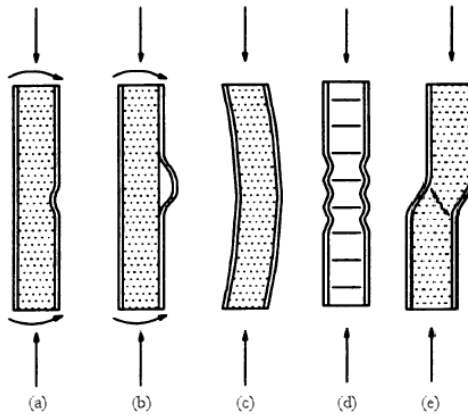


Figure 2.24 Buckling failure modes for a sandwich panel: (a) and (b) face wrinkling or local buckling, (c) global buckling, (d) intracell buckling or dimpling, (e) shear crimping [63]

2.4.1 Global or Overall Buckling for Sandwich Panels

Global buckling and shear crimping are the global buckling modes for a sandwich panel structure. Inadequate core shear rigidity or panel thickness can lead to global

buckling. Shear crimping can appear after global buckling. This is typically linked to a low core shear modulus or inadequate adhesive shear strength.

A sandwich panel that has an aspect ratio “ $a/b > 4$ ” (a represents the panel length and b denotes the panel width as mentioned in Chapter 2.3), and that has free edges without any load can be considered as a beam column. The critical global elastic buckling load can be defined in Eq. 2.34 and the axis system is identical as mentioned in Chapter 2.3 [64].

$$N_x = \frac{\frac{m^2 \pi^2 D_{11}}{k^2 a^2}}{\frac{n^2 \pi^2 D_{11}}{D_{12} k^2 a^2}} \quad (2.34)$$

where:

N_x : Critical buckling load in x direction as described in Chapter 2.3.

k : Buckling coefficient according to boundary and load conditions.

a : Length of the panel.

m : Number of half wave length.

D_{ij} : Elements of flexural stiffness matrix.

In the case of a sandwich panel with a small aspect ratio and all edges being restrained or loaded, the plate acts as an orthotropic plate. The critical global elastic buckling load of this type of panels can be determined in Eq. 2.35 and the axis system is identical as in Chapter 2.3 [64].

$$N_x = K_0 \frac{\sqrt{D_{11} D_{22}}}{b^2} + \frac{C \pi^2 D_0}{b^2} \quad (2.35)$$

where:

N_x : Critical buckling load in x direction, as described in Chapter 2.3.

K_0 : Buckling coefficient according to boundary and load conditions.

b : Width of the panel.

C : Panel Coefficient according to type of support.

D_{ij} : Elements of flexural stiffness matrix and $D_0 = D_{12} + 2D_{33}$.

The flexural stiffness matrix for orthotropic entire metallic sandwich panels can be given in Eq. 2.36 and Figure 2.25 shows buckling coefficients for an orthotropic plate [64].

$$D_{ij} = \begin{bmatrix} \frac{EI_1 h^3}{12a(1-v^2)} & \frac{vEI_1 h^3}{12a(1-v^2)} & 0 \\ \frac{vEI_1 h^3}{12a(1-v^2)} & \frac{EI_2 h^3}{12b(1-v^2)} & 0 \\ 0 & 0 & \frac{vEt_c h^2}{2(1+v)} \end{bmatrix} \quad (2.36)$$

where:

E : Young modulus of face plates and core.

h : Core height.

t_c : Equivalent thickness.

t_b : Thickness of bottom face plate.

t_t : Thickness of top face plate.

I_1 : Moment of inertia along longitudinal axis.

I_2 : Moment of inertia along transverse axis.

Equivalent thickness can be described in Eq. 2.37:

$$t = \frac{2t_t t_b}{t_t + t_b} \quad (2.37)$$

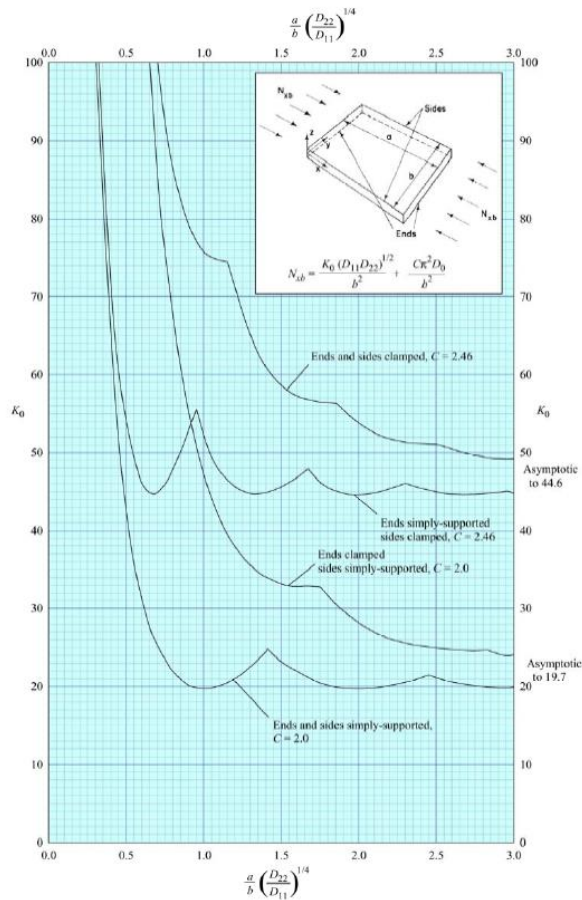


Figure 2.25 Buckling coefficient for an orthotropic plate [64]

2.4.2 Local Buckling for Sandwich Panels

Local buckling can happen in the top face plate, bottom face plate, or core of metal sandwich panels. Face or wrinkling and intracell buckling are the common failure modes of a local buckling. The face plates or core of the sandwich buckle locally and this situation can be known as an "elastic plate buckling". Its buckling direction, whether inside or outside, is related to the relative strengths of the core in compression and the adhesive tension in planarity. When it comes to intracell buckling, it can happen in cellular honeycomb core sandwich panels. This occurrence is limited to configurations with extremely thin facings and large core cells in sandwich structures. Therefore, this failure mode can trigger to face wrinkling with spreading along the neighbour cells [64].

Eq. 2.38 is the most frequently employed formula for predicting the critical local buckling stress for sandwich panels. It relies on the concept of the "equivalent plate segment" and it is situated between the stiffening members and the members themselves [64].

Loads parallel or perpendicular to the core direction can trigger buckling in the top face plate, bottom face plate, and core plate. Additionally, local buckling can be induced by a shear load in the plane of the top or bottom faceplate, as well as a combination of the load cases [64].

When the load is applied parallel to the core direction, the critical buckling stress of the top face plate can be determined using the Eq. 2.38:

$$\begin{aligned}\sigma_{crit} &= \sigma_e = 0.9kE \left(\frac{t}{b_c}\right)^2 \quad \text{for } \frac{\sigma_e}{\sigma_0} \leq 0.5 \\ \sigma_{crit} &= \sigma_0 - \frac{\sigma_0^2}{4\sigma_e} \quad \text{for } \frac{\sigma_e}{\sigma_0} > 0.5\end{aligned} \tag{2.38}$$

where:

σ_{crit} : Critical buckling stress.

σ_e : Elastic buckling stress.

σ_0 : Plate material yield stress.

k : Buckling coefficient, $k=4$ conservative approach.

E : Young's modulus of plate material.

t : Plate thickness.

b_c : Width of plates between two core elements.

When the in-plane load is vertical to the core direction, the impact of this loading condition can be determined using the Eq. 2.39 [64]:

$$\sigma_e = 0.9kE \left(\frac{t}{b}\right)^2 + 0.82E \sqrt{\frac{E_c t}{E h}} \quad (2.39)$$

where:

E_c : Young's modulus of homogenised core.

h : Core height.



CHAPTER 3

METALLIC PANEL DESIGN

3.1 Metallic Stiffened Panel Design

A skin contains multiple fuselage panels, whereas a shell or a stiffened panel is formed by the assembly of longitudinal stringers and panels. Main functions of the skin are [65]:

- Bearing and transferring the load induced by aerodynamic forces, particularly transverse loads.
- Collaborating with the stringers to support bending forces, encompassing both compression and tension.
- Resisting torsional forces against shear stress.
- Safely containing and maintaining fuel pressure within the aircraft.

Skin-stringer assembly is essentially known as cover of an aircraft and plays a crucial role for structural integrity. Stringers that lie in longitudinal direction can be riveted or bonded to the skin. Skin-string assembly commonly is used in numerous sections of an aircraft. For example, while the upper covers are designed to bear buckling that causes compression instability, the lower covers are tailored to withstand ultimate tensile and fatigue loads. A general layout of a metallic panel assembly can be seen in Figure 3.1.

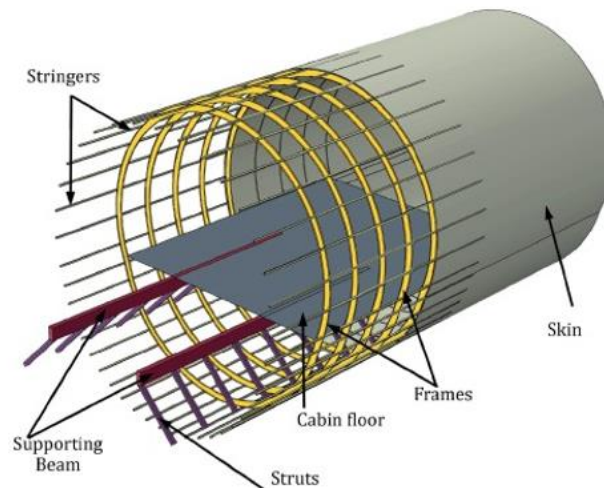


Figure 3.1 Example of a fuselage metallic panel [66]

The fuselage, except for helicopter fuselages, experiences loading due to pressurization which affects some sections and mechanical forces like buckling, bending and, shear stress. In order to support these loads, the fuselage panels often need to be strengthened using frames and stringers. Stringers are responsible for picking up longitudinal loads, frames carry transverse loads and distribute concentrated loads into fuselage shell, and the skin transfers both loads to whole structure through fasteners, adhesives etc [65]. Figure 3.2 shows a stiffened fuselage panel with frames that are placed in transverse direction.

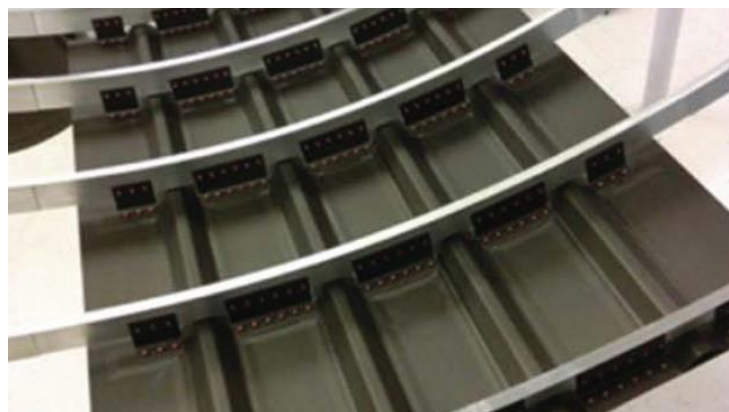


Figure 3.2 A stiffened panel [67]

A panel can be a flat plate, simple curved, or double curved metal formed-sheet, it requires a decision of the appropriate stringer or frame types before designing the panels. This decision is based on considering the specific loads that the panel will be subjected to. Different panels variations can be seen in Figure 3.3.



Figure 3.3 Airbus Beluga single and double curve panels [68]

Three distinct sorts of stiffening options can be appropriate for designing a panel [65].

- Stiffening with the panel solely reinforced by one-way frames.
- Stiffening with two-way discontinuous stringers.
- Stiffening with two-way continuous stringers.

One-way frame stiffening involves using a minimal number of parts, making it a lightweight and cost-effective option. However, it is most suitable when the load is primarily internal pressurization. In such instances, the skin thickness should be adequate to handle the longitudinal loads. Figure 3.4 shows a one-way stiffened panel example.

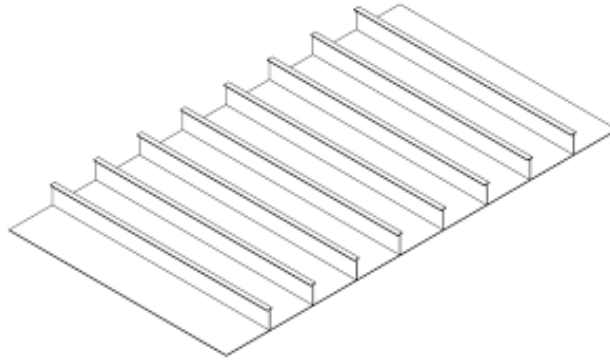


Figure 3.4 One-way stiffened panel [69]

Two-way stiffening with discontinuous stringers is used with continuous frames and simple parts. The use of stringers in this method makes maintainability straightforward. However, this type of stiffening gives a lower longitudinal load carrying capacity compared to others, and it involves components much. Figure 3.5 demonstrates a two-way stiffening with discontinuous stringers panel instance.

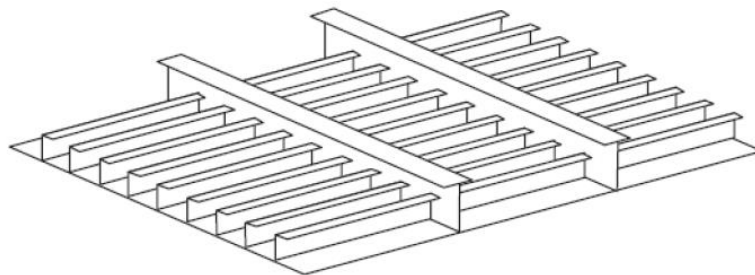


Figure 3.5 Two-way stiffened discontinues stringers panel [29]

Regarding bi-directional stiffening with continuous stringers, the load-carrying capacity is notably strong, even accommodating complex loading conditions. This type is commonly used for panel designs because of its advantageous. Figure 3.6 shows a two-way stiffening with continuous stringers panel example.

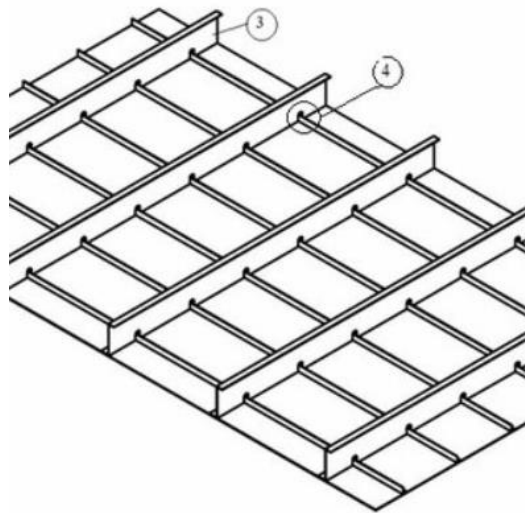


Figure 3.6 Two-way stiffened continuous stringers panel [70]

Some design steps for an airframe panel can be summarized below [65]:

- After selecting the type of stiffening panel, the fundamental geometry of the panel is established, encompassing design concerns such as material selection, initial thickness, frame and stringer spacing determination, and section of frame and stringer, position of structural parts and, assembly concern.
- Weight reduction concern such as pockets can be estimated for optimizing panel weight. The minimum panel thickness is governed by material attributes and manufacturing requirements. As for locations of the pockets, whether on the interior or exterior, it is linked with their usage purpose within aerodynamic or non-aerodynamic zones.
- In regions where fasteners are employed, the thickness aligns with fastener dimensions and fastener installation criteria.

3.1.1 General Overview of Loads on Metallic Panels

The primary load carrying component of the aircraft structure is the skin, which is further reinforced with stringers and frames that are located at regular intervals. Frames give support points for the stringers. The overall stiffness of the panel is influenced by the spacing between these frame support points. During static pre-sizing

of a panel, the design typically focuses on the skin and stringers, while the contribution of frames is not fully considered. Stiffened panels can withstand various types of loads, including tensile, compressive, and shearing forces.

Tension loads are distributed between the skin and the stringers, both contributing to carrying these loads. The static limit of the panel under ultimate load is determined by the material properties and the cross-sectional characteristics of the structure [65]. At that point, the frames serving as fail-safe components to prevent fatigue crack propagation in the skin. Since the frames are redundancy components, the distance between two successive frames becomes significant.

Shear loads in the aircraft skin are influenced by the material properties and can lead to skin wrinkling as shown in Figure 3.7 at a critical load. However, the critical load for skin wrinkling is generally higher than the critical shear loads of the panel sections [65]. When skin wrinkle occurs, they are typically at about 45 degrees in the specific direction, and this introduces new tension loads known as diagonal tension. After skin wrinkle occurs, the skin redistributes the extra loads that are composed of partly shear loads and partly diagonal tension. The diagonal tension is then transferred to the stringers, leading to a compression load in those regions and this load scenario is compensated by the stringers. At parallel, the frames contribute to stabilizing the fuselage shell.



Figure 3.7 Skin wrinkling [71]

In compression, the aircraft skin or shell will experience buckling when it reaches critical buckling load limit. As the compression load increases in time, super-stiffeners concept come into play to carry the load. Super-stiffeners concept is known as the

stiffeners and small parcels around the neighbouring skin [65]. The any failure mode appears when the super-stiffeners reach their buckling stress limit. In other words, the overall stability and load-carrying capacity of the structure are determined by the buckling behaviour of these super-stiffeners. Therefore, length of the stringers on those areas are crucial to reduce potential buckling failure mode. Prior to buckling failure mode, the load is carried by both the stringers and the skin until the compression stress in the skin is equal to the critical compression stress. Figure 3.8 shows an example of compression buckling of a civil aircraft fuselage.

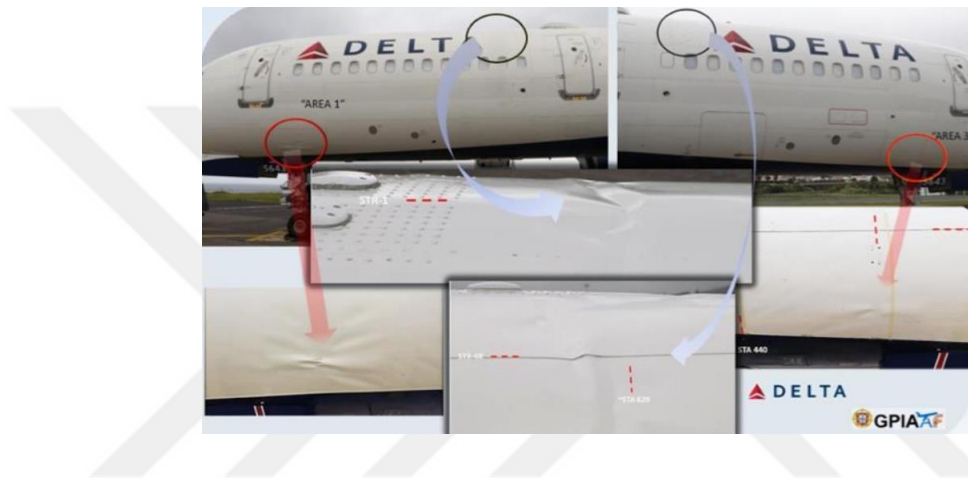


Figure 3.8 Compression buckling [72]

3.1.2 Material Choice for Metallic Panels

There are different types of metallic alloys that are used on metallic panels and these alloys are selected according to some criteria [65]:

- ❖ **Static strength:** The material that is used should withstand limit loads without experiencing permanent deformation and endure ultimate loads without undergoing failure.
- ❖ **Stiffness:** Deformation of the material that is used at limit loads must not compromise operations safely.
- ❖ **Weight:** Achieving aircraft performance is provided with a low weight.

- ❖ Corrosion: The material that is used must withstand surface and stress corrosion.
- ❖ Manufacturability: The material that is used must be suitable in terms of producibility.
- ❖ Cost: The material that is used has low raw material, detail manufacturing, assembly and, life of cycle costs.
- ❖ Other parameters: Temperature, impact resistance, electrical transparency, erosion and abrasion, fire zones, lightning.

Some metallic alloys which are employed in aerostructures mostly are given below [65]:

- Al 2XXX alloys clads with high strength and machinability as well as workability and surface finish capability.
- Al 7XXX alloys with high static properties, especially for lower shell structures.
- Titanium alloys with lightweight, excellent corrosion and fire resistance as well as high static properties.
- Al-Li alloys with high fatigue and damage tolerance attributes.

3.1.3 Manufacturing Methods for Metallic Panels

As part of the manufacturing process, panels are commonly shaped and machined [65]. Cylindrical fuselage panels often undergo roll-forming, while double-curved panels are subjected to stretching in two directions.

The size of the panel, particularly its width, must be verified with the available facility and tools. Due to limitations in stretching technology, panels intended for stretching should have restricted size and thickness.

There are two types of machining methods used in metallic panel manufacturing; mechanical and chemical machining [65]. Mechanical machining is used on metallic parts with using material removal tools so that the machined parts have desired specifications and shapes and it is a manufacturing process. On the other hand, chemical machining is a similar manufacturing process with using chemicals which remove materials on the parts. Chemical machining is significantly faster than mechanical machining. Additionally, chemical machining provides a generally superior surface condition, and the resulting panel exhibits better fatigue resistance compared to conventionally mechanical machining panels.

3.1.4 Metallic Stringer Design

The fuselage undergoes loading due to pressurization (except for helicopters and some sections of fixed wing aircraft) as well as mechanical loads such as bending and shear forces. To withstand these load conditions, the fuselage skin panels are reinforced with frames and stringers as mentioned in Chapter 3.1. The role of the stringers is to provide support for axial loads resulting from compression loads and tension loads or both [65]. Figure 3.9 shows typical stringer and its section with surrounding aero structures.

Typically, Aluminium Alloy 2024 T3 and T42 are employed for formed sheet metal and machined applications, while Aluminium Alloy 2024 T42 and T3511 are utilized for extrusions. Aluminium Alloy 2024 exhibits susceptibility to corrosion cracking but it offers superior fatigue life when compared to Aluminium Alloy 7075. Extrusion processes for stringers are often chosen for regions subjected to high tension and compression loads.

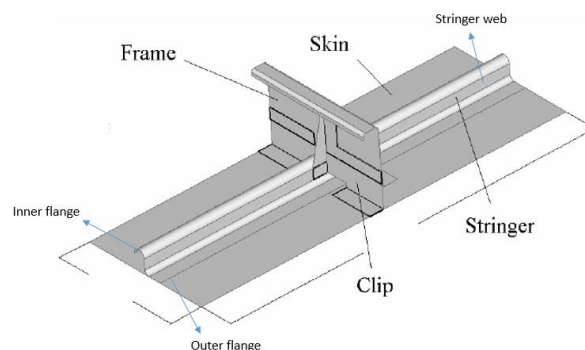


Figure 3.9 Typical stringer profile [73]

3.1.4.1 Different Cross Sections of Stringers

There are different types of stringers for each assembly and load conditions. All stringer types are generally used in an aircraft to provide additional strengths to the aero structures [65].

Z Cross Section Stringer Design: This configuration provides robust compression resistance. Z-shaped stringers are conveniently produced using bent sheets, machining materials and, extruded materials. Attaching this stringer design through riveting or its installation is uncomplicated due to the fact that it is easy to reach to the outer flange. These types of stringers commonly possess a flanged edge to prevent cap bending.

2 Cross Section Stringer Design: These types of stringers share nearly identical characteristics with the z design stringers, although their compression resistance is slightly less due to their lower inertia. Nevertheless, they exhibit improved tolerance to damage.

I Cross Section Stringer Design: This type of stringer is employed solely for areas where are subjected to significant compression loads. Compression warping failure of the stringer effectively can be postponed because of the mirror design, but it also results in heightened weight and expenses. Additionally, the installation of this stringer is tough due to the presence of an inner flange. It is usually produced through extrusion.

L Cross Section Stringer Design: This configuration is highly beneficial in terms of tension loading conditions but it has poor behaviour under compression applications. These stringers can be easily manufactured using roll forming or extrusion techniques and installation process is not considered challenging.

C Cross Section Stringer Design: This kind of stringer has similarities with z cross section stringer in terms of the inertia that affects the buckling and bending behaviour of a panel. This inertia is provided with additional top flange. On the other hand, assembling process of this stringer to panel can be difficult because of its close form structure.

Omega or Hat Cross Section Stringer Design: This stringer configuration offers notable compression resistance. The omega design can be formed by coupling two z design stringers in a back-to-back adjustment. However, certain limitations arise with the omega stringer design, stemming from particular factors such as minimum thickness, weight considerations, and geometric constraints. The installation process is challenging due to limited accessibility.

Figure 3.10 demonstrates to stringer sections which have different shapes.

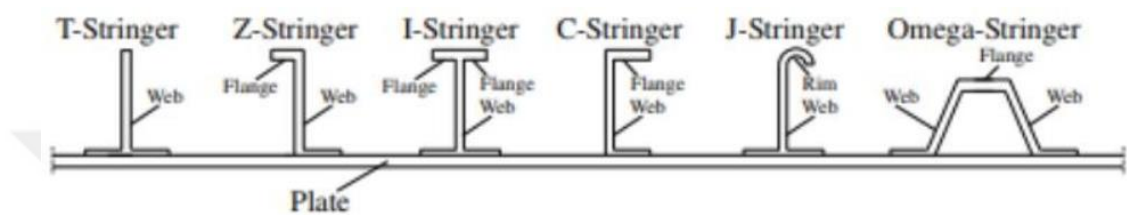


Figure 3.10 Different kinds of stringer types [74]

When determining the spacing of stringers, several factors need to be taken into consideration. These include ensuring stability in compression, tension and, shear zones, maintaining damage tolerance for the skin areas, accommodating the thickness of the skin panels, and conforming to assembly constraints etc [65].

Stringer sizing is also a crucial aspect to calculate the load-carrying capacity of the metallic stiffened panel and is an integrated part of optimizing the stiffened panel design.

3.1.5 Metallic Frame Design

Frames serve as integral components in both pressurized and non-pressurized sections of a fuselage, ensuring the purpose of maintaining the cross-sectional shape of the cabin during conditions of imposed limit loads [65]. Fuselage sections are split into stiffened panels, combined with skin, stringers, and frames, as demonstrated in Figure 3.11. Frames are fastened either directly to the skin or indirectly through shear webs, clips, or cleats etc. or in some cases, they possess integrated flanges that eliminate the need for clips or attachments.

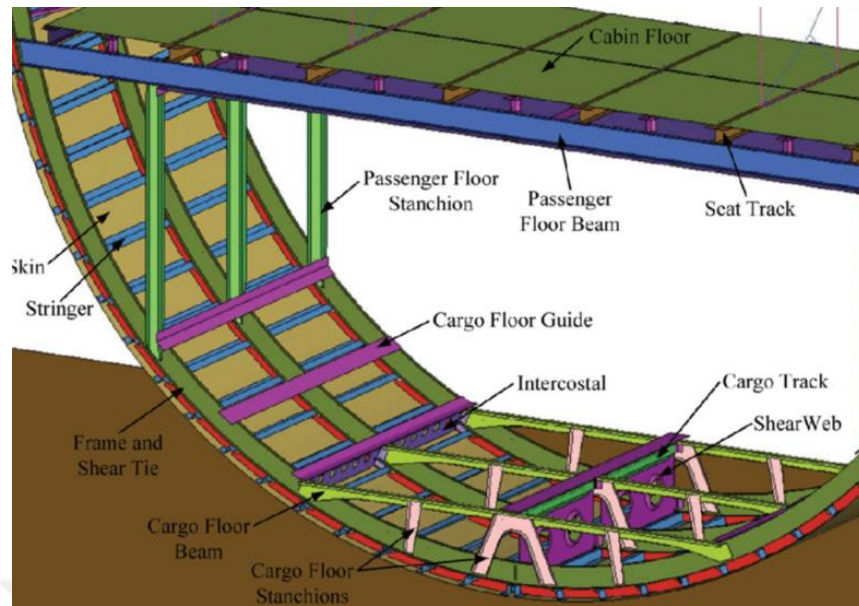


Figure 3.11 Typical fuselage section with frames and stringers [75]

Within an aircraft's fuselage structure, the primary loads resulting from fuselage bending, shear, buckling, torsion, and cabin pressure are predominantly borne by the skin, which is reinforced by longitudinal stringers as discussed in Chapter 3. The frames make a minimal direct contribution to carrying these loads. While the stringers generally take bending and buckling loads, the frames have torsion loads capability. When it comes to skin, all pressure loads and its derivatives are compensated by skin element. Nevertheless, frames hold a critical role in providing support to the skin-stringer panel structure. The frames facilitate the distribution of concentrated loads throughout the fuselage shell [65].

When determining the spacing between two successive frames, buckling failure mode needs to be taken into consideration. These defines the stability of the fuselage skin under compression loads, assembly constraints, damage tolerance, and other relevant considerations.

Frame sizing is also a crucial aspect to calculate the load-carrying capacity of the metallic stiffened panel and is an integrated part of optimizing the stiffened panel design [65].

Frames typically are manufactured using materials like Al 2XXX and 7XXX alloy series. Al Alloy 2XXX series materials offer superior fatigue life compared to Al Alloy 7XXX series materials that demonstrates higher static strength. Various manufacturing methods are employed, including extrusion, machining, and sheet metal bending. There are several important considerations regarding manufacturing methods. For example, when the long-machined frames are used, this situation may introduce assembly challenges so making sheet metal frames a preferred choice whenever feasible.

3.1.5.1 Different Cross Sections of Frames

The simplest and least complex approach, involves utilizing frames with a constant cross-sectional profile. In other words, using a limited number of frames that have different cross section across the whole the whole panel leads to fewer splices, resulting in reduced time, weight, and manufacturing expenses [65]. These frames can be manufactured through sheet metal bending, machining process or extrusion.

Machined profiles are frequently employed in regions where required to high load resistance [65]. These machined profiles can undergo shape optimization to align with the load requirements. The frame design must also account for deformations which are caused by local and global buckling.

Z cross section profiles offer distinct advantages in terms of bending or buckling behaviour and automated assembly processes. C cross section or L cross section profiles are preferred due to their efficient utilization of semi-finished materials in terms of machinability. Some kinds of different frame profiles can be seen in Figure 3.12.

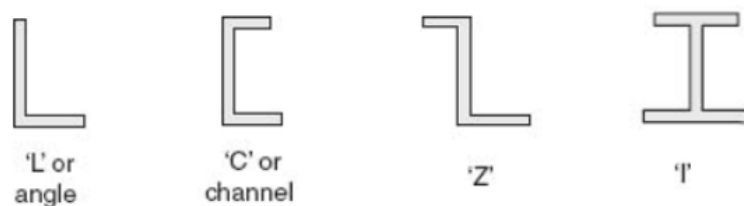


Figure 3.12 Different frame section profiles [76]

3.2 Metallic Sandwich Panel Design

Another viable configuration for metallic panel structures is the sandwich structure. This design consists of a low-density core positioned between two face sheets. Essentially, this concept benefits the principles that are akin to plate structures, involving skin and stringers. In this scenario, the face sheets handle the normal or axial forces while the sandwich core between them fulfils the role of providing diagonal shear functionality [65]. However, the distinctive characteristic of the sandwich structure lies in its ability to carry loads in two directions.

These structures find widespread application in assemblies like fuselage sidewall panels, floor and ceiling panels etc. Figure 3.13 shows the general layout of sandwich components.

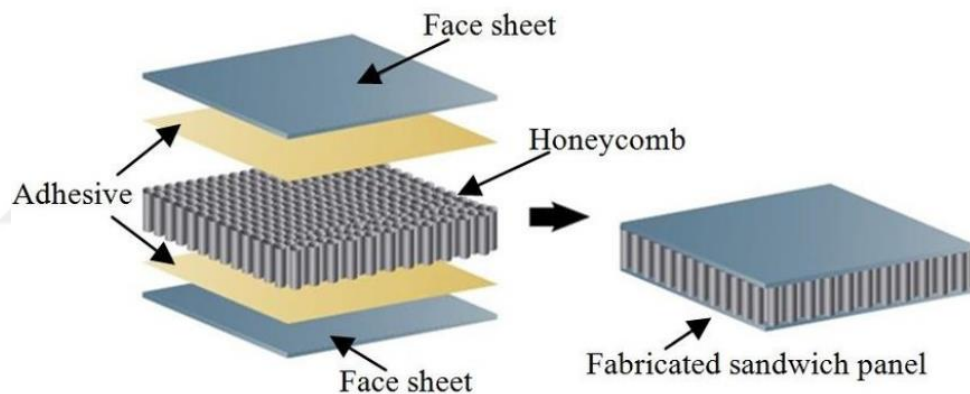


Figure 3.13 Definition of honeycomb sandwich structure panel [77]

Typically, a sandwich structure is made of two face sheets that are bonded with adhesive to a thick and lightweight core relative to face sheets or plates. Since face sheets serve as the primary load carrying components within sandwich structures, they should have adequate thickness. Face sheets carry in-plane, shear loads such as bending, buckling, transverse shear, torsion etc [65]. These face sheets can be composed of either metallic or polymer materials. Given that face sheets in sandwich structures are typically quite thin, various forms of local reinforcements can be used for additional strength, facilitating effective load transfer and extra resistance against

buckling and bending failure modes. These supplementary elements can encompass doublers, joints, stiffeners, ribs etc.

When it comes to metallic honeycomb cores that have fairly low density, these components give good mechanical properties in terms of shear and compressive loading condition [65]. Honeycomb structures are made by bonding ribbons together to form cells, and then expanding them to achieve the desired cell shape. In the transverse direction (W), pulling results in the expansion of the honeycomb. The core's cell shape may take on hexagonal (honeycomb) or square flex-core configurations and can be produced from either metal or polymer materials. The direction of ribbons (L) is vertical to the cell expansion direction (transverse direction) or parallel to the main load direction (longitudinal direction) of the primary panel. Notably, the thickness of the core significantly affects the material's capacity to withstand shear and compression stresses [65]. An example of core structure can be seen in Figure 3.14.

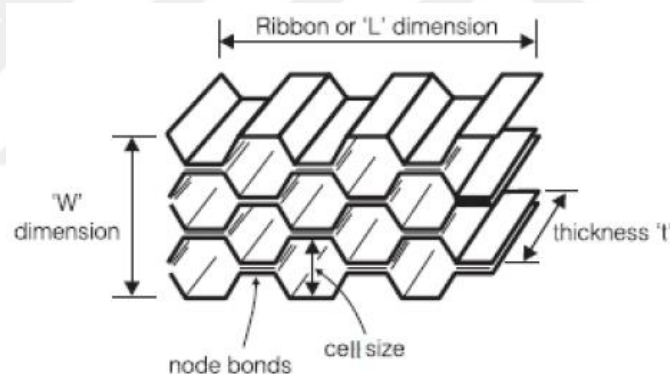


Figure 3.14 Honeycomb structure [78]

The utilization of sandwich structures is rooted in their favourable ratio of flexural strength that means buckling and bending stiffness to weight [65]. In other words, there are stiffness increases more against minimal weight gain. To demonstrate the advantages in terms of bending stiffness, Figure 3.15 illustrates the relative enhancements in bending stiffness, flexural strength, and weight for a specific sandwich example. The introduction of a low-density core material results in only a small weight increase, whereas the enhancement in bending stiffness can be substantial, spanning orders of magnitude.

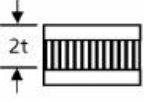
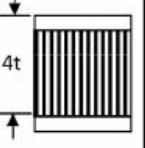
	Solid Material	Sandwich Construction	Thicker Sandwich
			
Stiffness	1.0	7.0	37.0
Flexural Strength	1.0	3.5	9.2
Weight	1.0	1.03	1.06

Figure 3.15 Effect of core thickness on sandwich panel stiffness [79]

In terms of aerostructure design, sandwich panel structures must align with various ranges of design criteria including structural strength and tolerance, manufacturability, lightning and electrical protection, corrosion, interchangeability, aerodynamic, maintenance etc.

Water ingress is crucial concern for life span of a honeycomb sandwich panel structure [65]. When the sandwich structure is subjected to direct environment conditions, condensed liquids, and contamination from other liquids, this situation occurs. To mitigate water ingress, it is obligatory to sustain a minimum face sheet thickness and the most available curing pressure should be applied during the curing process. Inserts can trigger to water ingress and the assembly process of inserts should be done cautiously during the installation. Additionally, the application of epoxy and an additional layer of adhesive film can further alleviate water ingress issues. Once the sandwich structure is composed, it is compulsory to seal the panel edges with edge sealant [65].

Advantages and disadvantages of sandwich structures can be listed below [65].

Advantages of sandwich structures:

- Superior stiffness to weight ratio.
- Enhanced inertia.

- Reduced material consumption in terms of manufacturability.
- Fewer structural components.
- Ease of handling.
- Acoustic benefits.
- Good thermal conductivity.

Disadvantages of sandwich structures:

- Challenges related to water ingress and sealing.
- Problems with damage tolerance.
- Limited applications in heavily loaded structures.
- Complex repair procedures.
- Complex mechanical joint performance with other parts.
- Limited fastener usage such as rivet and bolt.

3.2.1 Material and Manufacturing Methods for Sandwich Structures

While face sheets are generally made from Al 2XXX alloy series for a metallic sandwich panel structure in aerospace applications, aluminium cores which are made from Al 5XXX alloy series are extensively used in the same structure [65]. Specifications of Al 2XXX alloy series are described in Chapter 3.1.2. As for Al 5XXX alloy series give high strength and excellent corrosion resistance to the panel and this series is non heat treatable unlike Al 2XXX.

In general, three distinct methods exist for the manufacturing of a sandwich structure [65].

Single Shot Process: The outer and inner skins are bonded to the core during the curing process, they don't have pre-curing step. Sandwich structures which are produced

using single shot process exhibit superior structural features, strong bonding capabilities, and reduced weight. Nonetheless, there is a potential challenge for blocking water ingress as well as complexity of the method and cost.

Co-Bonding Process: The outer skin is cured while the inner skin remains uncured, bonding to the core during the curing process like in single shot process. This method offers benefits such as simplified tooling, straightforward manufacturing, and good structural features. Nonetheless, there are some challenges associated with quality control, potential weight increase, and the risk of water ingress.

Multiphase Process: The outer and inner skins are cured and subsequently they are bonded to the core structure. The advantages of this approach include good structural properties, and reduced susceptibility to water ingress. However, there are particular limitations such as manufacturing tolerances and extended curing slots.

3.3 Metallic Panel Designs of the Study

In this subtitle, six different metallic panel structure configurations are given and these panels are employed for linear eigenvalue buckling analysis.

3.3.1 Z Cross Section Stringer-Skin Metallic Panel

Metallic skin that is made from clad aluminium 2024 T3 has a length of 1100 mm, a width of 560 mm, an arc length of approximately 556 mm and, a radius of 2000 mm. The skin thickness is 1.016 mm.

This metallic panel consists of 4 frames, 5 z cross section stringers and considerable number of cleats that are connect the frames to the skin and has a physically interface the stringers. Figure 3.16 shows the metallic skin panel with other structural elements. While the frame spacing is 300 mm, the stringer spacing is 120 mm. The metallic panel is bi-directional stiffening with continuous stringers.

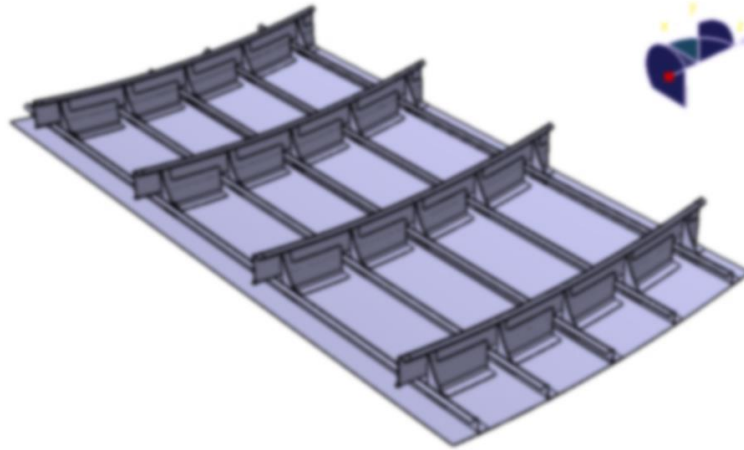


Figure 3.16 Z cross section stringer-skin panel

The z cross section stringer is made of 1.016 mm thick clad aluminium 2024 T3 and each stringer has a length of 1100 mm. General dimensions of the z stringer can be seen in Figure 3.17. Z cross section stringer is modelled in order to have the one line fastener path. The total mass of the z cross section stringers and the skin is 2.57 kg.

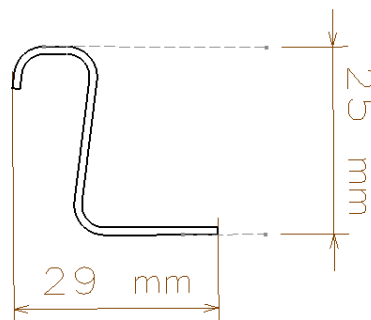


Figure 3.17 General dimensions of z stringer

3.3.2 Omega Cross Section Stringer-Skin Metallic Panel

Metallic skin that is made from clad aluminium 2024 T3 has a length of 1100 mm, a width of 560 mm, an arc length of approximately 556 mm and, a radius of 2000 mm. The skin thickness is 1.016 mm.

This metallic panel consists of 4 frames, 5 omega cross section stringers, considerable number of clips that are connect the frames to the skin and, considerable number of L

angles that have a physically interface the stringers. Figure 3.18 shows the metallic skin panel with other structural elements. While the frame spacing is 300 mm, the stringer spacing is 120 mm. The metallic panel is bi-directional stiffening with continuous stringers.

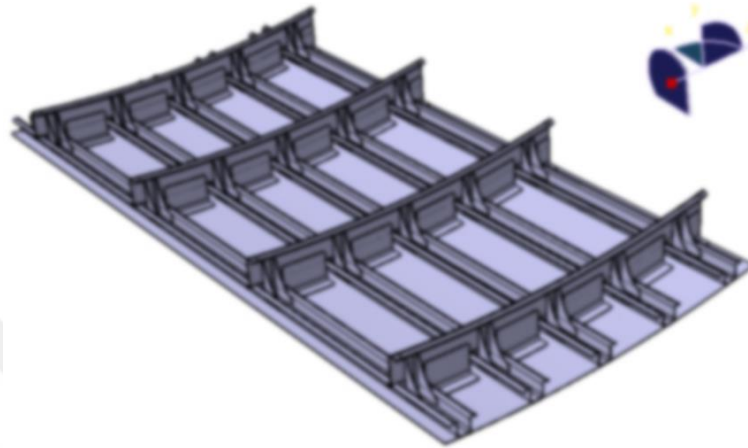


Figure 3.18 Omega cross section stringer-skin panel

The omega cross section stringer is made of 0.6 mm thick clad aluminium 2024 T3 due to the equivalence parameter of the cross-section areas among the stringers and each stringer has a length of 1100 mm. General dimensions of the omega stringer can be seen in Figure 3.19. Omega cross section stringer is modelled in order to have the one line fastener path. The total mass of the omega cross section stringers and the skin is 2.57 kg.

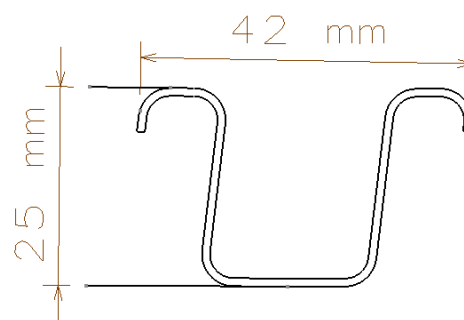


Figure 3.19 General dimensions of omega stringer

3.3.3 C Cross Section Stringer-Skin Metallic Panel

Metallic skin that is made from clad aluminium 2024 T3 has a length of 1100 mm, a width of 560 mm, an arc length of approximately 556 mm and, a radius of 2000 mm. The skin thickness is 1.016 mm.

This metallic panel consists of 4 frames, 5 c cross section stringers, considerable number of cleats that are connect the frames to the skin and has a physically interface the stringers. Figure 3.20 shows the metallic skin panel with other structural elements. While the frame spacing is 300 mm, the stringer spacing is 120 mm. The metallic panel is bi-directional stiffening with continuous stringers.

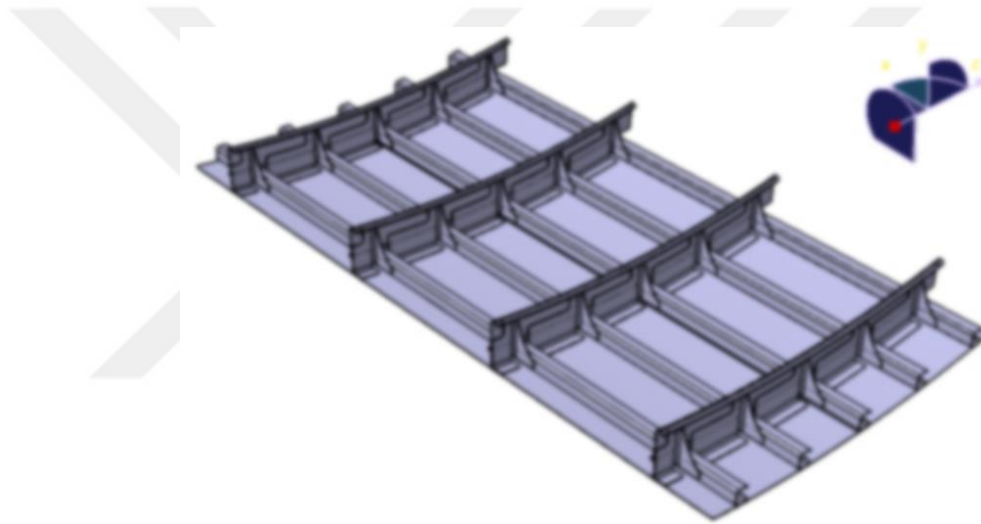


Figure 3.20 C cross section stringer-skin panel

The c cross section stringer is made of 0.8 mm thick clad aluminium 2024 T3 because of the equivalence parameter of the cross-section areas among the stringers and each stringer has a length of 1100 mm. General dimensions of the c stringer can be seen in Figure 3.21. C cross section stringer is modelled in order to have the one line fastener path. The total mass of the z cross section stringers and the skin is 2.57 kg.

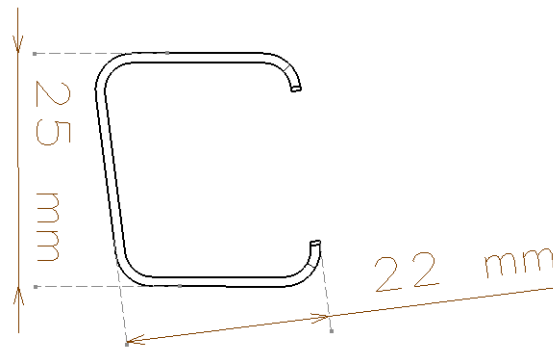


Figure 3.21 General dimensions of c stringer

3.3.4 Metallic Honeycomb Sandwich Panel

Metallic honeycomb sandwich panel consists of three major components. It has the outer skin or outer face plate that is made from clad aluminium 2024 T3 has a length of 1100 mm, a width of 560 mm, an arc length of approximately 556 mm and, a radius of 2000 mm. The outer skin thickness is 1.016 mm. When it comes to the inner skin or inner face plate of the panel, it has the same material with outer face plate but different dimensions and thickness which is 0.3 mm. The length and the width of the inner face plate is almost 1069 mm and 530 mm, respectively. The radius of the inner face plate is nearly 1975 mm. Honeycomb core material which has 24.7 mm thick is made from aluminium alloy 5052 and its cell size 3.2 mm and nominal density 3.1 pounds per cubic feet (pcf). The length and the width of the honeycomb core is almost 1070 mm and 531 mm, respectively. The total mass of the panel is 2.92 kg. Figure 3.22 shows the metallic honeycomb sandwich panel with other structural elements.

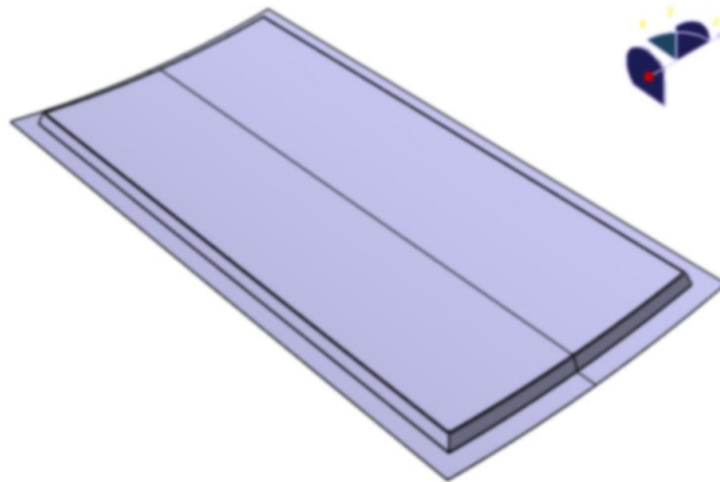


Figure 3.22 Metallic honeycomb sandwich panel

3.3.5 Metallic Honeycomb Sandwich Panel with Longitudinal Doublers

Metallic honeycomb sandwich panel with longitudinal doublers consist of four major components. It has the outer skin or outer face plate that is made from clad aluminium 2024 T3 has a length of 1100 mm, a width of 560 mm, an arc length of approximately 556 mm and, a radius of 2000 mm. The outer skin thickness is 1.016 mm. When it comes to the inner skin or inner face plate of the panel, it has the same material with outer face plate but different dimensions and thickness which is 0.3 mm. The length and the width of the inner face plate is almost 1069 mm and 530 mm, respectively. The radius of the inner pace plate is nearly 1975 mm. Honeycomb core material which has 24.7 mm thick is made from aluminium alloy 5052 and its cell size 3.2 mm and nominal density 3.1 pcf. The length and the width of the honeycomb core is almost 1070 mm and 531 mm, respectively. In addition, there are two metallic doublers that are made from clad aluminium 2024 T3 on the panel in longitudinal direction and their thicknesses are 0.3 mm. The length and the width of the double is roughly 1100 mm and 70 mm, respectively. The total mass of the panel is 3.00 kg. Figure 3.23 shows the metallic honeycomb sandwich panel with the longitudinal doublers as well as its section.

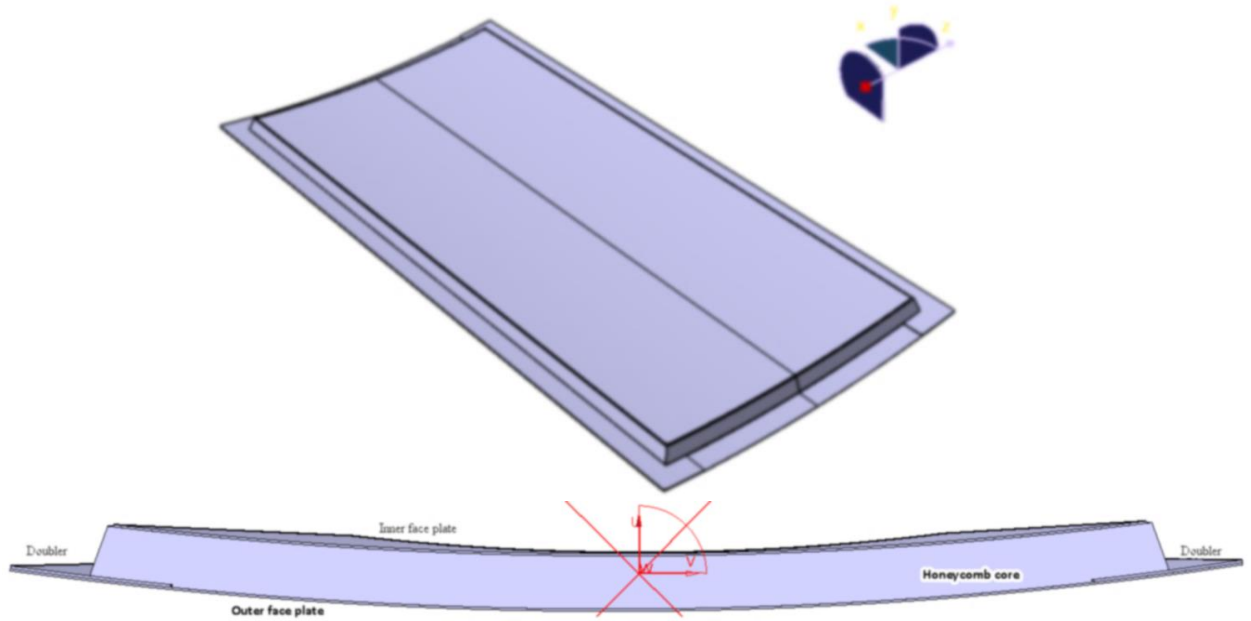


Figure 3.23 Metallic honeycomb sandwich panel with longitudinal doublers

3.3.6 Metallic Honeycomb Sandwich Panel with Transverse Doublers

Metallic honeycomb sandwich panel with transverse doublers consist of four major components. It has the outer skin or outer face plate that is made from clad aluminium 2024 T3 has a length of 1100 mm, a width of 560 mm, an arc length of approximately 556 mm and, a radius of 2000 mm. The outer skin thickness is 1.016 mm. When it comes to the inner skin or inner face plate of the panel, it has the same material with outer face plate but different dimensions and thickness which is 0.3 mm. The length and the width of the inner face plate is almost 1069 mm and 530 mm, respectively. The radius of the inner pace plate is nearly 1975 mm. Honeycomb core material which has 24.7 mm thick is made from aluminium alloy 5052 and its cell size 3.2 mm and nominal density 3.1 pcf. The length and the width of the honeycomb core is almost 1070 mm and 531 mm, respectively. In addition, there are two metallic doublers that are made from clad aluminium 2024 T3 on the panel in transverse direction and their thicknesses are 0.3 mm. The length and the width of the double is roughly 70 mm and 560 mm, respectively. The total mass of the panel is 2.96 kg. Figure 3.24 shows the metallic honeycomb sandwich panel with the transverse doublers as well as a part of its section.

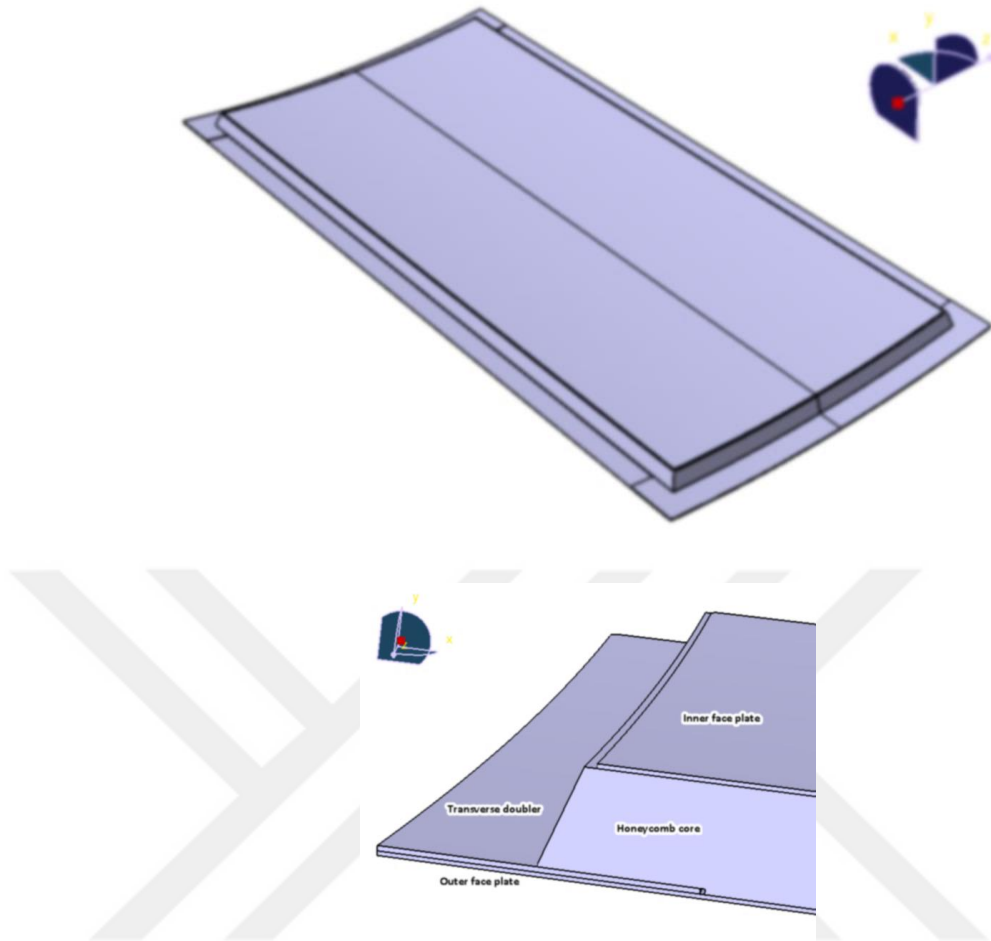


Figure 3.24 Metallic honeycomb sandwich panel with transverse doublers

3.3.7 Summary of the Geometrical Properties and Material Properties

All geometrical properties of the panel elements are given in the Table 3.1. In addition, Figure 3.25 and Figure 3.26 demonstrates the specifications of the aluminium alloy 2024 T3 and aluminium honeycomb core 5052, respectively. The thickness of the entire panels is approximately 26 mm. In this study, only skin and stringer elements are taken into consideration for skin-stringer metallic panels. Frames and other secondary elements are omitted in order to investigate the critical buckling loads of the skin-stringer panels. This is mandatory to simplify the model geometry so that the results are understandable and comparable. On the other hand, the metallic honeycomb sandwich panels are considered as a whole assembly.

Table 3.1 Geometrical and material properties of metallic panels

Geometric and Material Properties	Structural Components							
	Z Cross Section Stringer	Omega Cross Section Stringer	C Cross Section Stringer	Skin or Outer Face Plate	Inner Face Plate	Honeycomb Core	Doubler (LD)	Doubler (TD)
Thickness	1.016 mm	0.6 mm	0.8 mm	1.016 mm	0.3 mm	24.7 mm	0.3 mm	0.3 mm
Length	1100 mm	1100 mm	1100 mm	1100 mm	1069 mm	1070 mm	1100 mm	70 mm
Height	25 mm	25 mm	25 mm	N/A	N/A	N/A	N/A	N/A
Width	29 mm	42 mm	22 mm	560 mm	530 mm	531 mm	70 mm	560 mm
Material	Al 2024 T3	Al 2024 T3	Al 2024 T3	Al 2024 T3	Al 2024 T3	Al 5052	Al 2024 T3	Al 2024 T3

Specification	AMS 4037 and AMS-QQ-A-250/4*				
Form	Sheet				
Temper	T3				
Thickness, in.	0.008-0.009	0.010-0.128		0.129 - 0.249	
Basis	S	A	B	A	B
Mechanical Properties:					
F_u , ksi:					
L	64	64	65	64	66
LT	63	63	64	63	65
ST	...	...	...	...	...
F_y , ksi:					
L	47	47	48	47	48
LT	42	42	43	42	43
ST	...	...	...	...	...
F_{uT} , ksi:					
L	39	39	40	39	40
LT	45	45	46	45	46
ST	...	...	...	...	...
F_{uH} , ksi:	39	39	40	40	41
$F_{uH}^{3/4}$, ksi:					
(e/D = 1.5)	104	104	106	106	107
(e/D = 2.0)	129	129	131	131	133
$F_{uH}^{3/4}$, ksi:					
(e/D = 1.5)	73	73	75	73	75
(e/D = 2.0)	88	88	90	88	90
ϵ , percent:					
LT	10	d	...	d	...
E , 10^3 ksi	10.5				
E_{av} , 10^3 ksi	10.7				
G , 10^3 ksi	4.0				
μ	0.33				
Physical Properties:					
ω , lb/in. ³	0.100				
C, K, and α					

Figure 3.25 Material specifications of Al 2024 T3 [80]

Cell · Material · Gauge	HEXCEL Honeycomb Designation	ρ^*	Compressive						σ^{**}	Plate shear					
			Bare			Stabilised				L direction			W direction		
			Strength kPa		Modulus MPa	Strength kPa		Modulus MPa		Strength kPa		Modulus MPa	Strength kPa		Modulus MPa
			typical	minimum		typical	minimum			typical	minimum		typical	minimum	
5052 Hexagonal Aluminium Honeycomb															
1/16 - 5052 - 0.0007	101	5998p		6274p		1896p		3516p		621p	2206p		276p		
1/16 - 5052 - 0.001	144	10204x		10342x		2896x		5343x		724x	3585x		365x		
1/8 - 5052 - 0.0007 †	50	1862	1379	1999	1482	517	896	1448	1069	310	896	621	152		
1/8 - 5052 - 0.001 †	72	3585	2586	3758	2792	1034	1793	2344	1965	483	1517	1158	214		
1/8 - 5052 - 0.0015 †	98	5998	4482	6274	4688	1655	3103	3482	3137	676	2206	1875	283		
1/8 - 5052 - 0.002 †	130	9653	6895	10135	7584	2413	5171	4999	4619	931	3137	2758	372		
1/8 - 5052 - 0.003	192	15168p		16030p		6205p		7584p			4309p				
5/32 - 5052 - 0.0007	42	1379	1034	1482	1103	379	621	1138	827	255	689	483	131		
5/32 - 5052 - 0.001	61	2723	1965	2827	2068	758	1276	1862	1482	386	1207	862	182		
5/32 - 5052 - 0.0015	85	4757	3378	4964	3689	1344	2344	2896	2551	579	1862	1482	248		
5/32 - 5052 - 0.002	111	7446	5309	7791	5516	1965	3964	4068	3723	786	2586	2261	320		
5/32 - 5052 - 0.0025 ‡	135	10549	7377	11032	8136	2551	5516	5240	4757	965	3275	2896	386		
3/16 - 5052 - 0.0007	32	896	621	931	689	234	414	827	552	186	483	317	99		
3/16 - 5052 - 0.001 †	50	1862	1379	1999	1482	517	896	1448	1069	310	896	621	152		
3/16 - 5052 - 0.0015 †	70	3447	2482	3620	2654	1000	1724	2275	1931	469	1482	1103	207		
3/16 - 5052 - 0.002 †	91	5309	3861	5585	4137	1517	2689	3172	2827	621	2068	1682	265		
3/16 - 5052 - 0.0025	111	7446	5309	7791	5516	1965	3964	4068	3723	786	2586	2261	320		
3/16 - 5052 - 0.003	130	9653	6895	10135	7584	2413	5171	4999	4619	931	3137	2758	372		
1/4 - 5052 - 0.0007	26	586	414	655	483	138	276	586	414	145	345	221	76		
1/4 - 5052 - 0.001	37	1138	827	1207	896	310	517	965	689	221	586	393	112		
1/4 - 5052 - 0.0015	54	2206	1655	2344	1724	621	1034	1620	1241	345	1034	724	165		
1/4 - 5052 - 0.002	69	3309	2413	3482	2551	965	1586	2206	1827	455	1448	1069	205		
1/4 - 5052 - 0.0025	83	4619	3447	4757	3516	1310	2310	2827	2482	565	1827	1379	244		
1/4 - 5052 - 0.003	96	5861	4344	6067	4551	1620	2965	3413	3068	662	2172	1827	279		
1/4 - 5052 - 0.004 †	127	9377	6688	9791	7239	2344	4999	4826	4482	896	3034	2689	364		
3/8 - 5052 - 0.0007	16	207	138	310	138	69	172	310	221	83	207	138	48		
3/8 - 5052 - 0.001	26	586	414	655	483	138	276	586	414	145	345	221	76		
3/8 - 5052 - 0.0015	37	1138	827	1207	896	310	517	965	689	221	586	393	112		
3/8 - 5052 - 0.002	48	1793	1310	1862	1379	483	827	1379	1000	296	862	586	146		
3/8 - 5052 - 0.0025 ‡	59	2551	1862	2689	1965	724	1241	1793	1379	379	1172	793	179		
3/8 - 5052 - 0.003	67	3172	2310	3344	2448	931	1517	2137	1758	448	1379	1034	200		
3/8 - 5052 - 0.004	86	4964	3447	5137	3689	1379	2482	2965	2620	593	1931	1572	254		
3/8 - 5052 - 0.005 ‡	104	6688	4826	7033	5171	1827	3482	3758	3447	724	2413	2068	300		

Figure 3.26 Material specifications of aluminium honeycomb core 5052 [62]

CHAPTER 4

NUMERICAL ANALYSIS

4.1 General Numerical Analysis Phenomena

The standard method for conducting structural analysis on structural elements with complicated geometry under multiple loading conditions involves the use of finite element software [81]. Especially, there are complex problems in aerostructures to have to be solved. Finite element approach is grounded in a special case of the Ritz variational method since both approaches are fundamentally equivalent due to the fact that they each employ a set of presumed displacement functions to derive approximate solutions for given plate problems [48]. Moreover, both methods utilize the principle of minimum potential energy to create these functions stationary [48].

Traditional analytic approaches are limited in their applicability for solving all cases and these approaches seem unsuitable for the sophisticated cases. Therefore, since the presence of the different geometries and properties of the metallic panels imply to use a finite element software including numerical methods, the linear eigenvalue buckling analysis is used in this study.

In many structural cases, numerical analysis methods may be excessively hermetic and time-consuming for hand written calculations. Hence, the traditional approach converts to finite element tools and these tools also give precise results within a reasonable timeframe. Nevertheless, it is always crucial to employ an approximate method so that the meticulous examinations of the results can be controlled.

A wide arrangement of the finite element software options is available in the engineering software world. These programs may share similarities each other's and but, all of them exhibit more or less identical implementation and processing, giving to an anticipated similarity in the accuracy of results.

Skin-stringer panel assemblies or sandwich panel assemblies typically cannot be treated as orthotropic plates [48]. Therefore, the finite element method must be

employed within specified constraints. In this study, Altair Hyperworks software will be employed as a pre-processor for meshing the metallic panel structures and it will be used as a post-processor to simulate mechanical behaviour of the panel structures. Moreover, Altair Optistruct solver will be used as a solver for finite element models.

The objective is to investigate the linear buckling capacity of metallic skin-stringer panels that have different cross section stringer profiles and metallic honeycomb sandwich panels that have different structural configurations using a linear eigenvalue methodology throughout the analysis. To determine the elastic buckling capacity or critical buckling load along with the corresponding buckling modes, a linear eigenvalue buckling analysis is conducted under the uniaxial compression load condition. The material properties and boundary conditions will maintain same throughout the entire study.

4.2 Linear Eigenvalue Buckling Analysis Method

Linear buckling analysis, alternatively referred to as eigenvalue buckling analysis, identifies the elastic critical instability loads and their corresponding modes of instability under compressive loads [82]. This kind of analysis does not include material or geometrical nonlinearities, imperfections, and pre-buckling equilibrium configurations and so the stability equations take the form of linear differential equations with constant coefficients [52]. Additionally, it estimates the sensitivity of a structure that is subjected to a specific parametric loading, to particular deformation patterns. The linear buckling or eigenvalue buckling analysis is mentioned under Chapter 2 in detailed.

In this sort of the finite element analysis, achieving the critical buckling load involves a singular stiffness matrix solution. This means that the objective is to identify loads that present the model stiffness matrix singular [82]. Therefore, this does not allow to obtain nontrivial solutions to the problem and it can be shown in Eq. 4.1:

$$[K^{AB}] * \{v^A\} = 0 \quad (4.1)$$

where:

K^{AB} : Tangent stiffness matrix under loads such as distributed loads, pressures etc.

v^A : Nontrivial displacement solution vector.

A, B : Degrees of freedom .

The buckling loads are determined in relation to the base situation of the structure. In other words, the base situations of the structures may have pre-loads, “ P ”, even though these loads of the classical eigenvalue problems typically are equal to zero. Additionally, when the geometric nonlinearity is excluded, the base situation geometry is also equal to the original geometry of the structure [82]. In this study, all geometric and material nonlinearities are omitted and there are no any pre-load conditions.

Incremental load that can symbolize with “ Q^B ” is specified in the eigenvalue buckling analysis process. Since the magnitude of incremental loading will be adjusted by the load multipliers, “ λ_i ”, this magnitude is insignificant and it can be calculated with Eq. 4.2 [82]:

$$([K_0^{BA}] + \lambda_i * [K_{\nabla}^{BA}]) * \{v_i^A\} = 0 \quad (4.2)$$

where:

K_0^{BA} : Stiffness matrix associated with the base situation of the structure, if any pre-loads included.

i : related buckling number of the buckling modes.

λ_i : Eigenvalues.

K_{∇}^{BA} : Initial differential stress and load stiffness matrix because of Q^B .

v_i^A : Eigenvectors or the buckling mode shapes.

K^{AB} and K^{BA} : Symmetric stiffness matrices.

Considering the information provided in Eq. 4.2, it is straightforward to deduce that the expression for the critical buckling load is given in Eq. 4.3:

$$P^B = \lambda_i Q^B \quad (4.3)$$

In linear eigenvalue buckling analysis, the lowest value of the eigenvalues is taken into consideration and the subsequent eigenvalues have relevance mainly for local buckling and should be interpreted cautiously, as they may not always carry physical significance. In other words, this represents that the minimum loading magnitude will lead to substantial deflections or deformations in the structure's geometry, even leading to its fall [82]. These deformations are normalized deformation shapes with the specified boundary conditions and applied loads.

In a nutshell, once the related eigenvalue is acquired and it is multiplied by the applied load, it signifies the initiation of panel buckling when the calculated buckling is localized in a specific area. Afterwards, when the critical buckling load is calculated on the panel, it signifies global buckling and, thus, the maximum load-carrying capacity of the panel.

The buckling mode shapes are presented as normalized vectors, without the actual magnitudes of deformation at the critical load. They have normalization process to ensure that the maximum displacement component is set at 1.0 [82].

There are particular engineering assumptions and analysis constraints are valid for the linear buckling analysis:

- ✓ Deflections should remain within the small deformation regime.
- ✓ No any imperfections or nonlinearities in terms of geometry and material.
- ✓ Element stresses are assumed to remain in the elastic range.
- ✓ The distribution of internal element forces resulting from the applied loads remains constant.

4.2.1 Extraction Methods for Eigenvalues

The eigenvalue analysis solver offers a selection of seven solution methods for calculating eigenvalues. These solution methods; givens, modified givens, householder, modified householder, Lanczos, inversed power, and enhanced inversed power. However, the last three solution methods are frequently employed for the linear eigenvalue buckling analysis [65].

4.2.1.1 The Lanczos Method

The Lanczos method is recommended as the default solution approach for linear buckling analysis [65]. This method surpasses many limitations and incorporates the advantageous aspects of other methods. It efficiently extracts eigenvalues, and if an eigenvalue cannot be obtained within the specified range, a diagnostic message is generated. The Lanczos method ensures precise computation of eigenvalues and eigenvectors. The Lanczos method tends to be more efficient when numerous eigenmodes are needed for a system with a high degree of freedom. The Lanczos method's abbreviation is "EIGRL".

4.2.1.2 The Inversed Power Method

The inverse power method is categorized as a tracking method because it gives the first lowest eigenvalue and eigenvector within the desired range [65]. This method proceeds to identify the next higher mode and sweeps out its effects, so this keeps going successive. It means that each eigenvalue is obtained using an iterative solution process. However, it is important to note that depending on the chosen range of eigenvalues, the inverse power method can potentially miss modes during the solution process, turning the solution method into untrustworthy. As a result, the inverse method is not recommended for linear eigenvalue buckling analysis. The inversed power method's abbreviation is "INV".

4.2.1.3 The Enhanced Inversed Power Method

The enhanced inversed power method is akin to the inverse power method, but it adds the Sturm sequence logic to guarantee that all modes are gained within the specified

eigenvalue range [65]. By employing the Sturm sequence logic, the method obtains the information on the number of buckling modes in accordance with each trial eigenvalue. The enhanced inversed power method's abbreviation is "SINV".

4.2.1.4 Summary of the Extraction Methods of Eigenvalues

In general, the optimal solution method for a specific linear buckling model relies on three parameters [65]:

- ❖ The number of eigenvalues and their ranges.
- ❖ The size of the model.
- ❖ Available memory of the computer.

The Lanczos method is a versatile technique which is applicable to analysis problems of various scales, from small to large [65]. Its efficiency is augmented by using the minority of input matrices so, this makes it especially well-suited for medium to large-scale issues. The Lanczos method is very beneficial for reducing computational speed and disk space usage. As a result, due to its efficiency and robustness, the Lanczos method stands out as the preferred solution method for linear buckling analysis.

The enhanced inversed power method might be an optimal solution method when dealing with large models that exceed the available memory on the computer [65]. It is particularly useful when only a few modes are required for the analysis. The enhanced inversed power approach serves as a valuable backup method for verifying result accuracy when employing alternative solution methods.

4.3 Finite Element Analysis on Buckling

This section presents the numerical analysis road map for the metallic skin-stinger and honeycomb sandwich panels that are studied on this thesis. The panel geometries, material definitions are defined in previous chapter and this section contains element type and mesh properties, boundary and load conditions. In order to implement the finite element analysis steps, the Hypermesh software that covers each progress is used as pre-processor. The numerical modelling process initiates by generating the

geometry, followed by meshing and assigning material and element properties to the panels. The modelling process concludes with the specification boundary and load conditions.

4.3.1 Model Description and Mesh

In this study, the shell elements with four nodes are used for modelling the metallic panels. Thin panels no matter they are flat or curved are modelled as a shell structure in the finite element analysis since the shell models offer accurate responses for thin-walled structures without calculating the stresses along the thickness of the plate [80].

The four-node shell element is widely regarded as the most effective choice for analysing the thin structures due to the fact that this element remains free from clashing or locking issues and exhibits a high predictive capability and it also provides less computational time [81].

The shell element is considered with a 3D (three dimensional) continuum, based on the following assumptions [81]:

- Material particles initially aligned along a straight line perpendicular to the mid-surface of the structure maintain that alignment throughout the deformation.
- The stress in the direction perpendicular to the mid-surface is assumed to be zero.

Figure 4.1 demonstrates the shell elements with different nodes.

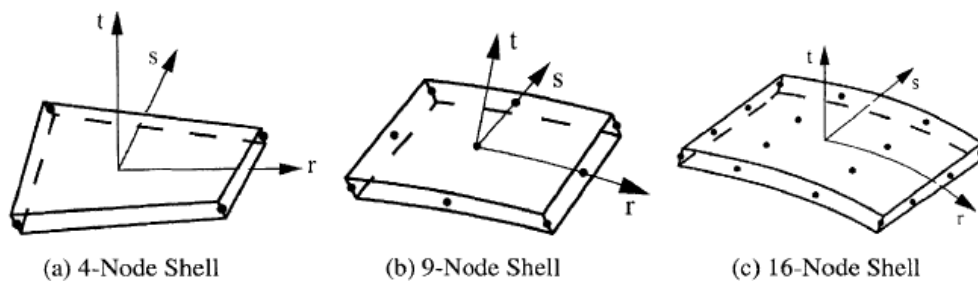


Figure 4.1 Shell elements with different nodes [81]

Mesh size for each part on the panels stays the same and it will be given in the following titles.

4.3.1.1 Metallic Skin-Stringer Panel Model Description and Mesh

In the Hypermesh, during the modelling phase, mid-surface option is used to create the surface of each component so it can be given two-direction thickness to represent the real model properly.

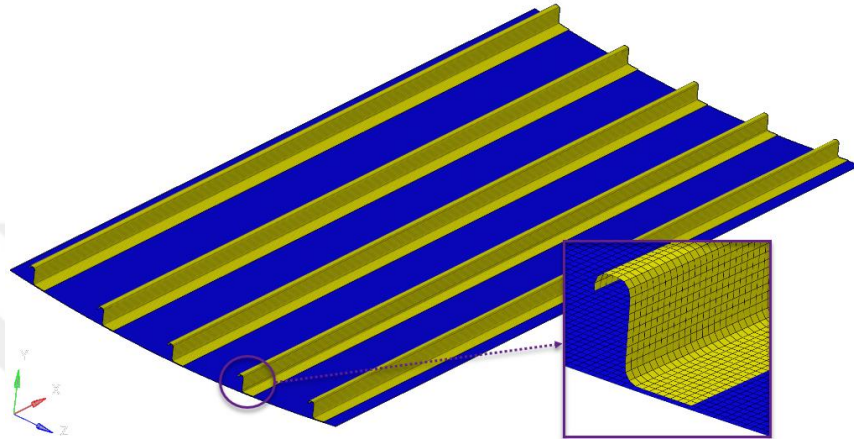
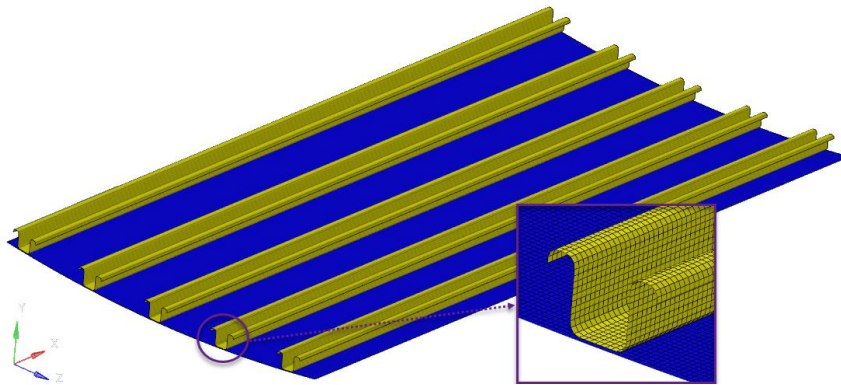
CQUAD4 element type, quadrilateral element, is used in order to model the skin-stringer metallic panels and it uses six degrees of freedom per each node. As a property card, PSHELL element property is employed so that it defines shell properties. In other words, the shell elements are employed for the analysis because of the fact that each component of the panels has relatively smaller thickness compared to the other two dimensions. When PSHELL is used, it is significant to underline that the computed stress outcomes are expressed in the element coordinate system [83]. Furthermore, alternative element types, such as solid elements, might exhibit a more rigid bending behavior, potentially deviating from the analytical solution. The main distinction between a shell element and a solid element lies in the node distribution [84]. In other words, a solid element has nodes across all its surfaces and the nodes are located on the corners of the faces, whereas a shell element only has the nodes on its mid-surface.

Since the metallic skin and all stringers have isotropic properties, MAT1 card is used to define mechanical properties. This card provides material specifications for linear, temperature-independent, isotropic materials [85].

The models are created with very fine mesh, approximately 2x2 mm element size. Therefore, such a dense mesh configuration results in pretty much precise forecast of displacements, stresses, and linear buckling behaviour in reasonable processing time for this study. Quantities of the elements for each component on the related panels can be seen in Table 4.1 and Figure 4.2, Figure 4.3, and Figure 4.4 demonstrate the metallic z cross section stringer-skin, metallic omega cross section stringer-skin, and metallic c section cross stringer-skin assembly mesh view, respectively.

Table 4.1 Number of elements for each skin-stringer panel

Panel Type	Number of Elements
Z Section Stringer-Skin Assembly	228250
Omega Section Stringer-Skin Assembly	277150
C Section Stringer-Skin Assembly	247500

**Figure 4.2** Z cross section stringer-skin assembly mesh view**Figure 4.3** Omega cross section stringer-skin assembly mesh view

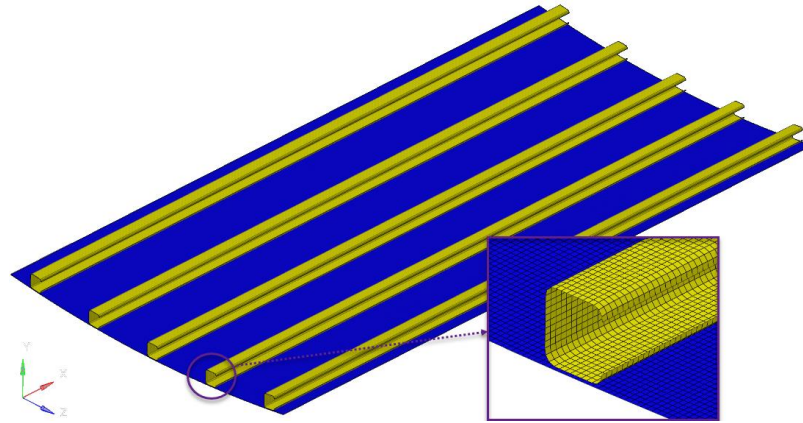


Figure 4.4 C cross section stringer-skin assembly mesh view

TIE contact property is used to define the interface between the lower surface of the stringers and the upper surface of the skin. This definition is surface to surface discretization. While the lower surface of the stringers is slave contact surface, the upper skin surface is selected as a main contact surface since the master surface should be smoother geometrically. If the materials and thicknesses of the components were different, the strongest material in terms of the mechanical properties would have master surface. The corresponding nodes for the contact areas between the skin and stringer look at each other so that they give a reasonable contact and results. According to the TIE contact property, the relative displacements between nodes is zero so each master and slave nodes move together as a group [85]. In other words, TIE contact property will not give a sliding movement between the skin and stringer. TIE contact is based on MPC (multi point constraints) which assign a tied contact between two surfaces.

In this thesis, TIE contact property is utilized because of the fact that the critical buckling load that is extracted from linear eigenvalue buckling analysis is investigated and TIE contact property method is more useful for this case. A TIE connect property can be seen in Figure 4.5 for z cross section stringer-skin metallic panel and the same situation is valid for other panels as well.

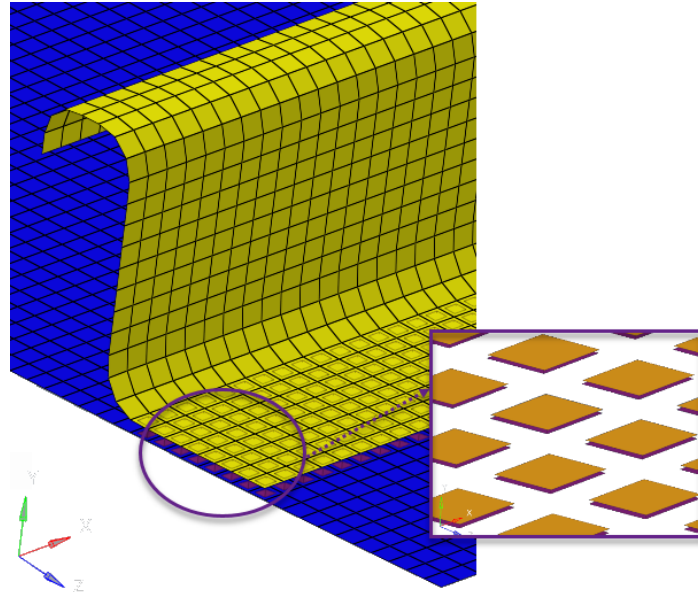


Figure 4.5 TIE contact property between Z cross section stringer and skin

4.3.1.2 Metallic Honeycomb Sandwich Panel Model Description and Mesh

In the Hypermesh, during the modelling phase, the aerodynamic surface of the panel is used to create the elements of the surface for each component. Unlike the metallic skin-stringer panel, the thickness of the metallic honeycomb sandwich panel components is given one way direction to represent the real model properly and so the metallic panel is configured from outer surface to inner surface.

There are three different types of modelling methods in order to create the sandwich panel numerically [36]. The chosen type is affected by the panel materials and geometry, instability mode. The first modelling technique uses the conventional shell elements. These models which are known as layered shell models may offer the overall behavior of the buckling phenomena, especially for thin sandwich panels and it has simplicity and practical [36]. The second method employs shell elements for the face skins or surfaces and solid three-dimensional elements for the core. These models which are known as layered shell/solid models may give the first approach for local and global behavior of the buckling phenomena. The precision of this technique is tied to the thickness modelling of the core material and it is costly compared to the first approach [36]. The third method involves a three-dimensional finite element model,

employing solid three-dimensional elements to represent both the face skins or surfaces and the core. These models which are known as three-dimensional solid models and their usage is reserved for the detailed local modelling due to the associated computational cost and it is the most laborious [36]. In this study, the layered shell models are used to model the metallic honeycomb sandwich panels and the other methods are the out of scope.

According to the layered shell element model method, the existent outer skin is used to model each layer by using two-dimensional shell element with at least three groups of layers [36]. The first set of layers is related to the laminate of one face sheet. The subsequent set of layers corresponds to the honeycomb core, while the final set of layers signifies the laminate of the other face sheet. The result of this approach defines an equivalent single-layer modelling which classical lamination theory is employed to calculate the stiffness coefficients of the honeycomb sandwich and discrete layer modelling [36]. For equivalent single-layer modelling, the laminate is created as a homogenous single layer structure with mechanical properties derived from the macro-mechanical characteristics of each layer within the lamina. In this representation, each layer shares a consistent and peculiar rotation across the cross-section of the honeycomb sandwich [36]. The layered shell models often employ shear-flexible shell elements since the core material usually contributes primarily to shear stiffness. Shell elements which ground on the classical Kirchhoff-Love theory can be utilized. Nevertheless, this approach overlooks the transverse shear flexibility in the honeycomb sandwich structure [36]. In order to reflect transverse shear flexibility, the discrete layer modelling which gives multi layered structure simulation can be used by employing specific modelling compliance and adjustments for the honeycomb sandwich panels [86].

Within sandwich panels, the core has a lower in-plane performance and it triggers to a lesser transmission of load compared to the face sheets [86]. Nonetheless, it is essential to take into account the shear stiffness and strength to shear deformation of the core for modelling sandwich panels, as they significantly influence the overall performance of the panel [86]. In this study, discrete layer modelling is used to reflect the real material properties for each component of the metallic honeycomb sandwich panels.

CQUAD4 element type, quadrilateral element, is used in order to model the metallic honeycomb sandwich panels and it uses six degrees of freedom per each node. As a property card, PCOMP element property is employed so that it defines shell properties and layered shell models. It also means that the shell elements are employed for the analysis because of the fact that each component of the panels has relatively smaller thickness compared to the other two dimensions. When it comes to the PCOMP element property, the general structure and specifications of an n-ply composite laminate material are defined by using PCOMP [85]. The properties of the outer skin, inner skin, honeycomb core, and doublers are embedded in PCOMP element property. Unlike the PSHELL element property, PCOMP contains out-of-plane shear stress as part of its output [84]. Since the epoxy glass which provides protection against corrosion for the panel does not carry any loads, it is not modelled in the Hypermesh. The bonding quality of the interfaces between each component are excellent by using PCOMP element property so each component is linked by coincident nodes [85].

Since the metallic skins and the doublers have isotropic properties, MAT1 card is used to define mechanical properties. On the other hand, MAT8 card which supplies material specifications for linear, temperature-independent, orthotropic materials for two-dimensional elements is employed [85]. While the honeycomb core is modelled, the longitudinal and transverse shear modulus play an important role to create the real properties since the core generally is undergone shear load distribution [83].

Element and PCOMP offsets are used since composite parts or the parts which behave like a composite may have different thickness distribution and variation and so it is critical to fulfil the tool surface of the related composites and their normal directions [87]. Therefore, the elements faces and offsets are taken into account for specifying the tool face as well as material direction. When the offset options are employed, RBE2 (rigid body elements) which connects first nodal plane and grid points to second nodal plane and grid points is used [87]. The relation between first and second point is established by using the MPC that is part of the RBE2.

Like the metallic skin-stringer panels, the models of the metallic honeycomb sandwich panels are created with very fine mesh, approximately 2x2 mm element size in order to get exact forecast of displacements, stresses, and linear buckling behaviour in

reasonable processing time for this study. Quantities of the elements for each panel can be seen in Table 4.2 and since each panel is modelled on the outer skin, all panels have the same element number.

Table 4.2 Number of elements for each honeycomb sandwich panel

Panel Type	Number of Elements
Honeycomb Sandwich Panel Assembly	155100
Honeycomb Sandwich Panel Assembly with Longitudinal Doublers	155100
Honeycomb Sandwich Panel Assembly with Transverse Doublers	155100

Figure 4.6, Figure 4.7, and Figure 4.8 demonstrate the metallic honeycomb sandwich panel without doublers, metallic honeycomb sandwich panel with longitudinal doublers, and metallic honeycomb sandwich panel with transverse doublers mesh view, respectively.

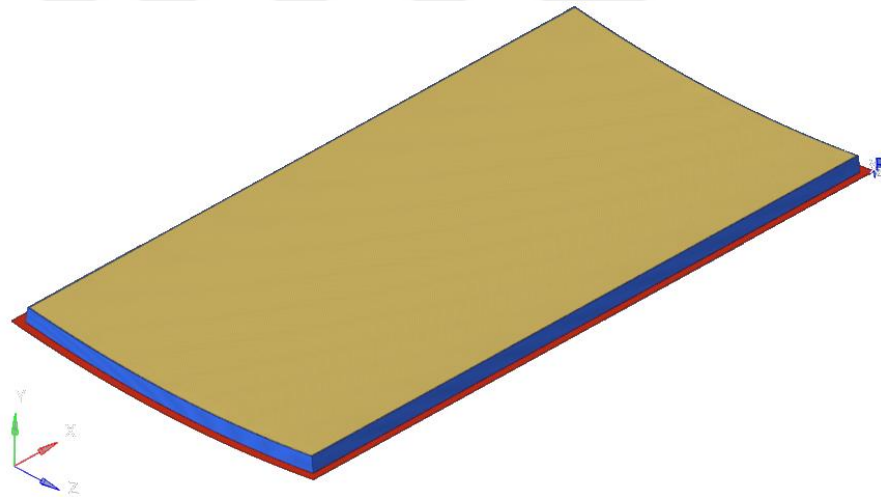


Figure 4.6 Metallic honeycomb sandwich panel without doublers

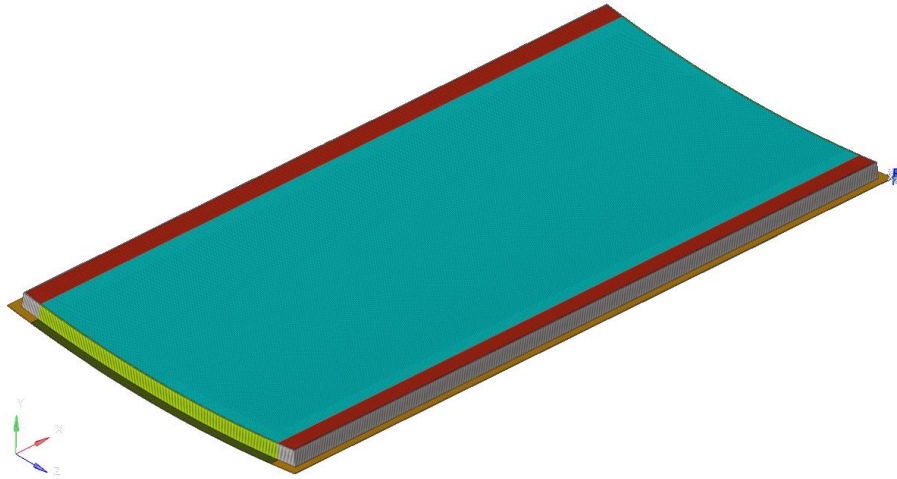


Figure 4.7 Metallic honeycomb sandwich panel with longitudinal doublers

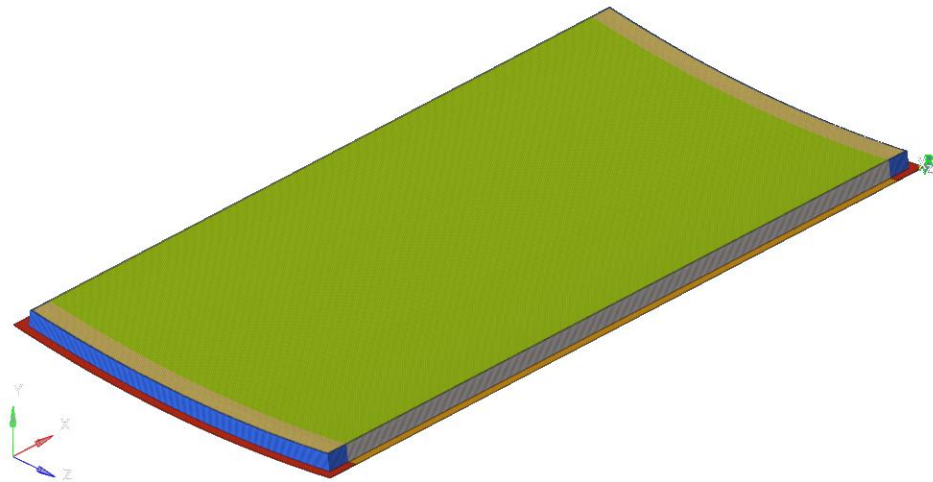


Figure 4.8 Metallic honeycomb sandwich panel with transverse doublers

4.3.2 Boundary Conditions and Loads

Both metallic skin-stringer panels and metallic honeycomb sandwich panel are very common structure in different sections on aircraft or rotorcraft structures. The analyses conducted in this study take into account the status and characteristics of the panels situated on the side of a helicopter fuselage. Since boundary conditions typically work to describe constraints that mirror the real-world limitations of the structure and its mating parts, they are crucial.

Boundary conditions preserve the rotations and displacements which are enforced by other structures theoretically [28]. In other words, they do not have exactly the dynamics of the real-world limitations. Nonetheless, boundary conditions are the representatives for showing a proximity of what occurs. According to the boundary conditions, the accuracy of the results can change and so it is significant to define suitable boundary conditions to see final results properly.

In this study, simply supported boundary conditions are used on the long and short edges of the panels which have the rectangular shape with curvatures for simplification. All skin and stringer nodes on the short edges are taken into account for skin-stinger panel because they have the same final end. On the other hand, only skin nodes on the short edges are considered for sandwich panel structure. These are related to the geometry layout. In reality and some cases for these panel structures, all edges can be considered without any rotations as well.

The boundary conditions that are applied on the panel structures is shown in Table 4.3 and Figure 4.9. While “ U ” signs to the translational movement, “ R_{ot} ” shows the rotational movement and the numbers 1, 2, and 3 which are used with translational and rotational movement symbols indicate X, Y, and Z axis, respectively.

Table 4.3 Boundary Conditions

Boundary Code	Suppressed DOF's
1	U_2 and Rot_3
2	U_2 and Rot_1
Middle Line	U_1 and U_3 (except for end nodes)

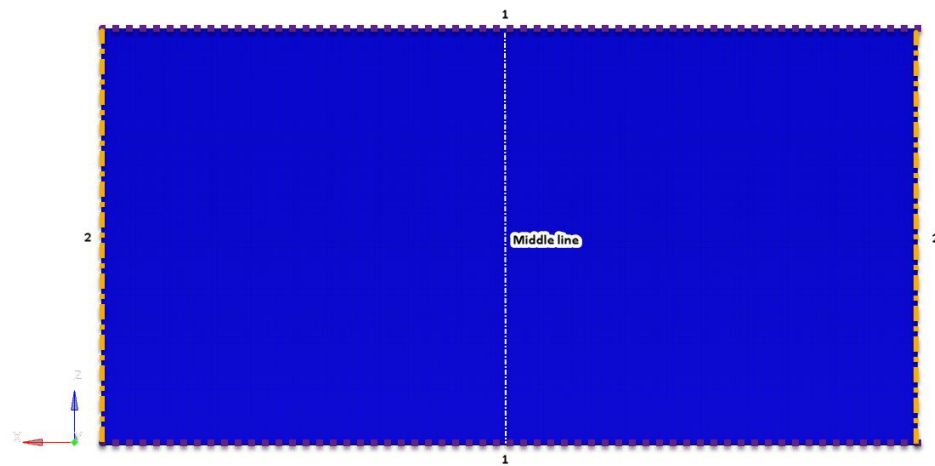


Figure 4.9 Boundary conditions

Displacements on the short and long edges are only free in the X and Z directions. Rotations on the short and long edges are constrained specifically in the X and Z directions, respectively and the displacements on the middle line are only free in the y directions but in the z directions for only ends. The boundary conditions, geometrical, and material properties will remain constant throughout the entire study.

Buckling is a failure phenomenon which is triggered by one or different combinations of loads that are acted on a structure, as elaborated in earlier chapters. Compression loading is the primary reason and it can lead to buckling, potentially resulting in the collapse of a structure.

In this thesis, only longitudinal uniaxial compression loading which is perpendicular to the short edges is taken into account. This loading is applied on the both short edges. Like boundary conditions, while the forces that are applied on the model have an effect on the stringer and skin nodes for skin-stringer panels, only skin nodes and elements are related to the forces for the honeycomb sandwich panel. All applied loads are considered to be applied gradually enough that inertia effects can be disregarded. In addition, pressure, temperature, and humidity effects are omitted for this study.

Compression force is applied on the panels is 1 N in the $\pm X$ direction, as demonstrated typically in Figure 4.10. This force is separated to all related nodes equally so that all

panels undergo to the same loading condition. Otherwise, the whole system can fall down as local at the weakest region of the structure.



Figure 4.10 Applied loads on panel structure

CHAPTER 5

RESULTS AND DISCUSSION

5.1 General Approach to Linear Eigenvalue Buckling Analysis

In this study, as mentioned before, both the base situation and the buckling perturbation process are considered to exhibit linear responses. The calculations for the critical buckling loads of the panels involve utilizing the linear buckling solution method to derive estimates and the analysis does not consider nonlinearities arising from material plasticity and geometric imperfections.

The eigenvalues are obtained as a multiplier of the input perturbation loading. Throughout the linear eigenvalue buckling analysis, the short edge compression loads are regarded as the “*live loads*” [82].

To validate the stress distribution across the panels, an initial step for the linear buckling analysis involves subjecting panels to a linear static analysis within a single load step and the stiffness matrix is obtained [82]. Examining the stress distribution during the linear static analysis allows to confirm that the behaviours of the panels align with expectations and that the boundary conditions work on the panel performance well.

After the linear static analysis and its verification, the linear buckling analysis phase is performed to find the global buckling mode and extract the eigenvalues so that the critical buckling load is calculated. The amount of the mode numbers or eigenvalues can be received as desired but the first eigenvalues have the meaning for the global buckling and the critical buckling load [82]. In addition, after all eigenvalues are obtained; they can be used for non-linear buckling and post buckling analysis in order to calculate the ultimate buckling loads.

5.2 Buckling Analysis of Metallic Flat and Curved Skin Panels

5.2.1 Buckling Analysis of Metallic Flat Skin Panel

Linear eigenvalue buckling analysis is carried out on the flat skin panel in order to demonstrate complying with the analytical solution that is calculated via Eq. 2.13. This analysis result creates a basis of the other analysis. This flat skin panel has the same dimensions in terms of thickness, width and length like the curved skin panel that is unique panel for this study.

All analysis steps are carried out as described in Chapter 5.1 and the stress and displacement distribution for the flat skin panel is demonstrated in Figure 5.1. The stress and displacement distribution are found to be suitable. Higher displacements predominantly occur on the short edges of the panel and their further, like estimated. In other words, the middle of the panel has the minimum displacements. The stresses and displacements distribute equal along the panel.

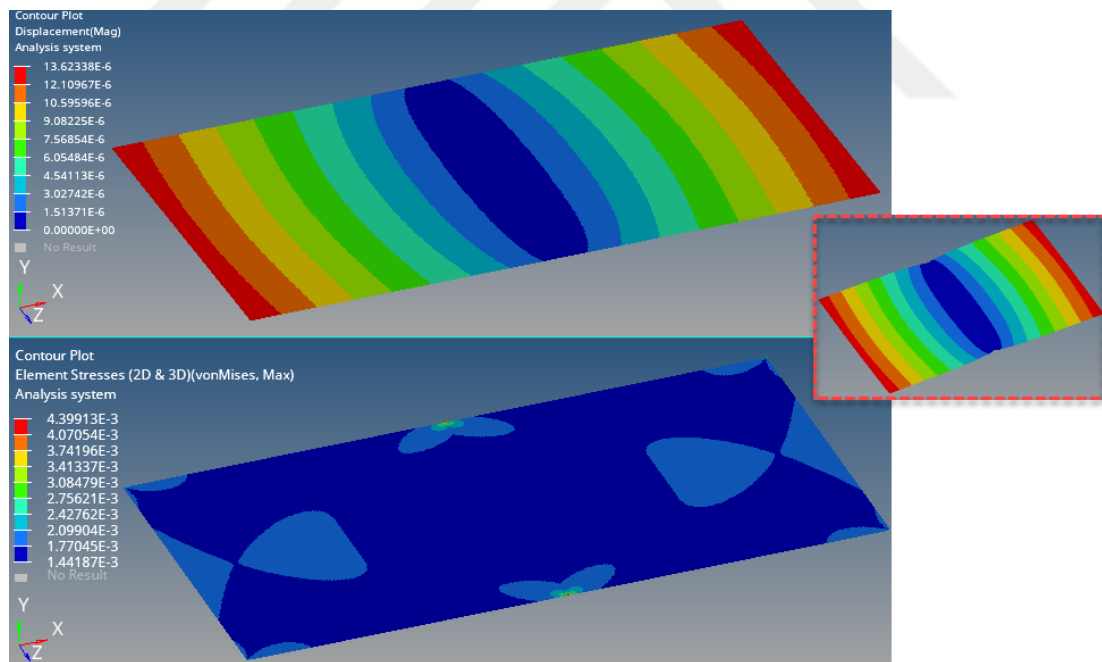


Figure 5.1 Linear static analysis results for flat skin panel

Figure 5.2 illustrates the acquisition of eigenmode configurations corresponding to various equilibrium states. In this study, four different buckling modes are calculated

for this panel configuration. Because after the first four modes, the buckling modes get meaningless shapes. It is important to highlight that the critical load of the panel aligns specifically with the first buckled form featuring four mode or waves.

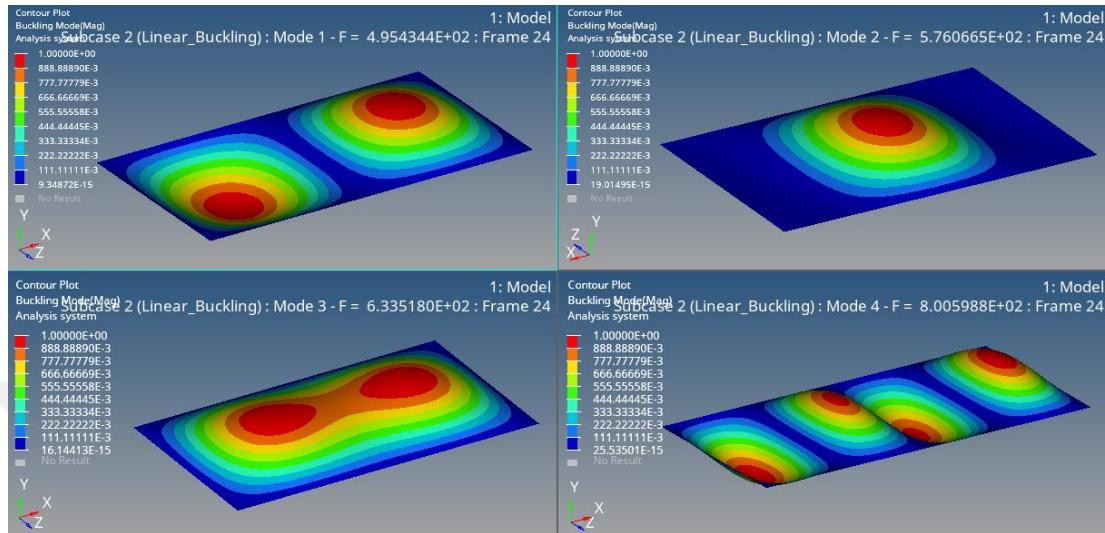


Figure 5.2 Eigenvalues of flat skin panel

According to the linear eigenvalue buckling analysis result, the value of the first extracted eigenvalue is 495 and since the applied load is 1 N, the critical buckling load that indicates the global buckling behaviour is 495 N. On the other hand, analytical solution, the corresponding answer of the numerical solution, is 492.5 N by using Eq. 2.13. The numerical result has a good correlation with analytical result and the deviation is under 1%. This is desired result in terms of accuracy [65]. Under the uniaxial loading condition, the first mode shape (m) reveals the formation of two half-waves along the longitudinal direction, contrasting with one half-wave (n) formed in the transversal direction. This result conforms to the graphic that is shown in Figure 2.15. Table 5. 1 shows the critical buckling loads for analytical and numerical solution on the simply supported metallic flat skin.

Table 5. 1 Comparison between analytical and numerical solution for metallic flat panel

Solution Type	Analytical Solution	Numerical Solution	Difference
Critical Global Buckling Load [N]	492.5	495	0.507%

5.2.1 Buckling Analysis of Metallic Curved Skin Panel

The unique metallic skin panel for this study undergoes a linear eigenvalue buckling analysis for comparison with the previously calculated metallic flat skin panel. Initially, a linear static analysis is conducted, followed by the performance of linear eigenvalue buckling analysis to determine the critical buckling load for the curved metallic panel.

Figure 5.3 illustrates the stress and displacement distribution of the metallic curved skin panel. The observed distribution proves to be satisfactory, with relatively higher displacements primarily concentrated at the panel's center, aligning with the anticipated pattern. The stresses and displacements are evenly distributed across the entire panel.

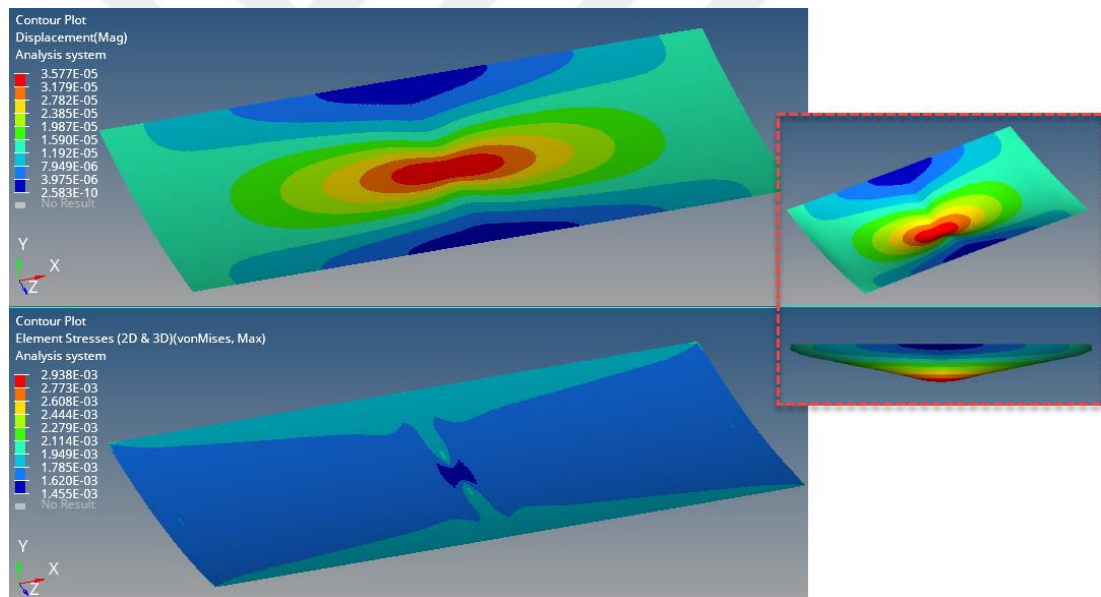


Figure 5.3 Linear static analysis results for curved skin panel

Figure 5.4 depicts the capture of eigenmode configurations corresponding to different equilibrium states. This study encompasses the calculation of four distinct buckling modes for this panel configuration. Because after the first four modes, the buckling modes get meaningless shapes. It is important to highlight that the critical load of the panel aligns specifically with the first buckled form featuring four mode or waves, like for the flat panel.

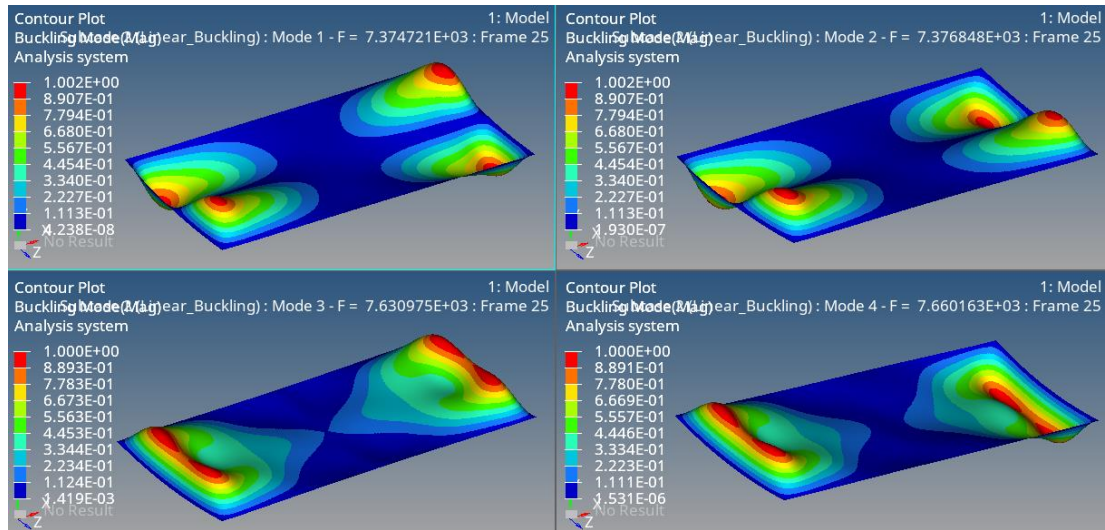


Figure 5.4 Eigenvalues of curved skin panel

As per the outcome of the linear eigenvalue buckling analysis, the initial extracted eigenvalue is 7374. Considering the applied load of 1 N, the critical buckling load signifying global buckling behavior is determined as 7374 N. However, the numerical solution for the metallic flat skin panel is 495 N. Since the curved panels are designed for safety, they are more conservative to flat panels. In other words, they have a bigger load capacity under compression. This means that a curved panel which has greater buckling strength can be considered as an identical flat panel with increased plate thickness or reduced slenderness [87]. The curvature of the panel has an effectiveness on the buckling factor. According to the analysis result, the critical buckling performance of the curved skin panel is bigger than the flat panel and the results can be seen in Table 5. 2. The buckling strength of the curved plate is significantly influenced by its slenderness ratio and it is valid for a flat plate as well.

Table 5. 2 Comparison critical buckling loads between flat skin panel and curved skin panel

Panel Type	Metallic Flat Skin Panel	Metallic Curved Skin Panel	Difference
Critical Global Buckling Load [N]	495	7374	1389.6%

5.3 Buckling Analysis of Metallic Skin-Stringer Panels

5.3.1 Buckling Analysis of Metallic Z Cross Section Stringer-Skin Panel

Linear eigenvalue buckling analysis is carried out on the z cross section stringer-skin panel in order to calculate the critical buckling load. The properties of the z cross section stringer are described in Chapter 3.

In Figure 5.5, the stress and displacement distribution of this panel is illustrated. The observed distribution proves satisfactory and the higher displacements predominantly occur at the center of the panel and on the middle stringer, like estimated. As for an overall structural integrity, the middle stringer plays an important role. In this area, there is a super-stiffener concept around the middle stringer due to the fact that load density. Since the compression forces are applied on the stringer nodes as well as the skin nodes, these regions demonstrate the relatively high stress distribution on the stringer compared to other regions. The stresses and displacements distribute equal along the panel Z axis.

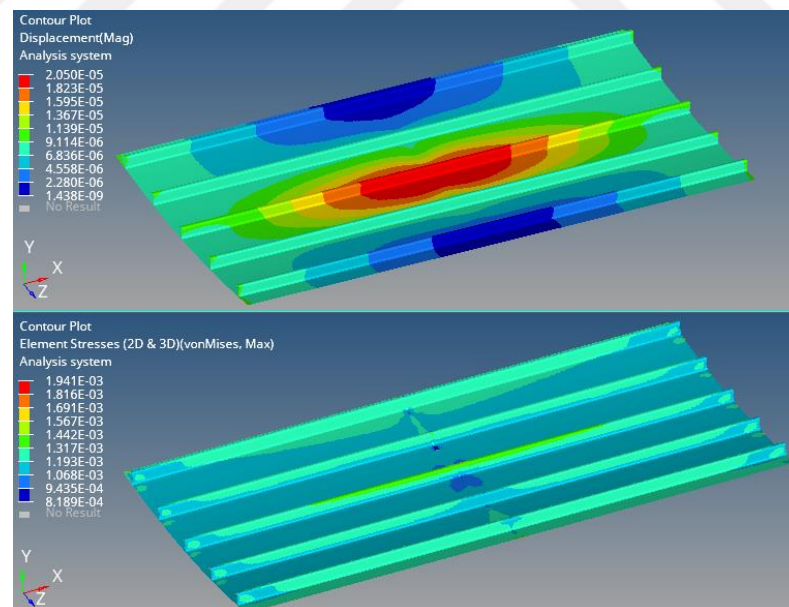


Figure 5.5 Linear static analysis results for z cross section stringer-skin panel

Figure 5.6 depicts the capture of eigenmode configurations corresponding to different equilibrium states. This study encompasses the calculation of five distinct buckling

modes for this panel configuration. Because after the first five modes, the buckling modes get meaningless shapes. It is important to highlight that the critical load of the panel aligns specifically with the first buckled form featuring five modes.

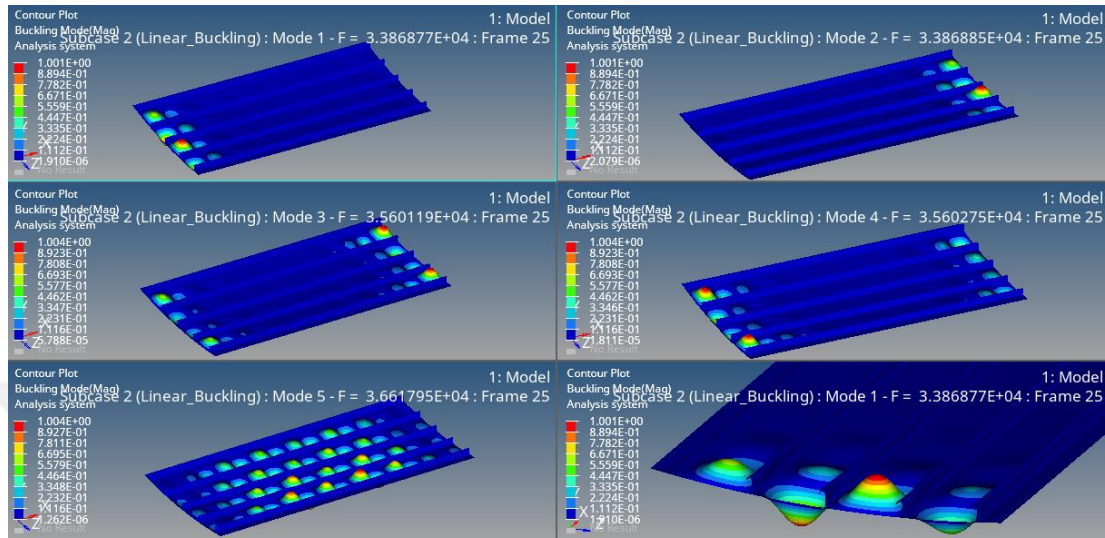


Figure 5.6 Eigenvalues of z cross section stringer-skin panel

As per the outcome of the linear eigenvalue buckling analysis, the initial extracted eigenvalue is 33868. Considering the applied load of 1 N, the critical buckling load signifying global buckling behavior is determined as 33868 N.

According to the St. Venant's principle, the stresses on a boundary that are quite distant from an applied load, remain relatively unchanged when the load is replaced with a statically equivalent load. The alteration in stress and strain distribution occurs only in the proximity of the load application regions [88]. In a nutshell, analysing the uniformly distributed stress in a suitable location where the loading occurs is considered adequate for the analysis.

St. Venant's principle can be applied on the shell structures that are investigated in this study because of the zero-resultant force and zero moment around the axis.

The first two buckling shapes show the mirror characteristic to each other, similarly third and fourth buckling shapes. This can occur because of the numerical boundary effect and it can be explained with the St. Venant's principle. On the other hand, the

fifth buckling shape can demonstrate the closest distribution of the global buckling shape. In other words, this shape might be anticipated in the real life due to the distribution of the buckling half waves and the St. Venant's principle supports this.

5.3.2 Buckling Analysis of Metallic Omega Cross Section Stringer-Skin Panel

Linear eigenvalue buckling analysis is carried out on the omega cross section stringer-skin panel in order to calculate the critical buckling load. The properties of the omega cross section stringer are described in Chapter 3.

In Figure 5.7, the stress and displacement distribution of this panel is illustrated. The observed distribution proves satisfactory and the higher displacements predominantly occur at the center of the panel and on the middle stringer, like estimated. As for an overall structural integrity, the middle stringer plays an important role. In this area, there is a super-stiffener concept around the middle stringer due to the fact that load density. Since the compression forces are applied on the stringer nodes as well as the skin nodes, these regions demonstrate the relatively high stress distribution on the stringer compared to other regions. The stresses and displacements distribute equal along the panel Z axis.

Figure 5.8 depicts the capture of eigenmode configurations corresponding to different equilibrium states. This study encompasses the calculation of six distinct buckling modes for this panel configuration. Because after the first six modes, the buckling modes get meaningless shapes. It is important to highlight that the critical load of the panel aligns specifically with the first buckled form featuring six modes. This panel configuration includes six different buckling shapes due to the fact that the panel firstly has started possessing the distribution of the buckling half waves in the middle of the panel with the sixth buckling shape.

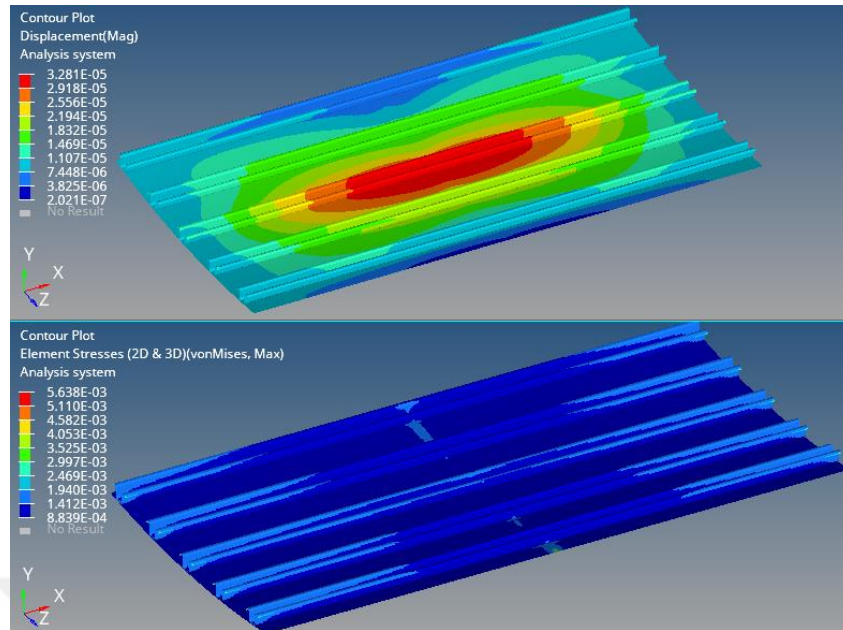


Figure 5.7 Linear static analysis results for omega cross section stringer-skin panel

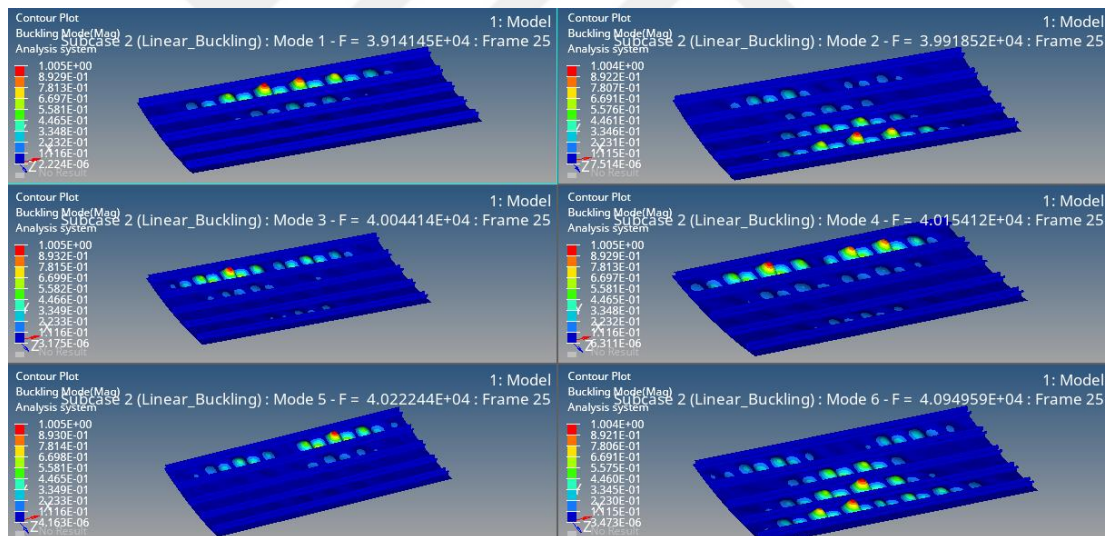


Figure 5.8 Eigenvalues of omega cross section stringer-skin panel

As per the outcome of the linear eigenvalue buckling analysis, the initial extracted eigenvalue is 39141. Considering the applied load of 1 N, the critical buckling load signifying global buckling behavior is determined as 39141 N.

Like the z cross section-stringer panel configuration, St. Venant's principle is also effective on this metallic panel type. The first five buckling shapes show that the half

waves mainly are located on the panel side and this can be considered because of the numerical boundary effect. On the other hand, the sixth buckling shape can demonstrate the closest distribution of the global buckling shape as a first impression. In other words, this shape might be anticipated in the real life due to the distribution of the buckling half waves.

5.3.3 Buckling Analysis of Metallic C Cross Section Stringer-Skin Panel

Linear eigenvalue buckling analysis is carried out on the c cross section stringer-skin panel in order to calculate the critical buckling load. The properties of the c cross section stringer are described in Chapter 3.

In Figure 5.9, the stress and displacement distribution of this panel is illustrated. The observed distribution proves satisfactory and the higher displacements predominantly occur at the center of the panel and on the middle stringer, like estimated. As for an overall structural integrity, the middle stringer plays an important role. In this area, there is a super-stiffener concept around the middle stringer due to the fact that load density. Since the compression forces are applied on the stringer nodes as well as the skin nodes, these regions demonstrate the relatively high stress distribution on the stringer compared to other regions. The stresses and displacements distribute equal along the panel Z axis.

Figure 5.10 depicts the capture of eigenmode configurations corresponding to different equilibrium states. This study encompasses the calculation of four distinct buckling modes for this panel configuration. Because after the first four modes, the buckling modes get meaningless shapes. It is important to highlight that the critical load of the panel aligns specifically with the first buckled form featuring four modes. This panel configuration includes four different buckling shapes due to the fact that the panel firstly has started possessing the distribution of the buckling half waves in the middle of the panel with the third buckling shape.

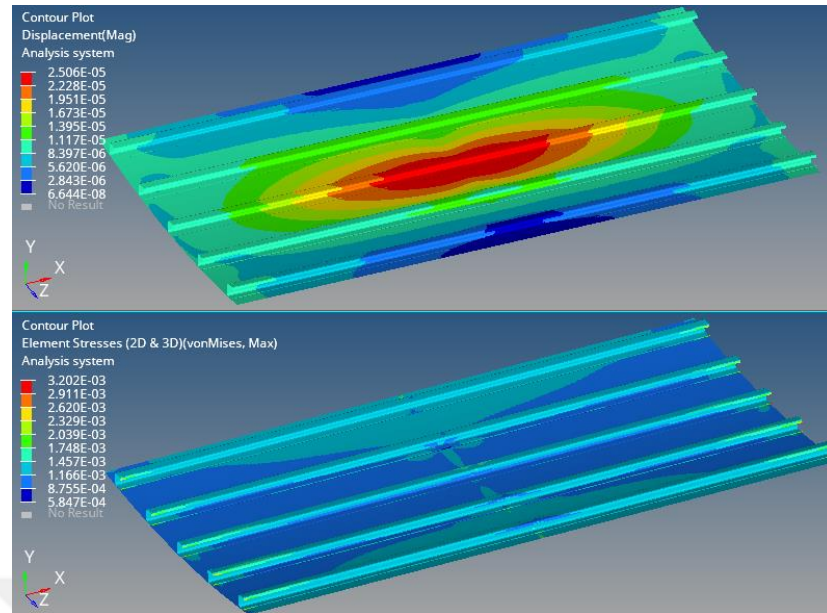


Figure 5.9 Linear static analysis results for c cross section stringer-skin panel

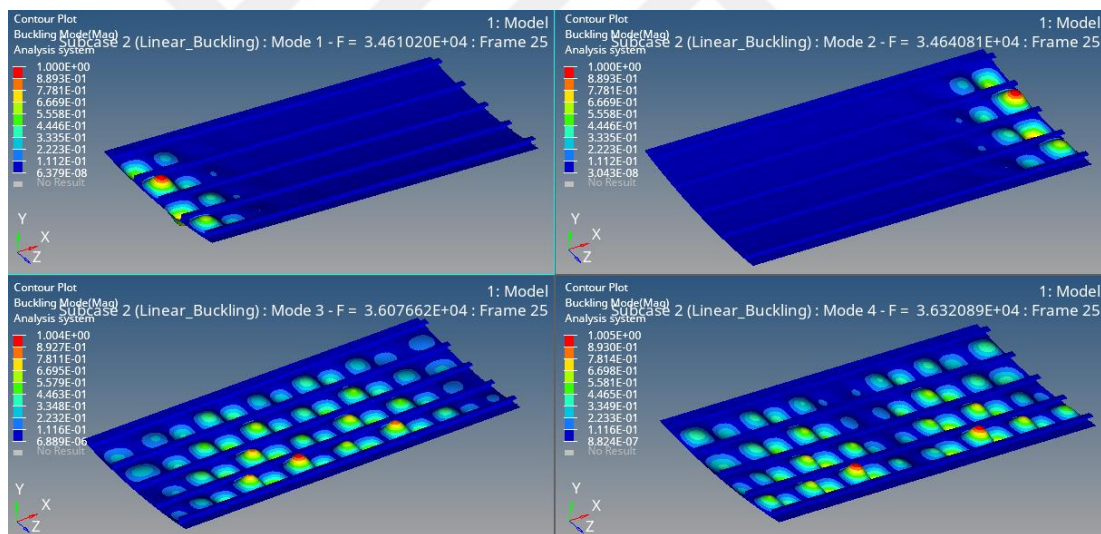


Figure 5.10 Eigenvalues of c cross section stringer-skin panel

As per the outcome of the linear eigenvalue buckling analysis, the initial extracted eigenvalue is 34610. Considering the applied load of 1 N, the critical buckling load signifying global buckling behavior is determined as 34610 N.

Like the other panel configuration, St. Venant's principle is also effective on this metallic panel type. The first two buckling shapes show the mirror characteristic to

each other and this can be considered because of the numerical boundary effect. On the other hand, the third buckling shape can demonstrate the closest distribution of the global buckling shape as a first impression. In other words, this shape might be anticipated in the real life due to the distribution of the buckling half waves.

5.4 Buckling Analysis of Metallic Honeycomb Sandwich Panels

5.4.1 Buckling Analysis of Metallic Honeycomb Sandwich Panel without Doublers

Linear eigenvalue buckling analysis is carried out on the metallic honeycomb sandwich panel that has not any doublers in order to calculate the critical buckling load. The properties of the panel are described in Chapter 3.

In Figure 5.11, the stress and displacement distribution of this panel is illustrated. The observed distribution proves satisfactory under the geometrical and boundary conditions like estimated. Relatively high stress distribution can be seen on the short edges because of the panel geometry. As for an overall structural integrity, the whole panel plays an important role. The stresses and displacements distribute equal along the panel Z axis.

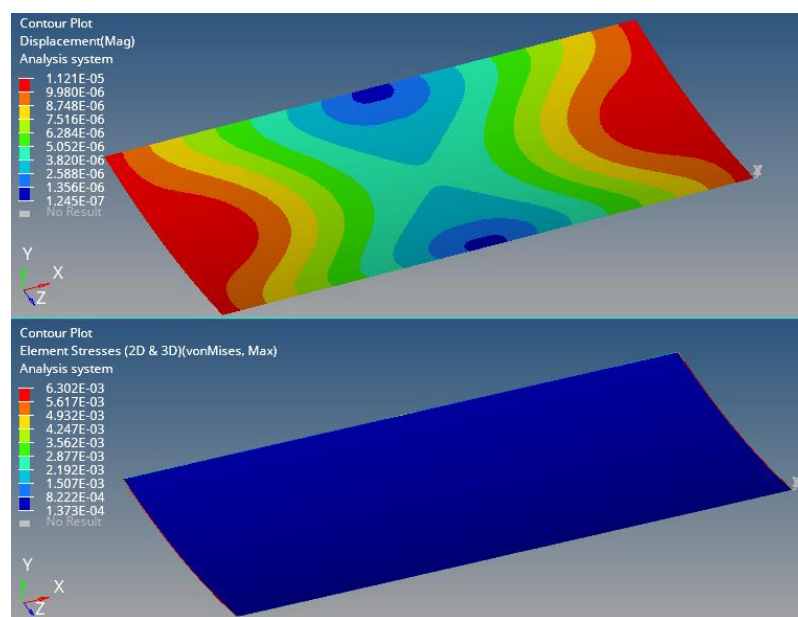


Figure 5.11 Linear static analysis results for honeycomb sandwich panel without doublers

Figure 5.12 depicts the capture of eigenmode configurations corresponding to different equilibrium states. This study encompasses the calculation of six distinct buckling modes for this panel configuration. Because after the first six modes, the buckling modes get meaningless shapes. It is important to highlight that the critical load of the panel aligns specifically with the first buckled form featuring six modes.

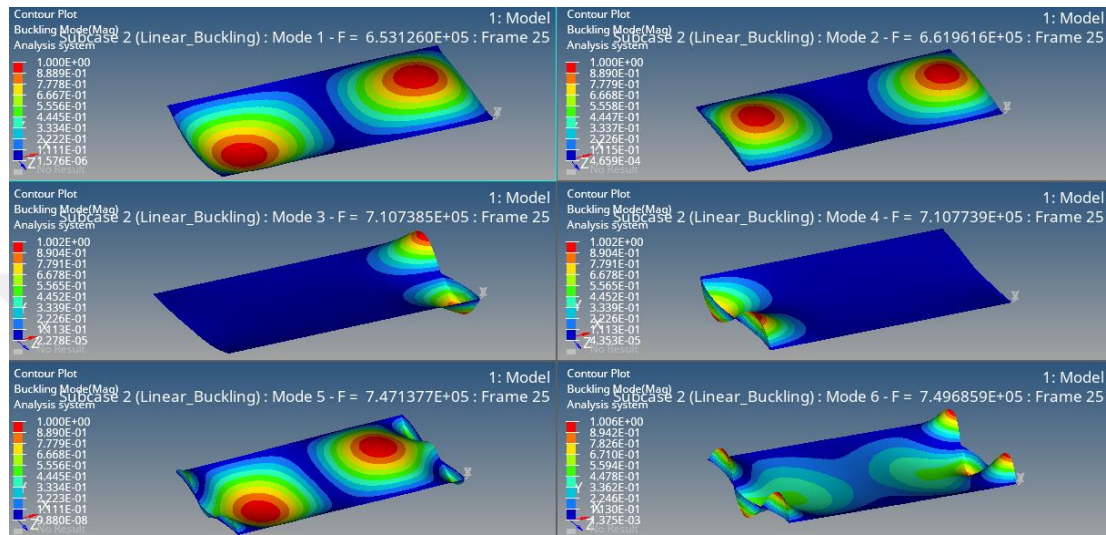


Figure 5.12 Eigenvalues of honeycomb sandwich panel without doublers

As per the outcome of the linear eigenvalue buckling analysis, the initial extracted eigenvalue is 653126. Considering the applied load of 1 N, the critical buckling load signifying global buckling behavior is determined as 653126 N. The first buckling shape can demonstrate the closest distribution of the global buckling shape for real cases.

5.4.2 Buckling Analysis of Metallic Honeycomb Sandwich Panel with Longitudinal Doublers

Linear eigenvalue buckling analysis is carried out on the metallic honeycomb sandwich panel that has longitudinal doublers in order to calculate the critical buckling load. The properties of the panel are described in Chapter 3.

Figure 5.13 demonstrates the stress and displacement distribution of this panel. The distributions of the stress and displacement are sufficient under provided conditions.

Relatively high stresses are distributed on the loaded edges or short edges. Since the longitudinal doublers are used on the panel, the biggest stress is seen on the outer skin between these doublers. The stresses and displacements distribute equal along the panel Z axis.

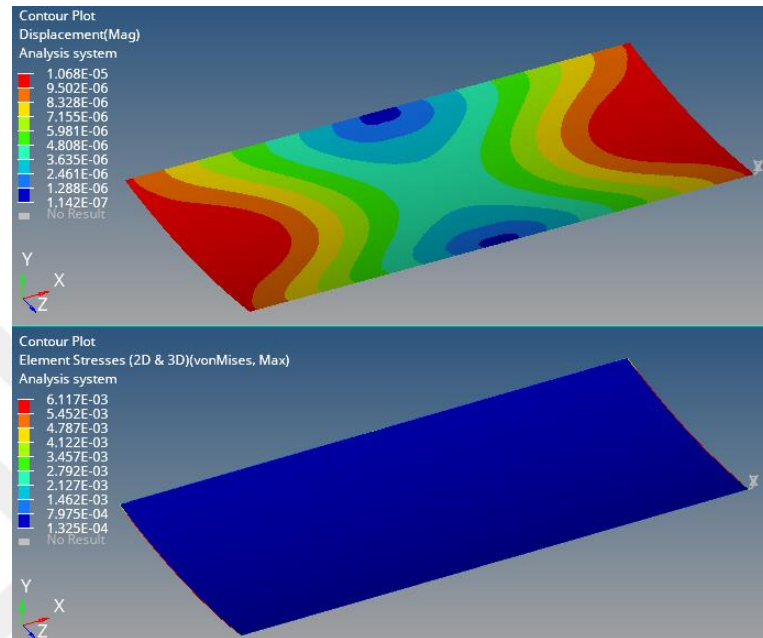


Figure 5.13 Linear static analysis results for honeycomb sandwich panel with longitudinal doublers

In Figure 5.14, the eigenmodes of this configuration can be seen as per the different equilibrium states. This study contains the calculation of four different buckling modes for this panel configuration. Because after the first four modes, the buckling modes get meaningless shapes. It is important to highlight that the critical load of the panel aligns specifically with the first buckled form featuring four modes.

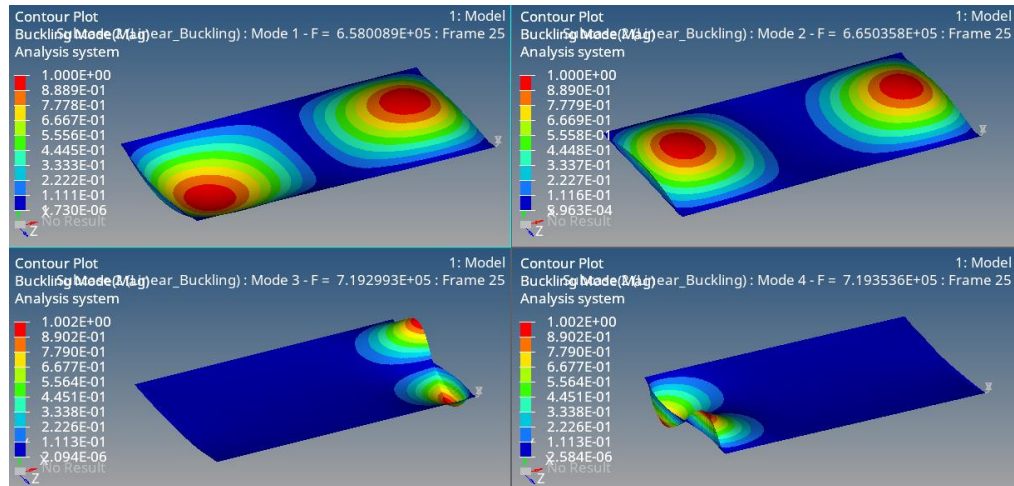


Figure 5.14 Eigenvalues of honeycomb sandwich panel with longitudinal doublers

According to the result of the linear eigenvalue buckling analysis, the first extracted eigenvalue is 658008. Considering the applied load of 1 N, the critical buckling load signifying global buckling behavior is determined as 658008 N. The first buckling shape can demonstrate the closest distribution of the global buckling shape for real cases.

5.4.3 Buckling Analysis of Metallic Honeycomb Sandwich Panel with Transverse Doublers

Linear eigenvalue buckling analysis is carried out on the metallic honeycomb sandwich panel that has the doublers which lie in transverse direction in order to calculate the critical buckling load. The properties of the panel are described in Chapter 3.

Figure 5.15 illustrates the stress and displacement distribution of this panel. The observed distribution proves satisfactory under the geometrical and boundary conditions like guessed. Relatively higher stress distribution can be seen on the short edges because of presence of the transverse doublers. The equivalent thickness of the outer skin on the short edges seems increased thanks to the transverse doublers and this affects the overall structural integrity. The stresses and displacements distribute equal along the panel Z axis.

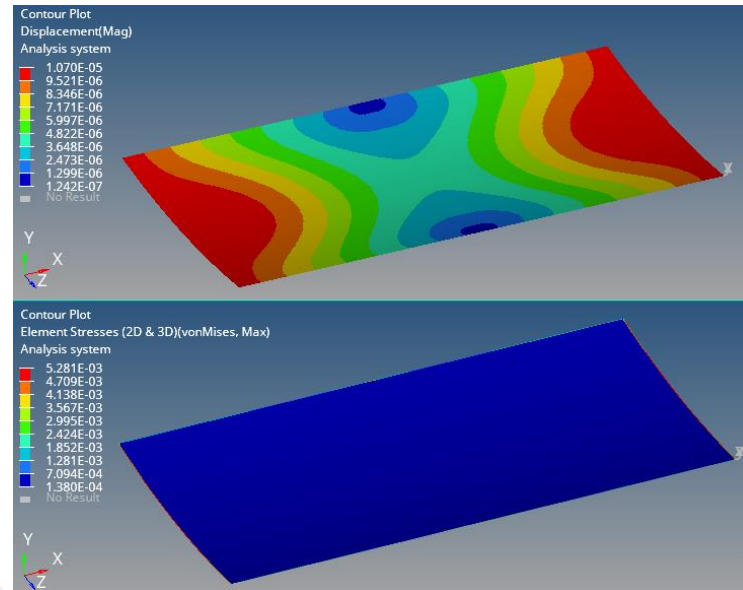


Figure 5.15 Linear static analysis results for honeycomb sandwich panel with transverse doublers

Figure 5.16 depicts eigenmodes of the linear buckling problem corresponding to different equilibrium states. This study includes the calculation of six different buckling modes for this panel configuration. Because after the first six modes, the buckling modes get meaningless shapes. It is important to highlight that the critical load of the panel aligns specifically with the first buckled form featuring six modes.

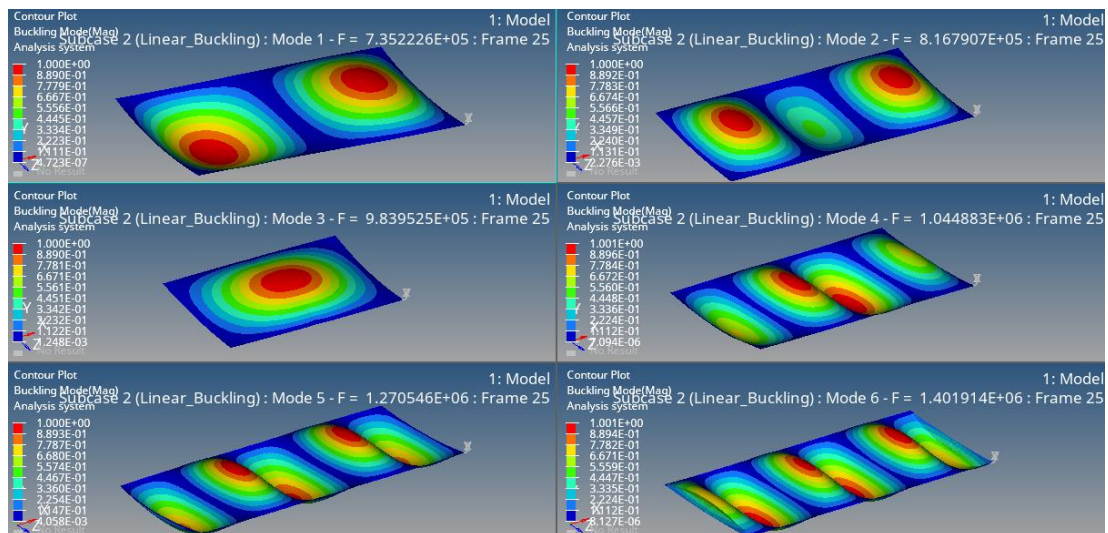


Figure 5.16 Eigenvalues of honeycomb sandwich panel with transverse doublers

As per the result of the linear eigenvalue buckling analysis, the initial received eigenvalue is 735222. Considering the applied load of 1 N, the critical buckling load signifying global buckling behavior is determined as 735222 N. The first buckling shape can demonstrate the closest distribution of the global buckling shape for real cases.



CHAPTER 6

CONCLUSION AND FUTURE WORK

6.1 Comparison of Buckling Behaviour of Metallic Skin-Stringer Panel Structures

Critical global buckling loads have been calculated by using linear buckling eigenvalue method numerically for each metallic stringer variation-skin panel. The summary of the calculations can be seen in Table 6.1.

Table 6.1 Critical global buckling loads for each stringer-skin panel

Panel Type	Critical Global Buckling Load [N]
Metallic Z Cross Section Stringer-Skin Panel	33868
Metallic Omega Cross Section Stringer-Skin Panel	39141
Metallic C Cross Section Stringer-Skin Panel	34610

According to the Table 6.1, even though the critical global buckling loads are similar to each other, the panel which has omega cross section stringer-skin combination is the most durable configuration among other panels, since the omega cross section stringer has one more outer flange compared to the other variations. An omega cross section stringer can be thought as coupling two z design stringers in a back-to-back adjustment. After the omega cross section stringer-skin panel combination, the panel that has c cross section stringers ranks as second in terms of global buckling performance. When it comes to the last stringer type, the panel which possesses z cross section stringers is very close to c cross section stringer-skin panel when the global buckling performance is thought since the inertia which influences the global buckling behaviour of a panel is similar.

In a nutshell with numbers in term of the critical global buckling loads, the panel which has omega cross section stringers shows approximately 13.09% better performance than the panel which has c cross section stringers, while demonstrating about 15.57% better performance than the panel which has z cross section stringers. When it comes to the comparison between the panel which has c cross section stringers and the panel

which has z cross section stringers, c cross section stringer-skin panel combination has better than z cross section stringer-skin panel combination with 2.19%.

According to the fuselage sections of the helicopters and desired requirements, all panel variants can be used to eradicate global buckling behaviour. All stringer types offer notable compression resistance. Indeed, the results of numerical study in this thesis show that.

In addition, manufacturability, assembly requirements, and maintainability issues should be taken into consideration for usage of these stringers. For example, attaching z cross section stringer design through fastening to the skin or its installation is uncomplicated due to the fact that it is easy to reach to the outer flange compared to the other two different types of stringers. On the other hand, the installation process is challenging due to limited accessibility for omega and c cross section stringers. As for manufacturability and cost, it is likelihood that the z cross section stringer is the best when the amount of used material, time to manufacture, and effort are considered.

Each panel configuration has the same weight because of the equivalent of geometrical and material properties. Since Al 2024 T3 is used for each panel, there will not be a difference in terms of corrosion, lightning, and electricity issues etc.

6.2 Comparison of Buckling Behaviour of Metallic Honeycomb Sandwich Panel Structures

Critical global buckling loads have been calculated by using linear buckling eigenvalue method numerically for each metallic honeycomb sandwich panel. The summary of the calculations can be seen in Table 6.2.

Table 6.2 Critical global buckling loads for each metallic honeycomb sandwich panel

Panel Type	Critical Global Buckling Load [N]
Metallic Honeycomb Sandwich Panel without Doublers	653126
Metallic Honeycomb Sandwich Panel with Longitudinal Doublers	658008
Metallic Honeycomb Sandwich Panel without Transverse Doublers	735222

According to the Table 6.2, The honeycomb sandwich panel that has transverse doublers along the short edges is the most resistant to the global buckling behaviour compared to the other panels, since transverse doublers give an additional continuous thickness on loaded edges. In other words, this creates extra strength for the whole panel. Although the thickness of these transverse doublers is very thin, the influence on the overall buckling strength is massive. Secondly, the honeycomb sandwich panel that has longitudinal doublers ranks as second in terms of global buckling performance. However this configuration is the heaviest panel type. Contribution of the different sorts of doubler can be seen on the metallic honeycomb sandwich panel which has not any doublers. Therefore, this panel is end of the list in terms of the global buckling behaviour.

In a nutshell with numbers in term of the critical global buckling loads, the honeycomb sandwich panel that has transverse doublers shows approximately 11.74% better performance than the honeycomb sandwich panel that has longitudinal doublers, while demonstrating about 12.56% better performance than metallic honeycomb sandwich panel which has not any doublers. When it comes to the comparison between the panel which has longitudinal doublers and the panel which has not any doublers, the honeycomb sandwich panel that has longitudinal doublers has better than the counterpart which has not any doublers with 0.75%.

According to the fuselage sections of the helicopters and desired requirements, all panel variants can be used to eradicate global buckling behaviour and they offer remarkable buckling resistance thanks to the excellent flexural strength. Indeed, the results of numerical study in this thesis show that.

In addition, manufacturability, assembly requirements, and maintainability issues should be taken into consideration for usage of these panels. As the number of parts on the honeycomb sandwich panels is increased, the assembly requirements, maintainability and manufacturability occurrences are enlarged. This can give additional cost and effort.

Each panel configuration has different weight because of the presence or absence of the doublers, as described in Chapter 3. The most important idea here is that minimal

weight gain gives stiffness increases more in terms of buckling behaviour. However, when the critical buckling load per each kg, the honeycomb sandwich panel that has longitudinal doublers is end of the list. On the other hand, the honeycomb sandwich panel that has transverse doublers shows perfect ratio for this aspect, as shown in Table 6.3. Moreover, Al 2024 T3 and Al 5052 are used for each panel, there will not be a difference in terms of corrosion, lightning, and electricity issues etc.

6.3 Overall Summary and Future Work

In this study, the different types of metallic skin-stringer and honeycomb sandwich panels which are used on a helicopter fuselage are compared in terms of linear global buckling behaviour under particular circumstances.

Based on the finite element analysis results, metallic honeycomb sandwich panels have global buckling strength more compared to metallic skin-stringer panels. Furthermore, the critical global buckling load magnitudes on metallic honeycomb sandwich panels per each kg is greater than metallic skin-stringer panels. These values can be seen in Table 6.3. In other words, bigger global buckling critical load values with minimum weight increase can be received by using metallic honeycomb sandwich panels. They give superior stiffness to weight ratio and enhance the overall inertia to the structural integrity.

Table 6.3 Critical global buckling loads per kg for each panel

Panel Type	Critical Global Buckling Load per Kg [N]
Metallic Z Cross Section Stringer-Skin Panel	13178.2
Metallic Omega Cross Section Stringer-Skin Panel	15230.1
Metallic C Cross Section Stringer-Skin Panel	13466.9
Metallic Honeycomb Sandwich Panel without Doublers	223673.3
Metallic Honeycomb Sandwich Panel with Longitudinal Doublers	219336.1
Metallic Honeycomb Sandwich Panel without Transverse Doublers	248385.8

In a nutshell with numbers in term of the critical global buckling loads, it is reasonable to compare between the honeycomb sandwich panel that has the best global buckling performance and its skin-stringer panel's counterpart. In addition, giving the

comparision between the honeycomb sandwich panel which does not include any doublers and its skin-stinger panel's counterpart plays an important role due to the fact that they have the least critical global buckling loads among their groups. Lastly, the comparing between the panel which has omega cross section stringers and the honeycomb sandwich panel which has not any doublers keeps significance in order to show how this type of the honeycomb sandwich panel has better global buckling performance compared to the panel which has omega cross section stringers. The honeycomb sandwich panel that has transverse doublers which has the best global buckling performance among honeycomb sandwich panel group shows approximately 1774% better performance than the panel which has omega cross section stringers that has the best global buckling performance among the skin-stringer panel group. The honeycomb sandwich panel which does not contain any doublers which has the worst global buckling performance among honeycomb sandwich panel group demonstrates roughly 1568% better performance than the panel which has omega cross section stringers, and nearly 1827% better performance than the panel which has z cross section stringers that has the worst global buckling performance among the skin-stringer panel group.

In terms of manufacturability, the metallic honeycomb sandwich panels reduce the material consumption and they have fewer structural components such as fasteners etc. since they are bonded together. This also signs reduced weight, cost, and effort. It is easy to assemble them. Nonetheless, the manufacturing and repair processes might be costly. They are very susceptible to water ingress, corrosion, and bonding issues that weaken structural capability. In terms of bonding quality, adhesives may not give the bonded surface quality sufficiently and so this can trigger disconnection problems between the components. These issues should be the main concerns for a panel design when the metallic honeycomb sandwich panel is thought.

When it comes to the metallic skin-stringer panels, they are conventional fuselage shell or wing shell structures. They are widely used in each aircraft/rotorcraft aerostructures. The reasons of this that they have generally costless, familiar, proven and traditional designs compared to the metallic honeycomb sandwich panels.

Consequently, three different kinds of metallic skin-stringer panels and three different types of metallic sandwich panels are investigated for a side of a helicopter fuselage in terms of the global buckling behaviour. All panels have good resistance to the global buckling behaviour under compression loads and the metallic honeycomb sandwich panels can be used in addition to the metallic skin-stringer panels for this purpose. Based on the structural and manufacturing, environmental requirements, their usage can be purposed for a side of a helicopter fuselage and it can give better buckling performance as well as weight. In addition, two panel concepts can be used together at the same time if they are in line with the structural requirements.

As for the future work, this study can be carried out with imperfection effects for non-linear buckling analysis and it can be investigated for post buckling regimes on the panels in order to see the ultimate buckling strength. Another recommendation is that these panels can be designed as pure or hybrid composites assemblies so that they can be worked on linear and non-linear buckling behaviours.

REFERENCES

- [1] Fuselage section of a Boeing 747, taken at the Science Museum in London, (July 2006). Retrieved from <http://www.wikipedia.com/>
- [2] Composition of Brazed Aluminium Honeycomb panels, (n.d). Retrieved from <https://honeytechpanels.com/>
- [3] Paul, A., & George, S. P. (2014, October). Buckling analysis of wing upper skin panels of a transport aircraft. *International Journal of Science, Engineering and Technology Research*, 3(10), 2868-2872.
- [4] Timoshenko SP, Gere JM. *Theory of Elastic Stability*. Second ed. New York: Mc Graw-Hill; 1961.
- [5] Stein, M., Ne, J., Buckling stresses of simply supported rectangular at plates in shear, NACA-TN-1222, 1947.
- [6] Kromm, A., Critical shear stress of rectangular plates with diagonal bracing, steel construction, Vol. 21, No. 10, 1952.
- [7] Lundquist, E. E., Kuhn, P., Katzoff, S., Lévy, S., Theodorsen, T., Harris, T. A., ... & Chiarito, P. T. (1942). Critical Compressive Stress for Flat Rectangular Plates Supported Along All Edges and Elastically *Restrained Against Rotation Among the Unloaded Edges* (No. 731-740). US Government Printing Office.
- [8] Bedair, O., & Sherbourne, A. (1995). Unified approach to local stability of plate/stiffener assemblies. *Journal of engineering mechanics*, 121(2), 214-229.
- [9] Thielemann, W. (1950). *Contribution to the problem of buckling of orthotropic plates, with special reference to plywood* (No. NACA-TM-1263).
- [10] Klinzmann, A., & Horst, P. (2008). Buckling analysis of triangularly stiffened cylindrical panels and their application to aircraft structures. In *49th AIAA/ASME/ASCE/AHS/ASC Structures, Structural Dynamics, and Materials Conference, 16th AIAA/ASME/AHS Adaptive Structures Conference, 10th AIAA Non-Deterministic Approaches Conference, 9th AIAA Gossamer Spacecraft Forum, 4th AIAA Multidisciplinary Design Optimization Specialists Conference* (p. 2121).
- [11] Pflüeger, A., Stability problems in elastic statics, Springer, 3rd edition 1975.
- [12] Wilckens, D., Odermann, F., & Kling, A. (2013). Buckling and post buckling of stiffened CFRP panels under compression and shear-test and numerical analysis. In *54th AIAA/ASME/ASCE/AHS/ASC Structures, Structural Dynamics, and Materials Conference* (p. 1824).

- [13] Bai, R., Lei, Z., Wei, X., Tao, W., & Yan, C. (2017). Numerical and experimental study of dynamic buckling behavior of a J-stiffened composite panel under in-plane shear. *Composite Structures*, 166, 96-103.
- [14] Seydel, E. D. G. A. R. (1933). Über the buckling of rectangular, isotropic or orthogonally anisotropic plates under shear stress. *Engineer Archive*, 4, 169-191.
- [15] Budiansky, B., Connor, R. W., & Stein, M. (1948). *Buckling in shear of continuous flat plates* (No. NACA-TN-1565).
- [16] Rockey, K. C., & Cook, I. T. (1962). Shear buckling of clamped and simply-supported infinitely long plates reinforced by transverse stiffeners and a central longitudinal stiffener. *Aeronautical Quarterly*, 13(2), 95-114.
- [17] Paik, J. K., & Thayamballi, A. K. (2000). Buckling strength of steel plating with elastically restrained edges. *Thin-Walled Structures*, 37(1), 27-55.
- [18] Muameleci, M. (2014). *Linear and nonlinear buckling analyses of plates using finite element method*. MS Thesis, Linköping university, Department of management and engineering, Linköping.
- [19] Bisagni, C., & Vescovini, R. (2009). Analytical formulation for local buckling and post-buckling analysis of stiffened laminated panels. *Thin-Walled Structures*, 47(3), 318-334.
- [20] Stamatelos, D. G., Labeas, G. N., & Tserpes, K. I. (2011). Analytical calculation of local buckling and post-buckling behavior of isotropic and orthotropic stiffened panels. *Thin-Walled Structures*, 49(3), 422-430.
- [21] M. Shama, *Buckling of Stiffened Panels*, pp. 267–304. Springer Berlin Heidelberg, 2013.
- [22] Mantzaroudis, V. K., & Stamatelos, D. G. (2020, December). An approximate closed-form buckling solution for the local skin buckling of stiffened plates with omega stringers: The case of antisymmetric cross-ply and angle-ply laminations. In *Structures* (Vol. 28, pp. 1196-1209). Elsevier.
- [23] Kollar, L. P., & Springer, G. S. (2003). *Mechanics of composite structures*. Cambridge University Press.
- [24] Mittelstedt, C., & Beerhorst, M. (2009). Closed-form buckling analysis of compressively loaded composite plates braced by omega-stringers. *Composite Structures*, 88(3), 424-435.
- [25] Ullah, S., Zhou, J., Zhang, J., Zhou, C., Wang, H., Zhong, Y., ... & Li, R. (2020). New analytic shear buckling solution of clamped rectangular plates by a two-dimensional generalized finite integral transform method. *International Journal of Structural Stability and Dynamics*, 20(02), 2071002.

- [26] Yu, C. (2003). *Buckling of rectangular plates under intermediate and end loads*. MS Thesis, National university of Singapore, Department of civil engineering, Singapore.
- [27] Britt, V., & Simmons, F. (2005). Failure Analysis of Large Stiffened Metallic Fuselage Panels Subjected to Compression Loading. In *46th AIAA/ASME/ASCE/AHS/ASC Structures, Structural Dynamics and Materials Conference* (p. 2008).
- [28] Mateus, A. (2021). *Buckling and ultimate buckling strength for stiffened panels*. MS Thesis, Technical university of Lisbon, Department of naval architecture and ocean engineering, Portugal.
- [29] Georgiou, D. (2019). *Buckling and ultimate strength of stiffened panels*. MS Thesis. National technical university of Athens, Department of naval architecture and marine engineering, Greece.
- [30] Ahamed, S. R., Luqman, S. M., Purushotham, G., Dsouza, V., & Farzana, S. (2018). Comparison of buckling strength between metallic and composite panel. *Materials Today: Proceedings*, 5(1), A1-A10.
- [31] Sayyad, A. S., & Ghugal, Y. M. (2017). Bending, buckling and free vibration of laminated composite and sandwich beams: A critical review of literature. *Composite Structures*, 171, 486-504.
- [32] Abrate, S., & Di Sciuva, M. (2017). Equivalent single layer theories for composite and sandwich structures: A review. *Composite Structures*, 179, 482-494.
- [33] Carlsson, L. A., & Kardomateas, G. A. (2011). *Structural and failure mechanics of sandwich composites* (Vol. 121). Springer Science & Business Media.
- [34] Brischetto, S. (2017). Exact three-dimensional static analysis of single-and multi-layered plates and shells. *Composites Part B: Engineering*, 119, 230-252.
- [35] Ingrole, A., Hao, A., & Liang, R. (2017). Design and modelling of auxetic and hybrid honeycomb structures for in-plane property enhancement. *Materials & Design*, 117, 72-83.
- [36] Rose, C. A., Moore, D. F., Knight, N. F., and Rankin, C. C., 2002, "Finite Element Modelling of the Buckling Response of Sandwich Panels," Proceedings of the 43rd AIAA/ ASME/ ASCE/ AHS/ ASC Structures, Structural Dynamics, Materials Conf., Denver, CO.
- [37] Hua, H., Belouettar, S., Potier-Ferry, M., and Makradi, A., 2009, "A Novel Finite Element for Global and Local Buckling Analysis of Sandwich Beams," *Composite Structures*, 90, pp. 270–278.

- [38] D'Ottavio, M., Polit, O., Ji, W., & Waas, A. M. (2016). Benchmark solutions and assessment of variable kinematics models for global and local buckling of sandwich struts. *Composite Structures*, 156, 125-134.
- [39] Shah, U., & Kapania, R. K. (2019). Failure of Alternate Honeycomb Core Sandwich Panels. In *AIAA Scitech 2019 Forum* (p. 0206).
- [40] Petras, A., & Sutcliffe, M. P. F. (1999). Failure mode maps for honeycomb sandwich panels. *Composite structures*, 44(4), 237-252.
- [41] Paik, J. K., Thayamballi, A. K., & Kim, G. S. (1999). The strength characteristics of aluminium honeycomb sandwich panels. *Thin-walled structures*, 35(3), 205-231.
- [42] Magnucki, K., Jasion, P., Krus, M., Kuligowski, P., & Wittenbeck, L. (2013). Strength and buckling of sandwich beams with corrugated core. *Journal of theoretical and applied mechanics*, 51(1), 15-24.
- [43] Vinson, J. R. (1986). Optimum design of composite honeycomb sandwich panels subjected to uniaxial compression. *AIAA journal*, 24(10), 1690-1696.
- [44] Coburn, B. H., & Weaver, P. M. (2016). Buckling analysis, design and optimisation of variable-stiffness sandwich panels. *International Journal of Solids and Structures*, 96, 217-228.
- [45] Jeyakrishnan, P. R., Chockalingam, K. K., & Narayanasamy, R. (2013). Studies on buckling behavior of honeycomb sandwich panel. *The International Journal of Advanced Manufacturing Technology*, 65, 803-815.
- [46] Aircraft Structural Analysis Training Books, TML-EAADS Unclassified, January 2012, Netherlands.
- [47] Column Buckling: Types and Causes of Buckling, (n.d). Retrieved from <https://civilrack.com/>
- [48] Szilard, R. (2004). *Theories and applications of plate analysis*. John Wiley & Sons, Inc., Hoboken, New Jersey.
- [49] DNV Class guidelines, 2018, Norway.
- [50] Yao, T. and Fujikubo, M., 2016. *Buckling and ultimate strength of ship and ship-like floating structures*.
- [51] Calculating the Johnson-Euler Buckling Load, (June, 2022). Retrieved from <https://blog.engineeringpaper.xyz/>
- [52] Brush, D. O., Almroth, B. O., & Hutchinson, J. W. (1975). Buckling of bars, plates, and shells.
- [53] MSC Nastran 2017.1, Implicit Nonlinear (SOL 600) User's Guide.

- [54] Column buckling calculator, (December, 2021). Retrieved from <https://calcresource.com/>
- [55] Bak, M., 2014. Non-linear Buckling analysis using Workbench v15. CAE Associates.
- [56] I. Sheinmann, M. Bass and O. Ishai, “*Effect of Delamination on Stability of Laminated Composite Strip*”, Composite Structures, Vol. 11, p 227-242, 1989.
- [57] Hard Landings and Shear Buckling, (August, 2019). Retrieved from <https://www.eng-tips.com/>
- [58] Local buckling in airplane fuselage (Belesis and Labeas, 2010). Retrieved from <https://www.researchgate.net/>
- [59] Volume Problems in Real Crustal Deformation, (n.d). Retrieved from <http://homepage.ufp.pt/>
- [60] Bruhn F.R. *Analysis and Design of Flight Vehicles Structures*, 1973.
- [61] Paik, J.K. and Thayamballi, A.K, 2003. *Ultimate limit state design of steel-plated structures*. Chichester: John Wiley& Sons.
- [62] Structural Materials Handbook Part 3, Space Engineering, March 2011, Netherlands.
- [63] Zenkert D, “*An Introduction to sandwich construction*”, Chameleon Press Ltd., UK, 1995.
- [64] Practice Guide for Sandwich Structures in Marine Applications, New Rail, University of Newcastle, (n.d).
- [65] Aircraft Structural Design Training Books, TML-EAADS Unclassified, January 2012, Netherlands.
- [66] ScienceDirect, Aircraft Structures for Engineering Students, (2013). Retrieved from <https://www.sciencedirect.com/topics/engineering/fuselage/>
- [67] Kumar, J. V., Sreenivasulu, V., Rao, P. D., Kiran, C. U., & Kumar, V. (2013). Static and dynamic analysis of aircraft stiffened panel. *International Journal of Innovations in Engineering and Technology*, 3(2), 134-139.
- [68] Airbus A330-743L Beluga XL Moulds in 1:400 Scale & Detailed Comparison, (November, 2020). Retrieved from <https://www.yesterdaysairlines.com/>
- [69] Paik, J. K., & Duran, A. (2004). Ultimate strength of aluminium plates and stiffened panels for marine applications. *Marine technology and SNAME news*, 41(03), 108-121.
- [70] Cabin, F. S. P. S. T. (2012). Finite Element Modelling and Analysis of Fuselage Stiffened Panel Subjected to Cabin Pressurization.

- [71] Wing panel wrinkle, (September, 2011). Retrieved from <https://vansairforce.net/>
- [72] Excessive derotation on landing buckled Delta 757's fuselage, (May, 2020). Retrieved from <https://www.flightglobal.com/>
- [73] Zhou, Z. Q., Chiu, W. K., & Bannister, M. K. (2006, December). Health monitoring of aircraft fuselage structures using ultrasonic waves. In *Proceeding of 25th international congress of the aeronautical sciences, Hamburg, September of* (pp. 3-8).
- [74] Bekdaş, Y., Büyükkoyuncu, M., Çiftçi, E., TAI, T. A. I., Erdurmaz, B., Gülbaş, B., & Taşdemirci, A. Optimization of Integrally Stiffened Panels in Aircraft Wings.
- [75] Xianfei, Z., Yunwen, F., Xiaofeng, X., & Qiang, Q. (2017). Evaluate the crashworthiness response of an aircraft fuselage section with luggage contained in the cargo hold. *International journal of crashworthiness*, 22(4), 347-364.
- [76] How should I decide on the type of stringer to be used in an Aircraft's wing and fuselage, (November,2021). Retrieved from <https://aviation.stackexchange.com/>
- [77] Mekki, K., Derigent, W., Rondeau, E., & Thomas, A. (2017). Wireless sensors networks as black-box recorder for fast flight data recovery during aircraft crash investigation. *IFAC-PapersOnLine*, 50(1), 814-819.
- [78] Jędral, A. (2019). Review of testing methods dedicated for sandwich structures with honeycomb core. *Transactions on Aerospace Research*, 2019(2), 1-14.
- [79] Alderliesten, R. (2018). Introduction to aerospace structures and materials. Delft University of Technology, the Netherlands.
- [80] Federal Aviation Administration. (2013). MMPDS-08: Metallic Materials Properties Development and Standardization. Washington, D.C.
- [81] Pedatzur, O. (2003). An evaluation of finite element models of stiffened plates subjected to impulsive loading (Doctoral dissertation, Massachusetts Institute of Technology).
- [82] ABAQUS, "Version 6.14 user's manual", SIMULIA, World Headquarters, Providence, RI, USA, 2014.
- [83] MSC Nastran 2021.4: Linear Static Analysis User's Guide. U.S.: Hexagon, 2021.
- [84] Klarholm, D. (2016). Offset modelling of shell elements: a study in shell element modelling using Nastran.
- [85] Altair Hypermesh, "Version 2019 User's Manual", USA, 2019.

- [86] Basha, M. (2016). *The mechanical buckling behaviour of composite sandwich structure made from hybrid glass/carbon face sheets and rigid foam core*. Ecole Polytechnique, Montreal (Canada).
- [87] Park, J. S., Paik, J. K., & Seo, J. K. (2018). Numerical investigation and development of design formula for cylindrically curved plates on ships and offshore structures. *Thin-Walled Structures*, 132, 93-110.
- [88] St. Venant's Principle, (2022). Retrieved from <https://help.solidworks.com/>

