Optimal and robust design of materials for wind turbine

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Optimal and Robust Design of Materials for Wind Turbine

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Summary

The world today is continuously striving towards carbon neutral clean energy technology. Hence, renewables like wind power systems are increasingly receiving the attention of mankind. Energy production is now no more the sole criterion to be considered when installing new megawatt (MW) range of turbines. Rather some important design parameters like material choice, cost, carbon footprint, embodied energy, life cycle impact efficiency and, above all, the structural rigidity needs to be considered holistically. Accordingly, these issues are followed up in this dissertation from the wind industry perspective.

The study, at the outset, performs a detailed life cycle assessment (LCA) of three wind farms to explore long term impacts of today's MW range wind turbines. Out of these, the vertical axis wind farm performs the best; followed by offshore horizontal axis and onshore horizontal axis farm. Corresponding studies consequently discover most adverse life cycle impact contributing materials and subsequently replace them with superior ones which eventually reduce overall LCA impacts by 30%, 25%, 7% for modified HAWT onshore, HAWT offshore and vertical axis farm, respectively. Eventually, a comparison of all major renewable and non-renewable power generation sources reveal the wind farms as competing remarkably with all the technologies except the hydro-kinetic ones.

Next, the study aims to distinguish best blade and tower materials in comparison to presently used ones in line with multiple constraint and compound objective based design optimization principle whereby mass, carbon footprint, material cost and embodied energy minimization are achieved simultaneously from material indices derived of aerodynamic and structural performance equations. In this way, final onshore wind turbine blade material ensures 74% mass, 17% carbon footprint, 30% embodied energy reduction in

comparison to existing Vestas 3.0 MW glass/epoxy blade. In turn, a cast iron BS 900/2 material tower ensures 6% mass, 81% carbon, 80% energy content and 90% cost reduction in comparison to present tower material. Natural composite materials like oak, *lignum vitae* also enter into small scale onshore blade design space for low wind regions. Composite materials appear as one of the most potential material families from both material indexing and life cycle assessment study.

Hence, the study is extended on how to ensure material efficiency in composite structure through fracture characterization under experimental biaxial loading. Owing to their anisotropic nature, composite materials respond differently under multi-axial loading conditions with reference to their uniaxial behavior and, hence, characterization of its multi-axial stress properties is of utmost importance for wide scale, efficient usage in numerous structures including wind turbine blades. A new biaxial test frame is developed, relevant optimized geometry is achieved, relevant geometric factors corresponding to cruciform profile is determined and, based on these, glass fibre composite material is characterized for its fracture performance with different crack dimensions and orientations. Under biaxial tensile-tensile loading condition, composite material is able to sustain more load than uniaxial case which opens the door for less conservative design and increased material efficiency.

A companion study is performed in pursuit of a mass minimal design for thin cylinder structure under eccentric compressive loading, as a representative form of wind turbine tower. In this discourse, axial crushing failure is studied first under non-uniformly distributed loading. Next, leading compressive failure modes such as global buckling, yield, local buckling are studied with respect to non-dimensional load and shape factor. Optimal failure boundary zones are attained simultaneously with minimal mass and optimum shape factor based

design perspective. Final failure boundary plot serves as a design tool for wind turbine tower where eccentric compression is one of the loading conditions. Findings from these studies can be deployed to harness massive scale wind energy from structurally more promising, economically more competitive and environmentally more clean and green turbine.

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Nomenclature

ρ	Density of material
ρ_a	Density of air
ρ_{elr}	Electrical resistivity of material
Ω	Blade rotational speed (rpm)
σ	Failure strength of thin shell structure
σ_b	Maximum compressive stress due to bending moment
σ_c	Uniform compressive stress due to axial load
σ_f	Fatigue strength of material (Chapter 3); Failure strength of notched specimen (Chapter 5)
σ_{flap}	Flapwise bending stress
σ_{gb}	Global buckling failure stress
σ_{lb}	Local buckling failure stress
σ_{shell}	Edgewise bending stress
σ_x	Transverse stress
σ_y	Axial stress (Chapter 5); Yield failure stress (Chapter 6)
σ_{yff}	Far-field uniform y-direction stress, away from crack tip
σ_{ys}	Yield strength of the material
ψ	Solidity ratio
ν	Poisson ratio
α	Thin shell strength reduction factor
ϕ_B^e, ϕ	Elastic bending shape factor; Ratio between radius and thickness
ϕ_{opt}	Optimal shape factor
γ	Thin shell safety factor
$2a$	Crack length
c	Average chord length
d	Average chord width
e	Eccentricity
$\bar{e}$	Eccentricity ratio
m	mass

$\dot{m}$	Wind mass flow rate
m_{blade}	Blade mass
m_{gb}	Mass with respect to global buckling equation
m_{lb}	Mass with respect to local buckling equation
m_{min}	Minimal mass
m_{spar}	Spar mass
m_{shell}	Shell mass
m_{tower}	Tower mass
m_y	Mass with respect to yield equation
r	Radial distance from rotor axis to blade element
r_g	Radius of gyration
$t/2$	Blade shell thickness
t	Tower mean wall thickness (Chapter 3); Thickness of thin shell (Chapter 2, 6)
w_i	Weight factor percentage
w_{max}	Maximum transverse displacement in global buckling mode
A	Cross-sectional area
$A_{equiv. x}$	Equivalent loaded area under transverse force
$A_{equiv. y}$	Equivalent loaded area under axial force
A_2	Rotor swept area
$An-on$	Onshore turbine ancillary sub-assembly
$An-off$	Offshore turbine ancillary sub-assembly
A/E	Acidification/Eutrophication
C	Carcinogen (Chapter 3); Distance between neutral axis to tower outer skin (Chapter 4)
C_c	Coupling constant
C_{CO}	Cost per unit weight
C_{CO2}	Carbon footprint per unit weight
C_{EE}	Embodied energy per unit weight
C_f	Distance from neutral axis to spar end in flapwise load direction

C_e	Distance from neutral axis to shell end in edgewise load direction
C_m	Per unit material cost
CC	Climate Change
D	Tower mean outer diameter (Chapter 3); Diameter of thin shell (Chapter 2, 6)
D_{bottom}	Tower bottom outer diameter
D_{top}	Tower top outer diameter
E	Ecotoxicity (Chapter 3); Young's modulus of material (Chapter 4-6)
$\dot{E}_{extracted}$	Extracted power from rotor
EQ	Ecosystem Quality
F	Aerodynamic force on rotor
$F-on$	Onshore turbine foundation part
$F-off$	Offshore turbine foundation part
FF	Fossil Fuels
$G_{1C,I}$	Mode I critical strain energy release rate
$HAWT$	Horizontal axis wind turbine
HH	Human Health
I	Area moment of inertia of the cross-sectional area
K_I, K_{II}	Mode I, mode II stress intensity factor; respectively
$K_{K_{IC,I}}$	Mode I critical interlaminar stress intensity factor
$K_{K_{IC,T}}$	Mode I critical translaminar stress intensity factor
L	Tower length (Chapter 3); Length of thin walled structure (Chapter 2, 6)
LU	Land Use
M	Minerals (Chapter 3); Bending moment (Chapter 4)
M_{bend}	Bending moment at tower base
M_{edge}	Blade edgewise bending moment
M_{flap}	Blade flapwise bending moment
MI	Material index
OL	Ozone layer depletion

P	Buckling load on tower (Chapter 3); Axial load (Chapter 6)
P_{avg}	Average crushing load
P_x	Transverse force
P_y	Axial force
PDF	Potentially disappeared fraction
R	Radiation (Chapter 3); Blade radius (Chapter 4)
Re	Resources
RI	Respiratory Inorganics
RO	Respiratory Organics
T	Rotor thrust
$T-on$	Onshore tower
$T-off$	Offshore tower
V_1	Free stream wind velocity
V_2	Wind velocity through rotor
$VAWT$	Vertical axis wind turbine
W	Width of the notched specimen
X_1, X_2, X_3	Eccentricity based factors
Y_1, Y_2	Non-dimensional factor associated with K_I and K_{II} of cruciform geometry

Chapter 1

Introduction

1.1 Background

Electricity is one of the greatest discoveries in the history of mankind. Its versatile use and limitless set of applications paved the way of human civilization since time immemorial. Now-a-days, it influences the human being in every sphere of life and appears to be the backbone of contemporary civilization. The major breakthrough in power production system ushered in the day the world discovered its first electricity generation station on the eve of 19th century. Since then, worldwide electricity consumption increased by leaps and bounds throughout the 20th century. Contrary to this, energy supply scenarios have changed greatly in recent decades. Energy decision makers now encounter some grave concerns which were unknown and, hence, untouched in previous centuries. For instance, world's fossil fuels that accelerated the drive to the age of modern civilization and powered the world's industries have now been marked as pre-dominant green house gas emitters in earth's atmosphere [1]. Burning fossil fuels release large scale green house gases (GHG) and numerous other

environmental contaminants that ultimately leads to global warming which appears as a key problem of the hour. The United Nations (UN) sponsored intergovernmental panel on climate change (IPCC) provides a grave summary that the global temperature may rise by up to 5.8°C over the next century [2]. Simultaneous forecasts from a large number of renowned estimates about rapid reduction of various fossil fuel sources also appear as a major concern. Coupled with these facts, other burning issues, viz., volatile crude oil prices, energy insecurity and geo-political instability matters arise as key deciding factors. These grim realities eventually provide huge impetus for embarking on a state-of-the-art energy platform with clean and green outlook. As being one of the most ubiquitous, in-exhaustible and sustainable energy on planet earth, wind power plays a consequential role in this context. Due to its enhanced competitiveness, wind power industry is experiencing spectacular growth in recent years. With government thrust and deployment of latest technological know-how, new megawatt (MW) range turbines have eventually been evolved; some of which are even larger than latest largest aircrafts (e.g., Boeing 787, Airbus A380). Wind turbines are already installed in commercial basis in more than 80 countries, and combined rated capacity of these turbines reached to about 300 GW [3].

Many researchers are simultaneously exploring new avenues of cutting edge researches in various spheres of this technology. Additionally, as national electricity price is rapidly rising in many countries, renewable energies like wind power systems evolve to be more competitive with conventional grid-connected electricity [1, 4]. From this perspective, this thesis focuses on key research areas of very large wind turbine in the spheres of material optimization, life cycle impact reduction, fracture assessment of composite laminates under bi-axial

fracture loading, and failure mode of tubular towers under eccentric compressive loading.

1.2 Objectives of the thesis

The objective of the present discourse eventually centers on the following topics:

- performing a life cycle assessment (LCA) study within promising wind turbine technologies and subsequent development of impact reduction strategies
- optimization of wind turbine material in light of clean and green energy perspective with superior structural performance, material, cost, carbon footprint and energy efficiency
- fracture characterization of composite laminates under bi-axial loading condition
- competitive failure boundary generation for thin-walled circular tube under eccentric compressive load and prescription for minimum mass design, as a representative form of wind turbine tower.

1.3 Scope of the thesis

In life cycle impact assessment study, the study covers a life cycle inventory (LCI), life cycle assessment (LCA) and life cycle impact analysis (LCIA) of three 50 MW wind farms. One of these farms comprises an on-shore horizontal axis wind turbine, one comprises an offshore horizontal axis wind turbine, while another comprises of a vertical axis wind turbine. *Eco-indicator 99* impact assessment method, which is one of the most widely life cycle analysis methodology, is used for LCA calculation. Eventually, high impact contributing life cycle processes are substituted with low LCA impact carrying ones and

process routes for future better and greener wind farm designs. All major renewable and non-renewable power generation technologies are additionally compared in a uniform unit.

Next, in material selection study, onshore and offshore wind turbine blade and tower materials are optimized based on multiple constraints and conflicting objective based strategies; where mass, carbon footprint, material cost and embodied energy efficiency are achieved simultaneously from material indices derived of structural performance equations. Minimization of various objectives has been carried out based on commercially available *Cambridge Engineering Selector* (CES) software which covers around 4000 engineering materials widely used in today's designs. All the afore-mentioned objectives are duly studied to attain best blade and tower materials which are finally compared with presently used turbine materials.

Next study covers on fracture characterization of blade composite material under experimental biaxial loading. With recognition to imperative necessity for investigating the biaxial properties of various anisotropic materials, a new test frame is developed. Initial key scopes relevant to this project included design of an optimal specimen on the basis of loading and fixture constraints of the test-frame, experimental validation of the test-frame, biaxial characterization of composite laminates and associated fracture failure envelope generation. All these topics are covered in this dissertation for cross-ply, glass-fiber reinforced epoxy matrix composite laminates. Relevant fracture characterization of composite material is performed on the basis of optimized cruciform specimen geometry which resembles plate or gently curved structures, widely used in many composite designs. 40 mm and 45 mm crack lengths oriented at 0° and 45°

are studied hereby to generate relevant fracture profile for symmetric cross-ply glass fibre reinforced plastic laminate.

In another study, a mass minimal design of thin cylinder structure is pursued under non-uniformly distributed compressive loading. For this purpose, axial crushing failure for thin shell structures is studied experimentally with a modified three-point bending tool. Leading compressive failure modes are studied consequently with respect to non-dimensional load and shape factor. Optimal failure boundary zones are attained simultaneously with minimal mass design perspective. Final failure profile serves as an important tool for lots of relevant structures including wind turbine tower under compressive loading based study.

1.4 Outline of the thesis

The thesis is composed of seven chapters including this. Remaining chapters are arranged as follows:

Chapter 2 provides relevant literature review and novelty on four major aspects of the thesis, viz., life cycle assessment and impact reduction, material optimization, biaxial fracture characterization of anisotropic composite material and competitive failure boundary generation of thin cylinder structure under non-uniformly distributed loading.

Chapter 3 addresses life cycle inventory (LCI), life cycle analysis (LCA) and life cycle impact analysis (LCIA) results of three wind farms.

Chapter 4 addresses onshore and offshore wind turbine optimal material selection based on multiple constraints and conflicting objectives. This chapter comprises two aspects: first one centers on material optimization issues of blade while second one targets material optimization of tower.

Chapter 5 addresses the fracture assessment of GFRP composite laminates under bi-axial loading.

The axial compression of thin-walled tubular structures under eccentric loading is presented in Chapter 6. And, finally, the major conclusions of the current research and scope for future research are given in Chapter 7.

Chapter 2

Literature review

2.1 Introduction

A wind turbine is an engineering structure that converts wind's power into electrical power. This is in contrast to a windmill that converts the wind's power into mechanical power. There are two major classes of wind turbine - horizontal and vertical. Conventional wind turbines, i.e., horizontal axis wind turbine (HAWT) spins about a horizontal axis and the vertical-axis wind turbine (VAWT) spins about a vertical axis. Earliest mention of the use of wind power as well as a comprehensive history of wind energy utilization can be explored in number of references [1, 4]. Mankind has used wind power as a source of energy for thousands of years. During the medieval era, wind power was one of the most utilized sources of energy together with hydro power in Middle Eastern region, India, China and Iberian Peninsula. Nevertheless, with the advent of industrial revolution, particularly with the invention of steam engine, wind power lost its interest. Subsequent invention of alternating current (AC) power

generation technologies from fossil fuel sources greatly curbed the development of wind power.

Fig. 2.1: Schematic diagram of a horizontal axis wind turbine [5]

Finally, it was the advancement in aircraft and aerospace technology in the initial decades of 20th century that accompanied a renewed interest in this field. In those

days, per unit wind power generation price was, nevertheless, too high to compete with other power generation technologies [1]. However, grid electricity price increased a lot since then and, simultaneously, many avenues of wind turbine researches have flourished. Per unit wind power generation cost also simultaneously declined over the years [4]. Currently, wind power is a fully established branch in the world energy scenario and is highly regarded accordingly. Figure 2.1 shows a horizontal axis wind turbine with its major components.

Energy production is now no longer the sole criterion to be considered when installing new megawatt (MW) range of wind turbines. Rather some important design issues like structural stability, cost efficiency, environmental impact, embodied energy and reduced weight appear as dominating design issues. The wind turbine design issues are followed up in successive sections depending on these perspectives.

2.2 Life cycle assessment of wind farms and corresponding impact reduction

This section reviews the earlier researches conducted in the domain of life cycle assessment of various wind energy structures. Wind power, as being a renewable source of energy, is perceived to be more competitive in many aspects than traditional fossil fuel based energy generation technologies. Particularly, in comparison to solar power, it is more economic and specially in high latitudes ($>40^\circ$ north or $>40^\circ$ south) it will be long time for solar power to achieve the parity with existing fossil fuel based plants; whereas wind power can be commercially installed in any part of the world with wind speed around 5 m/s or

more. However, from environmental perspective, it is very important to determine how much the environmental effect the wind power carries. To this implication, an overall life cycle inventory (LCI) and life cycle impact assessment (LCIA) study of various wind farms is of profound importance. Besides evaluating the inventory and emission aspects of solely the operation phase, life cycle inventory and impact assessment study performs a cradle to grave investigation of the stand-in technology. To say precisely, it covers the environmental impact of a system from material extraction and refining to manufacturing, transport, operation and maintenance phase to until the final disposal phase. Guidelines of LCI and LCIA are epitomized by industry wide well-reckoned standardization bodies. Out of these, LCA guidelines by International Standard Organization (ISO) cover life cycle documentations extending from LCA principle and framework to goal, scope definition, inventory analysis and interpretation according to ISO 14040 and 14044 standards [6-7]. Based on these guidelines, the study evaluates existing wind farm LCA study and, henceforward, redesigns it for a better carbon neutral, greener environment.

Worldwide only a few researchers have, so far, explored LCA studies for onshore and offshore turbines. Out of these, a study by Fleck and Huot [8] modeled a 400 W rated power, 1.17 m diameter rotor, off-grid, onshore wind turbine to explore its long term environmental impact based on green house gas (GHG) emission and, compared it with a diesel generator system; the diesel generator system produces at least 14 times more emission than the wind power system.

Martinez et al. [9] brought forth an LCA sensitivity analysis of a 2 MW onshore Gamesa wind turbine by varying the data of maintenance, materials, energy and

recycling phases. For nacelle and rotor, it considered 85% of total consumed materials, while for foundation and tower, the cut-off criterion was 95%. Electrical distribution network based LCA processes were completely neglected in the study and finally it reported turbine manufacturing stage as the most environmental impact carrying stage. In accompanying sensitivity studies, emission of respiratory inorganic substances and consumption of fossil fuel impact categories are witnessed to vary in a prolonged range.

Weinzettel et al. [10] performed a process based LCA study on a conceptual floating wind turbine. The turbine is assumed to be installed 50 km from the coast-line at a depth of 100-300 m. One mooring point is set at ocean bottom and the tower is joined to it by a torsion leg. The study did not count any secondary forming process or design activities. The conceptual floating turbine is finally compared based on average life cycle impact values of offshore wind power and natural gas plant. The study finds comparable impact values in various impact categories (abiotic depletion, global warming, acidification, human toxicity) while marine eco-toxicity data is found to be widely deviated and misleading; as the authors mentioned.

A life cycle energy and GHG emission analysis of two representative horizontal axis wind turbines from medium and large scale categories was conducted by Crawford [11]. These are of 850 kW and 3 MW generation capacity with comparable capacity factor. End of service life scenarios were not addressed in the study as because material recovery at the end of project life is never guaranteed. Life cycle analyses for power distribution components and establishment works also have been neglected. After deducting consumed

energy in different life cycle phases, the 20 year study highlighted net life cycle energy of 850 kW and 3 MW turbines as 588 TJ and 2049 TJ, respectively.

All these aforesaid studies, as one important thing in common, proposed more rigorous LCA studies to be conducted, specially, for large scale wind turbines. With the progression of life cycle analysis thinking, impact assessment methods also have improved over the years and, accordingly, employing one of the robust methodologies for LCA calculation also deserves significant attention. Moreover, the wind turbines are never compared in any article at sub-assembly level which is significantly important to determine which parts of the wind turbine are actually contributing more impact. How different primary, secondary and tertiary manufacturing processes are contributing to overall LCA impact is another important issue that left untouched. Furthermore, it is of utmost importance to identify how each and every engineering material contributes to overall wind turbine life cycle; this can eventually pave the way to substitute the existing materials with environment friendly ones. In addition, life cycle analysis of vertical axis wind turbine technology is entirely in its infancy. There is no referred work published in this domain, as prominent citation databases affirm. The present dissertation admires all these issues as potential areas of LCA study. Rigorous LCA studies on all present wind turbine technologies will ultimately identify most adverse impact contributing materials and subsequently replace them with superior ones. Accordingly, these issues will be focused more comprehensively in Chapter 3.

2.3 Material selection for horizontal axis wind turbine blade and tower

Within the last decades, number of engineering materials has increased tremendously. A plethora of new materials is discovered by research community and it is now a well-known estimate that the number of materials has already exceeded 80000 [12]. Many traditional materials that ran the wheels of industrial revolution have been successfully replaced with more suitable, superior quality materials. Wind turbine industry has simultaneously deployed better quality alloys, composite and sandwich materials. Although these materials are meeting today's industry demand, still there is ample opportunity to optimize the wind turbine material selection process from structural and environmental point of view. New families of materials comprising green and recyclable ones appeared recently [13]. Per unit material cost decreased as well for lots of existing materials due to advanced manufacturing and process techniques. Out of these inexpensive materials, some may meet the wind turbine structural demands with even lower mass and environmental impact than the presently used ones. In this context, a thorough investigation on wind turbine material selection process is of utmost importance in the avenue of multiple constraint and compound objective based design optimization procedure.

Fig. 2.2: Load distribution on a horizontal axis wind turbine [14]

Wind turbine, with its increasing size due to 'efficiencies of scale' and 'specific output' concern, is now one of the largest man-made structures. Over a design life of 20 years, this rotating machine endures highly turbulent aerodynamic loads and numerous fatigue producing stress cycles. A large turbine typically experiences 10^8 to 10^9 cycles at an annual average 4000 hours operation [15]. This

may be compared to many other manufactured items which would be unlikely to experience more than 10^6 cycles over their entire life [15]. In addition to the aerodynamic loadings, there is gravitational, centrifugal, gyroscopic, inertia, brake and other loads [16-18]. Interaction of these loadings together with blade aerodynamic profile makes the real-time loading scenario very complex. Designing such an aerodynamic structure simultaneously for decreased mass, cost, environmental impact and embodied energy obviously leads to a compound objective based design optimization problem. Figure 2.2 shows axial and tangential component of the aerodynamic force acting on a three-bladed horizontal axis wind turbine.

Within the last decades, many material selection and optimization problems are explored in scientific and design community. Out of these, Zhu et al. [19] proposed a knowledge-based design support system (KDSS) for optimal material selection of energy absorbers based on axiomatic design principle [20]. They presented three functional requirements (so called 'what we want to achieve' factors) with corresponding, relevant design solutions ('how to achieve' factors). A purpose-built software, Swinburne Energy Absorber Design Supporter (SEADS), is used to demonstrate the procedural aspects of the material selection. Functional requirements of this process include knowledge based optimal design search destined for ranking the materials, knowledge-based optimal design calculation that influences the selection if appropriate data are unavailable and concurrent 'supporting knowledge demonstration' process in the form of user guide, video and/or failure analysis which guides the entire decision compromise paradigm.

Waterman et al. [21] evaluated various existing computerized material data and information systems in pursuit of further rationalizing and streamlining the concomitant objectives and requirements of the materials and design engineers. Sapuan [22] highlighted genuine importance of computer aided material selection systems with added emphasis on material optimization procedures. Distinct knowledge sharing patterns are explained in addition to how knowledge based system (KBS) contributes towards advanced material selection method and product design.

In turn, Sirisalee et al. [23] focused on multi-objective optimization which is hinged on utility function approach with trade-off surfaces, Pareto front, associated dominated and non-dominated solution. Trade-off surfaces and their tangential material contours offer the best compromising solution between different conflicting objectives. The study demonstrated material selection strategy for one disk brake caliper and another mini disk player casing, based on conflicting objectives; which can provide sound representation of trade-off surfaces for maximum three objectives.

Overall, all material screening tools can be categorized in five major sub-classes; viz., artificial intelligence based method, cost per unit property method, questionnaire based method, materials in products selection (MiPS) tools and index and chart based methods [24]. Out of these, artificial intelligence based methods work on how to make computers perform intelligent things for the processing of unstructured and scattered knowledge for solving the complex problems. Now-a-days, no engineer expects to know more than a small division of ever-growing body of information in material behavior and here is the necessity of artificial intelligence based methods. These methods employ

interactive computer programs and databases to sort out competitive material with desired properties [25-27].

Cost per unit property method explores trade-off between cost and desired performance indicator [28]. Since cost is one of the most important factors in material selection process; it is logical to consider it at the start of selection process. A target cost is generally set in this method to remove the materials above the expected limit. Thus, the final choice is a trade-off between cost and performance. Cost/unit tensile strength (\$/MPa) is usually considered the most important design criteria and, accordingly, materials with lower cost/unit strength are preferred over others. Main limitation of this process is that it considers only one property as critical and ignores the others.

Questionnaire based methods employ question based tools to determine user-product interaction aspects [29-30]. One questionnaire based material selection process classified performance requirements of material into two main categories: (i) rigid or go-no-go requirements and (ii) soft or relative requirements [28]. Rigid requirements should be met by the material. Such requirements are used for initial selection of materials whereby unsuitable ones are eliminated. Soft, or relative, requirements, on the other hand, are subjected to negotiations and trade-offs. Another research group suggested a list of questions for achieving an optimal design solution, which are as follows [29]:

- Have all the related discrete material properties been obtained and understood?
- Have all the environmental conditions been taken into consideration?
- Have all the economic limitations been taken into consideration?
- Will the design conditions change with time?

- Have the effects of materials processing conditions been considered?
- Have the effects of quantities and rate or production of components been effectively considered?
- Has future raw material accessibility been considered?

Van Kesteren et al. [24] proposed a question tool for user-interaction aspects of materials. The tool is composed of a list of questions for different phases of user-product interaction and a checklist of sensorial properties. Another material selection guideline is proposed by Farag et al. [28] based on five performance requirements; viz., functional requirement, process-ability requirement, cost, reliability and resistance to service condition. Here, functional requirement covers material properties, process-ability requirement encompasses manufacturing issues, reliability considers overall design setting and resistance to service condition covers several environmental features that the product or structure is intended to experience.

Index and chart based method, as another alternative, makes a single, straight forward assessment of engineering materials. Index and chart based methods are systematic and unbiased in their goals to optimize the performance objectives [31-32]. Material and process selection approach proposed by Ashby et al. falls into this category [33-34]. Ashby method is applied in this dissertation to find optimum material for wind turbine blade and tower application. Ashby approach, at the outset, translates the design requirements into performance objectives. Performance objectives are basically aims or targets that are achieved throughout the design exercise. The minimum or maximum level to which these objectives are achieved are dictated by constraints. Simple mathematical transformation of equations leads to get the appropriate material index for each

performance objective guided by selected design constraints. Surviving materials are then ranked to sort out the best material.

Earlier, Grujicic et al. [35] have explored Ashby material optimization approach for horizontal axis wind turbine blade. This study, however, considered only one objective (mass minimization) and did not consider any flapwise or edgewise bending load which is significantly important from wind turbine aerodynamics perspective. Wind turbine power rating goes square of the blade size and also more wind speed can be obtained with longer tower (hellman exponent effect) to generate more power (as wind turbine power goes cubic to wind speed) [15]. In the same time, both blade and tower mass scales with increasing size. Hence, material consumption of wind turbine is increasing with wind turbine size growing bigger in quest for more power generation outlook. As highlighted by Breton and Moe [99] and Dutton et al. [100], rotor mass is a potential design factor for further development of onshore and offshore wind turbine industry. In addition, increasing mass also increases gravity load in blade with implied increase in centrifugal, braking and gyroscopic/yaw force. Hence, a comprehensive mass minimization study is of significant importance based on proper material index. For tower, being the heaviest component of wind turbine, its minimum weight material selection is also crucial; however, it is not explored in any study before.

Moreover, there are other objectives, e.g., less expensive, more environment friendly design of wind turbine components which are significantly necessary for enhanced adoption of this power technology. Thus a multi objective material optimization of wind turbine blade and tower is of significant importance based on appropriate material index. Accordingly, a multiple constraint and compound

objective based material optimization study is presented in Chapter 4 for horizontal axis wind turbine blade and tower material based on four objectives; viz., mass, carbon footprint, cost and embodied energy minimization.

2.4 Fracture characterization of glass fiber composite material under biaxial loading

As afore-mentioned LCA and multi objective material selection studies progress in Chapter 3 and Chapter 4, respectively, composite materials appear as one of the most potential material families for wind turbine application. Accordingly, subsequent research is extended on how to ensure material efficiency in composite structure through fracture characterization under experimental biaxial loading. Fracture mechanics primarily deals with various discontinuities inside a material, e.g., presence of crack, imperfection, flaws and their effect on failure. In presence of discontinuity, materials usually fail earlier to their neat state. Due to their inherent anisotropy, the situation plays more complex role in composite materials. Moreover, whereas loading direction does not play a role for ultimate failure behavior in isotropic materials; this is generally not the case for anisotropic material. Further to that, composite materials fail catastrophically (brittle mode) and, specially, presence of imperfection is also a normal occurrence for these materials due to their distinct manufacturing processes that differ from metal manufacturing. From this perspective, fracture characterization of biaxial or multi-axial failure response of composite material is a pivotal research area. Contemporary research, nevertheless, did not progress as much as the attention is required in this direction. This stands as a significant limitation towards wide scale, efficient usage of composite material in various engineering

structures. A brief description on composite materials follows next with their ever-increasing usage role in wind turbine industry.

Composite materials are some engineered or naturally occurring materials constituted from two or more base components of different physical and chemical properties that remain separate and distinct at both macroscopic and microscopic scale within a processed structure. Because of high specific strength and stiffness, these materials are widely used in our time. Numerous engineering products extending from tiny items to boats, hulls, racing car bodies and military equipments, massive aircrafts and space vehicles are being manufactured of composite materials. In pursuit of attaining even higher specific strength and specific stiffness, many design communities endeavor to develop engineering structures completely based on composite materials. An all-composite military vehicle, high mobility multi-purpose wheeled vehicle (*HMMWV or Hummvee*), is introduced in line with this objective. Boeing 787 dreamliner, one of the latest aircraft, also focused on massive use of composite materials at 50% by weight and 80% by volume, respectively [36].

Wide scale usage of composite materials in wind turbine blade and hub is also common for long time. Specially, glass fiber reinforced plastic (GFRP) composite, with its high fatigue and stiffness properties, gained much popularity in wind industries. GFRP, designed during the end of 1940's, is itself one of the most widely used materials within composite family. GFRP material has many variations depending on different fibers such as A-glass, C-glass, D-glass, E-glass, R-glass, S-glass, etc. So far, in wind turbine blades, E-glass and S+R glass fiber based composites are extensively used. Out of these, E-glass is the most widely used one. It carries effective insulation and low water absorption rate in

addition to low cost. S+R glass also carries high performance properties with accompanying low cost. Figure 2.3 shows a composite wind turbine blade in a constituent scale.

Fig. 2.3: Various scale of wind turbine blade construction [37]

World's first FRP composite wind turbine rotor blade can be traced back to 1959 [38]. Ulrich Hütter first developed a 34 m wind turbine rotor using glass-fibre composite material. Often referred to the wind industry as '*father of composite wind turbine blade design*'; his ideas significantly influenced later designs. Variants of his design were commissioned by different blade manufacturers until late eighties. Specially, early Danish commercial wind turbine blades largely

followed Hütter's monolithic skin based blade design. From manufacturing point of view, it was a good installation strategy but it lacked sufficient strength carrying capability due to unavailability of shear web or box support structure which bears massive flapwise loading resulting from the axial component of aerodynamic force. At the end of 1970s, single shear web concept appeared to improve the design to withstand the wind force better for larger blade series. This also reduced tip deflection and further improved the stability of blade skins. Thus the use of unidirectional fibers placed above and below the shear web became more prevalent.

Fig. 2.4: Evolution of structural design concepts, materials and processes for FRP composite wind turbine blades (1975-present) [1, 2, 3 indicate blades with monolithic design, single web and double web/spar concept, respectively] [37]

Later on, introduction of sandwich panels at leading and trailing edge section of blade contributed to establish the basis of a design that would be considered the wind turbine industry norm until early 1990s [37]. During the past 15 years, largely due to increasing gravity loads with longer blade length, one last significant change in structural topology occurred with the introduction of a second shear web [as shown in Fig. 2.4]. This configuration helped concentrate the loads within a “box spar” included between the shear webs; thereby, relieving the leading edge and trailing edge section from in-plane loads. Two main strategies are now favored by manufacturers to construct blades with this configuration. They either build the spar in a single step with the non-structural skins or use secondary bonding of the two skins and shear webs to form the blade structure.

In retrospect, the last 30 years of FRP composite wind turbine blade development has yielded three distinct structural concepts: the monolithic skin concept (Hutter’s design), the single shear web design and the double shear web/box spar concept. Past three decades also noticed considerable improvements in blade manufacturing techniques as most manufacturers moved from the classic hand lay-up process to vacuum infusion methods or pre-pregging. Measures were taken to improve the blade performance with improved resin systems from basic polyester-based matrix to vinylester and epoxy matrices. The wind energy composite market, thus, witnessed an impressive steady growth rate of 20% over the couple of years and is poised to surpass the aerospace market with 618,000 tons annual consumption compared to 515,000 tons in 2013 [39]. Figure 2.4 shows chronological development of structural design concepts, materials and processes for composite wind turbine blades.

One of the major drawbacks of extensive usage of composite materials, however, endures in characterization of its multi-axial failure response which has not yet pursued, so far, as per necessity. Composite materials used in wind turbine, aircraft, military, space and many other industries are regularly subjected to combination of loads acting in multiple directions. Owing to their anisotropic nature, composite material strengths obtained using uniaxial testing procedures often do not reflect actual behavior under such realistic, multi-axial loading conditions. This stands as a significant limitation towards wide scale and efficient usage of composite material in engineering structures.

Additionally, the ability to successfully model and simulate the multi-axial behavior of composite materials for optimum use in structural applications largely depends on composite material constitutive relations and associated failure criteria. Proper choice of failure criteria requires rigorous multi-axial experimental characterization that eventually ensures wide applicability of the failure model to adequately describe the material behavior under variety of complex loading conditions. With regards to these facts, only uniaxial test results of composite materials are, however, explored so far to predict the multi-axial failures which appear to be quite insufficient.

A comprehensive failure study under the name of World-Wide Failure Exercise-I (WWFE-I) was performed during 1996-2004 where 19 different composite material failure theories were compared with existing experimental data of tubular specimens under 2D stress state [40-42]. Overall strategy and procedure of the exercise comprised of an explanation of own theory by prominent composite scientists and succeeding application of their corresponding theory to solve 14 challenging test cases set by the organizers. Depending on these case-

studies, 19 different theories were compared where Tsai's empirical quadratic failure criterion and Puck's phonological failure theory (based on Mohr's hypothesis of 'the stresses on the fracture plane are decisive of fracture') appeared to produce relatively better results; still, the most successful theories still can predict the behavior of composites under different loading condition only for maximum 75% cases [42-44]. Later on, same group of researchers initiated two other WWFE exercises; with WWFE-II focusing on validation and benchmarking of composite material failure theories under tri-axial stress state while still ongoing WWFE-III addresses damage/matrix crack development, initiation of matrix-driven delamination and ultimate failure [45]. WWFE-II results are similar to those obtained from WWFE-I, with only very few failure theories maintaining the ability to predict the behavior of composites within $\pm 50\%$ correlation of test data for around 75% of the cases. In addition, it was also understood from the exercises that there is a lack of reliable experimental data that could be compared with these theories. A prioritized future work direction from these exercises can be viewed as more reliable and rigorous formulation of composite failure theory is necessary based on rigorous case studies [42, 45]. It is needless to mention that this reliable formulation of composite failure mapping theories should cover reliable biaxial and multi-axial experimental data from other geometries rather than only tubular shape (e.g. cruciform, dog-bone, iosipescu shape).

Over past decades, worldwide, a few research laboratories have attained experimental biaxial testing capabilities for anisotropic composite material. These can be divided into two major sub-classes – (i) single loading system and (ii) double or even more loading system. Single loading system fixes the biaxial stress ratio on the basis of loading fixture configuration or specimen geometry

[46]. Test methods including hydraulic bulge test, cantilever beam bending, anticlastic bending of flat plates and rhomboidal shapes come out of this category. In multiple loading systems, the magnitude of loading in individual axis determines the biaxial load and, hence, the corresponding stress ratio. Axial loading together with internal/external pressure (tubular specimen), axial loading together with simple torsion (tubular specimen), flat cruciform specimen, round bar under combined torsion and bending fall into this category. Out of this, testing with tubular specimens is very popular as it is capable of maintaining a constant load ratio during testing. However, it has its own shortcomings: composite structures are mostly flat or gently curved rather than of tubular shape and, moreover, the free edge effect is not available in case of tubular shapes. Elastic instability issues, difficulty in fabrication are other major concerns that cannot be neglected while considering tubular specimen [46].

Accordingly, to match with structural shapes more closely, multi-axial loading on a cruciform shape specimen is much more desirable. Cruciform shape specimens are tested under in-plane tension/compression loading along two perpendicular arms [47]. The load is applied by hydraulic or pneumatically controlled cylinders driven by actuator mechanism. One actuator system in each loading direction usually causes side bending of the specimen while two actuator based system provides more reliable test condition with negligible bending effect [47].

Smits et al. [46] have recently developed a cruciform specimen based biaxial testing facility with four servo-hydraulic actuators. This system offers test arrangement in tensile-tensile direction; hence, limiting the experimental capability only in the first quadrant of biaxial stress space. In addition to static

loading, fatigue loading can be applied up to a frequency of 20 Hz. Corresponding research group has proposed an optimal cruciform material based on the loading and fixture constraints [46].

They also tested non-symmetric GFRP cruciform specimen with $[(\pm 45/0)_4/\pm 45]$ layup sequence; made out of flat plates produced by LM Wind Power (formerly LM Glasfiber) through resin transfer molding (RTM) process. These specimens were tested in tension-tension loading for five different biaxial cases until a final failure is achieved. The failure points outside the failure boundary rings showed that existing failure theories are very much conservative in their predictions (see Fig. 2.5) and, hence, the present author understands that more light weight designs still can be ensured in case of tensile-tensile loading [48].

In turn, Welsh and Adams [49] focused on biaxial testing of thickness-tapered cruciform specimens made of symmetric cross-ply $[0/90]_{10s}$ IM6/3501-6 carbon/epoxy laminates. They studied various shape variations of carbon-epoxy specimen to ensure a true biaxial failure scenario. These include arrangements of two wedge grip alignment holes to provide precise alignment of wedge grips, tapering of loading arm width between wedge grip (region where specimen is fixed to testing machine) and gage section of specimen to increase the likelihood of failure in central gage region, increasing of loading arm width in wedge grip to the full dimension of test fixture to lower the uniaxial stress and failure probability in loading arms, evaluation of two corner radius (0.125 in, 0.25 in) between arms to reduce stress concentration, CNC milling in gage section (round milling section, square milling section) to ensure a stress-riser in central section and usage of thicker laminates in loading arms to ensure the failure to occur in biaxially loaded gage section. Finally, they performed biaxial tests of their

optimal specimen. A desirable failure mode was achieved in gage section for all test cases, but the authors expected a higher strength values in all stress ratios and are unsure why it is unexpectedly low. Figure 2.6 presents associated experimental biaxial strength data.

Fig. 2.5: Experimental and theoretical failure envelopes considering linear and elastic behavior of GFRP material [ϵ_x and ϵ_y indicate first ply failure strain in transverse and longitudinal direction, respectively] [47]

Bert et al. [50] have investigated biaxial behavior of glass fibre woven cloth reinforced epoxy laminates with central reduced thickness section in a cruciform specimen. In this study, ultimate biaxial failure loading was achieved to be considerably higher than uniaxial failure loading. Another research group studied the influence of hole, inclusion on quasi-isotropic $[0/\pm 45/90]_s$ SP-286T300 graphite/epoxy plates under equi-biaxial tensile loading [51]. It was observed that non-linear strain variations around the notch boundary related to first failure of fibre plies oriented in radial direction. Maximum failure strain at the notch boundary stood almost twice the uniaxial ultimate strain of the un-notched sample. It further appears that notch sensitivity of the laminate decreases in

tensile-tensile biaxial stress state. In a follow up experiment, same research group performed similar procedures using graphite/epoxy samples with different orientation $[0_2/\pm 45]_s$. The results followed the trend of the earlier experiment where failure strain of the notch boundary is higher than that of the un-notched uniaxial sample [52].

Doong et al. [53] evaluated E-glass/epoxy composite plates with $[0]_{10}$ and $[0/90/0/90/0]_s$ laminates under planar biaxial compressive testing. Stress biaxialities were attained by controlling linear actuator displacement along the specimen edge and compressive strengths are found to be insensitive to load biaxiality. While group of researchers have focused for various forms of biaxial study, still there is a lack of consensus upon the composite material failure behavior within the wider scientific community [54].

Fig. 2.6: Experimental biaxial strength data of $[0/90]_{10s}$ IM6/3501-6 carbon/epoxy laminate [49]

In view of this, a standard working group was formed recently as a consortium between QinetiQ, National Physical Laboratory (NPL) and Free University of Brussels (VUB) to propose internationally acceptable test methods for evaluation of multi-axial strength of fibre reinforced composites [54]. Their multi-axial test facility consists of four hydraulic actuators, each with a static rating of ± 50 kN. The group initially focused for biaxial response of planar cruciform quasi-isotropic composite laminates under tension-tension loading. The specimen possessed a centrally milled gage section with end-tab configuration in all four arms and corner rounding between the adjoining arms. After initial elastic study, a displacement control loading rate of 1 mm/min is applied in the specimen and finally an interlaminar shear failure developed in thickness-reduced taper section [54]. This failure also accompanied a corner rounding failure near two adjoining arms at 28.9 kN load. Overall, this failure did not occur in biaxially loaded region; hence, the consortium decided to modify the geometry without any central milling section and with the provision of wider end tab configuration. But new problem arose in this configuration as failure propagated in adhesively bonded region between the end-tab material and base composite laminate. Thus the group is still looking for a satisfactory solution of the problem in proper cruciform specimen design and has postponed all biaxial tests until true such failure is achieved in desired specimen zone.

It is to add in this junction that the presence of a stress raiser in the form of a hole or crack in the central gage section can initiate all failure to occur in the biaxially loaded central gage section [51-52]. The present work accordingly aims to present notch based biaxial experimentation study based on optimal cruciform shape specimen structure. Chapter 5 will extend these issues in broader arrangements. Research on cruciform specimen carries additional novelty as there is no

published article based on biaxial fracture characterization of composite material with this particular geometry. In addition, most composite designs are plane or gently curved and a cruciform geometry better resembles these shapes in comparison to tubular or other specimen.

2.5 Failure mode map for tubular columns under eccentric loading

In this section, literature review will be covered in line with competitive failure mode mapping of thin shell structure as a representative form of wind turbine tower. Various compressive load cases will be studied with eccentric and non-uniformly distributed loading pattern. In real world, many structures experience non-uniformly distributed loading rather than uniform compressive one. Wind turbine tower is a typical example of such scenario whereby nacelle weight remains at an eccentricity to the tower neutral axis. The eccentric compressive loadings are supposed to shift the failure boundary regions in a typical failure plot which significantly affects relevant optimal design issues in subsequent steps. Moreover, some failure modes under eccentric loading may show completely different pattern in comparison to non-eccentric condition while some other failure modes may not prevail in their entirety. Hence, this is an important research area to focus in case of thin shell structure design.

Within various compressive modes relevant to metal based thin shell structure, common ones include: (i) onset of plasticity or yield failure, (ii) global buckling, (iii) local buckling, (iv) axial crushing, (v) inversion of shell, (vi) splitting of shell, (vii) lateral indentation, (viii) lateral flattening, etc [55]. In case of wind turbine tower, lateral indentation and lateral flattening are the cases unheard as well as unexpected as respective transverse aerodynamic forces actually contribute to

distributed cantilever bending load rather than any above-mentioned kind of point load failure. Additionally, inversion of shell or invertube failure is not possible as there is no die shape arrangement inside the hollow tower. Tube splitting also does not prevail due to similar reason. Hence, plastic flow of material, global buckling, local buckling and axial crushing are dominant compressive failure modes for wind turbine tower; other research group, e.g., Lavassas et al. [56] also analyzes the design of a prototype 1 MW steel tower based on similar compressive loadings. Out of these, plastic failure occurs when the structure can no longer bear any higher loading after its critical yield strength property. Global buckling failure occurs when the tubular column length is higher than a critical value. Though eccentricity condition does not affect the critical global buckling load, developed stress due to eccentric condition is different than non-eccentric uniform or point load condition. Similarly, local buckling failure mode is also affected by eccentricity. Liu et al. [57] has explored various eccentric conditions on local buckling of steel tubular column and found that ultimate failure load has greatly reduced with an increasing load eccentricity ratio. Usually, for a given cross-sectional area, a higher amount of inertia can be obtained with a thin-wall section which reduces its mass eventually. However, if the shell section is too thin, local buckling occurs even before overall column buckling or yielding. Thus, local buckling defines a limit to how much thin a section can be made.

Axial crushing is another prominent compressive failure mode for various ductile thin wall structures. Since the pioneering work of Pugsley and Macaulay [58], Alexander [59], number of researchers and scientists in the last half century focused on failure modes of these structures with corresponding energy dissipation and crashworthiness evaluation. Accordingly, axial crushing is seen

to occur in circular tube [60], square tube [61], tubular ring [62], honeycomb cells [63], multi-corner columns [64], stepped thin-walled tubes [65], corrugated tubes [66] and many others. Thin wall tube, as a widely used structural element, resembles a major sub-class of axial crushing failure-prone structure. Besides, circular cylindrical shell structures provide constant operating force, stable plastic collapse mechanisms and comparatively high energy absorbing capacity; hence, these structures are widely used in energy and vibration absorption systems.

There are various modes of axial crushing failure depending on diameter to thickness ratio (D/t) and length L of the thin walled structure. For a thin cylinder, diamond or non-axisymmetric failure mode dominates; whereas, for thick cylinder, ring or concertina or axisymmetric mode prevails [see Fig. 2.7]. The diameter to thickness ratio limit between thin and thick cylinder varies among the published literature. Pugsley [67] pointed out the ratio at 100, while Tvergaard [68] mentioned about a range of 50-100. Tvergaard also highlighted a link between the structure's diameter to thickness ratio and material's yield strength to modulus of elasticity ratio. According to his findings, concertina mode can dominate in case of even smaller D/t ratio (< 50) if σ_{ys}/E of the tube material is high (in the range of 0.1 or more).

Some researchers, additionally, pointed out a mixed mode failure based on structural parameters [69]. According to their classification,

diamond mode occurs	if $D/t > 80$
ring or concertina mode occurs	if $D/t < 50$ and $L/t < 2$
mixed mode happens	if $D/t < 50$ and $L/t > 2$.

According to Pugsley and Macaulay [58], a general estimate of mean axial load for diamond or non-axisymmetric mode failure behavior can be expressed as:

$$P_{avg} = 6\sigma_{ys}t (10.05t + 0.38D) \quad (2.1)$$

Fig. 2.7: Axial crushing of aluminum 6060-T5 alloy - (a, b) axysymmetric or ring or concertina mode; (c, d) mixed mode [70]

In turn, for ring or concertina mode failure, Alexander [59] presented a rigid-plastic analysis based on bending and stretching energy. His mean crushing load expression finally stands as:

$$P_{avg} = 6\sigma_{ys}t \sqrt{Dt} \quad (2.2)$$

Later on, Jones and Abramowicz [71] proposed a modified version of Alexander's expression which carries good correlation with experimental results:

$$P_{avg} = \sigma_{ys}t \frac{3.44 + 6\sqrt{Dt}}{0.86(1 - 0.66\sqrt{t/D})} \quad (2.3)$$

The above-mentioned failure modes are highlighted in Figs. 2.7-2.8 for Al 6060-T5 aluminum alloy.

Fig. 2.8: Non-axisymmetric or diamond mode axial crushing of Al 6060-T5 [70]

For a typical wind turbine tower, the diameter to thickness ratio varies from 30 to 60 [72]. The ratio agrees well with all tower structure, whether it is made of steel or concrete or being a hybrid design (concrete and steel). Accordingly, tower structures can be considered as thin shell structure. Interestingly, different axial crushing failure studies that are conducted so far, upon diverse structures, only focused on uniformly distributed loading. Hence, there is no published literature on axial crushing in the domain of non-uniformly distributed or eccentrically distributed loading. This study finds this eccentric load case as a potential area of further research. Accordingly, the effect of eccentric loading on various compressive failure modes of thin-wall tubular columns will be explored in Chapter 6. Based on the analysis, it will additionally propose optimal mass based equations.

2.6 Summary

This chapter reviewed works in the areas of life cycle assessment, material selection, characterization of cruciform shape composite material under experimental biaxial loading and competitive failure mapping of thin shell structure under non-uniformly distributed loading. Number of published research articles and reports are critically reviewed in every section with accompanying comments, interesting points, questions and inconsistencies. Novelties of each future problem are critically addressed. Eventually, corresponding research works will be described in next chapters.

Chapter 3

Life cycle assessment of wind farms and corresponding life cycle impact reduction

3.1 Introduction

This chapter focuses on life cycle assessment (LCA) and life cycle impact analysis (LCIA) of three wind farms: one onshore horizontal, one offshore horizontal, another vertical axis farm. All leading life cycle impact and damage categories are evaluated. Per kg life cycle processes are compared, based on which, highest impact contributing ones are sorted out. These materials are replaced with more eco-friendly ones to ensure green designs for future installation. Different wind turbine technologies are also compared each other; both in overall turbine and sub-assembly scale. Finally, all leading power generation technologies are compared with various wind power technologies in terms of life cycle impact values.

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3.2 Life cycle assessment model design

3.2.1 Life cycle assessment goal and scope

The first goal of the study is to evaluate three 50 MW wind farms in terms of their impacts. The study, accordingly, focuses on ten Repower 5M turbine; both for HAWT onshore and offshore farms. With rated output of 5 MW and rotor diameter of 126 meters, it is currently one of the largest wind turbines; and, so far, it is successfully been commissioned in number of onshore and offshore locations. Turbines of same rated power and same technology are selected to allow comparison between onshore and offshore technologies. Though rotor and nacelle components of both turbines are almost same; site-specific differences exist in tower, foundation and balance of system components. Consequently, for vertical application, Sandia on-grid, two-bladed, Darrieus turbine is chosen. Due to lower energy conversion efficiency, dynamic instability problems, fatigue and catastrophic failure issues, there are only few vertical axis wind turbine manufacturers in the world; and, all of them have the commercial designs in a low kilowatt range (<100 kW). Accordingly, getting reliable LCI data for a larger scale vertical axis wind turbine is much more challenging in comparison to horizontal axis machine. However, the present VAWT design has been in smooth operation for a long time and the LCI data of the same is quite accurate as Sandia National Laboratories published a few documentations about this turbine.

Darrieus turbine usually offers lesser weight, lower blade load and simpler foundation than horizontal axis machine [74]. However, how the Sandia machine competes environmentally with the horizontal ones is real focus here. Present 1.6 MW Sandia machine is one of the largest in its category with 45.72 m rotor diameter, 68.58 m tower height and 5.18 m ground clearance [75]. Amount of

electricity generation from all these horizontal and vertical turbines are quantified and it equals to 148.92, 223.38, 109.07 GWh/year with an equivalent capacity factor of 34% (HAWT onshore), 51% (HAWT offshore) and 24.9% (VAWT) for ~8 m/s wind speed at hub height, respectively [75-76]. Capacity factor of a power plant appears as the ratio of its actual output over a period of time to its full nameplate capacity.

3.2.2 Life cycle assessment methodology

Overall LCA study is performed based on ISO standards 14040 and 14044 [6-7]. The outcome of LCA study depends on various factors such as the life cycle processes under consideration, inventory allocation, impact assessment method, evaluation step and functional unit – to name a few. A schematic overview of the life cycle processes is highlighted in Fig. 3.1. To ensure conservativeness in the analysis, neither reuse nor recycling is considered in the study. Also necessity of inventory allocation does not come into the picture as wind farm produces nothing except electricity.

Eco-indicator 99 impact assessment method is used for the study. A tool named ‘impact assessment method selector’, developed by PRé Consultants (an independent LCA organization and world pioneer in LCA research and corresponding software development), has compared all leading impact assessment methods in terms of various criteria; viz., single scores, weighting set, time perspective, geographic coverage, geographic acceptance, simplicity versus scientific quality and completeness. Eco-indicator 99 impact assessment methodology tops that list followed by other impact assessment methods, such as, GWP, EDIP, Impact 2002+, Energy, CML92, CML2002, EI95 and EPS method

[77]. Accordingly, many scholarly articles are published based on *Eco-indicator 99* method.

Based on model uncertainties, Eco-indicator 99 performs LCA evaluations by three different ways. Out of these, egalitarian perspective includes all-inclusive calculation of life cycle impacts; even if the impact is lightly proven to the scientific community or only affecting in a rare longer term in comparison to the project life time. Individualist perspective works on the other end; i.e., it deals with only short termed and very robustly proven facts. Conversely, hierarchist perspective deals in the middle of two extremities; i.e., it covers all life cycle effects/impacts about which there is a scientific consensus in a moderate life time perspective. This LCA study focuses on hierarchist calculation methodology to establish the calculation on stronger scientific consensus as well as not to avoid even moderate life time impacts.

ISO Standards of LCA processes outline 5 key stages in life cycle impact calculation which are as follows [6-7]:

- Characterization
- Damage assessment
- Normalization
- Weighting
- Single score

Out of this, characterization is the obligatory level of impact assessment. Only a small bunch of LCA processes calculate impact in all 5 stages; these include Eco-indicator 99, EPS 2000 and ReCiPe method. In present Eco-indicator 99 based LCA calculation method, every life cycle substance that contributes to an impact

category is multiplied with a characterization/equivalence factor which represents the relative contribution of corresponding substance; e.g., the characterization factor of CO₂ in the impact category 'climate change' equals to 1, whereas the characterization factor of CH₄ in same impact category stands at 21 - this indicates the release of one kg CH₄ causing the same amount of climate change as that of 21 kg CO₂.

In damage assessment stage, impact categories with common units are classified and added together to provide a uniform value; the impact categories of Eco-indicator 99 include human health, ecosystem quality and resources. 'Human health' is measured by weighted average disability caused by different diseases in companion unit of 'disability adjusted life years' (DALY). Here, a damage value of 1 DALY indicates the decrease of one year life for one average individual [78]. 'Ecosystem quality' is evaluated by 'potentially disappeared fraction' (PDF) of plant species with the unit PDF*m²*yr; here, a damage of 1 PDF*m²*yr means all species will disappear from 1 m² area for 1 year. Whereas, damage category 'resources' is measured by 'surplus energy' in MJ/kg extracted material (this MJ/kg unit indicates the expected increase of extraction energy per kg extracted material). Here a damage value of 1 MJ/kg represents an additional MJ/kg of energy will be necessary in further extraction of the same material due to lower remaining reserve [78].

Human health damage category gives a combined effect of various impact categories; viz., climate change, ozone layer depletion, carcinogenic substance, organic and inorganic respiratory effects and ionizing radiation. Damage category 'ecosystem quality' measures the change in ecosystem due to base technology with adding up the life cycle impact points of land use,

acidification/eutrophication and ecotoxicity whereas, 'resources' damage category covers fossil fuel and mineral depletion impact category.

After characterization, all scores are presented on a relative scale with a percentage form ranging from 0 to 100%. This representation provides good insight for comparison of different substances. There is however possibility to misinterpret the result as a substance displaying 100% may refer to 100% of very high or low figure. In addition, different impact category indicators usually possess different units – so 100% of any one is in no way comparable to another. Because all impact categories do not possess equal importance; a better alternative is to introduce a common uniform unit; this is what Eco-indicator 99 method carries on subsequently by dividing the characterization results with a reference value per impact categories. This reference value equals to life cycle impacts caused by one average European person over a time period of one year.

Fig. 3.1: LCA processes of wind farms

Normalized values, evaluated thus, show which effects are large or small in relative terms but say nothing about their relative importance; e.g., how bad global warming is compared to resource depletion or so on. Hence, in consequent weight based analysis, specific weighting factor is applied to each of the normalization values. This represents the results in a scale of severity. In last stage, Eco-indicator 99 provides total single score by adding up all weighted scores for each LCA process.

Rated power generation (50 MW) and unit electricity generation (1 kWh) are used as functional unit for inter-farm comparisons; while for process scale life cycle, 1 kg mass unit is used as functional unit (FU). Industry standard wind turbine life-time of 20 years is assumed for all farms.

3.2.3 Life cycle inventory of HAWT onshore and offshore farms

Inventory data relating to HAWT onshore and offshore turbines are mostly agglomerated from Repower website and other supplier sources. These include online brochure of the wind turbine from its manufacturer (Repower) [79], press releases from the manufacturer [80-81], official seminar presentation by manufacturer [82], conference proceedings from the manufacturer in two international conferences of wind energy [83-84], separate technical reports on Repower 5M by National Renewable Energy Laboratory (NREL) [85], Energy Research Centre on Netherlands (ECN) [86] and Institute for Energy, University of Bochum, Germany [76]. A few other data regarding materials, resources and consumables are collected from prevalent statistical sources and manufacturing datasheets [87-89]. Relevant life cycle inventory (LCI) databases are used for process definition. Few assumptions are made in conjunction that seems to be

consistent with overall farm scale. Some up-scalings are opted based on previous published researches [90]. It is, nevertheless, difficult to gather all necessary suppliers' information about full particulars of the turbines. Hence, the focus is implied here to collect data, at least, for all well-known components.

Repower 5M total blade weight stands at almost 54 ton (see Table 3.1). Each blade is 61.5 m long and weighs 17.74 ton and is made of glass fiber reinforced epoxy prepreg composite material [85]. It is assumed to contain 70% glass fibre and 30% epoxy resin by weight. Due to necessary cut-off during pre-pregging, actual fiber weight will be more. Hence, fiber weight is scaled up by 10%.

Next, Repower 5M possesses a 18 m long, 6 m wide, 6 m high nacelle consisting covering, bed frame, gear, generator, transformer, converter, yaw system, holding brake, onboard crane, corrosion, lighting protection system, etc. [80]. Including all components, total nacelle weight stands at 315 ton [82]. In addition to manufacturing of these individual components, final finishing e.g. welding, metal cutting processes and subsequent assembly are accomplished at turbine erection stages. The following description lists data on most significant nacelle components.

Turbine main shaft and bed frame are dominantly made of steel and cast iron whereas nacelle cover is manufactured of glass fiber reinforced epoxy composite material. Accordingly, glass fiber inventory and emission data is estimated from glass fiber, at plant,ecoinvent process. It includes all consumption and emission data from overall glass fiber manufacturing process chain including transportation. Similarly, epoxy resin, liquid, at plant/RER S ecoinvent process is selected for epoxy material [91].

HAWT gearbox set is predominantly made of steel and iron. Steel low alloyed, at plant/RER Secoinvent process is selected for steel material. For cast iron, cast iron, at plant/RER S process is chosen. This ecoinvent process includes entire cast iron production phases ranging from transport of raw material and other inputs to electric arc furnace, to melting, refining and casting. Energy consumption and emission data is allocated according to average EU based cast iron plant datasets.

Generator and transformer are modeled according to environmental product declaration (EPD) data of the manufacturer. Generator mainly consists of steel and copper with minor percentage of other materials [87]. For steel, same ecoinvent process has been chosen, as previous. Copper data is estimated based on ecoinvent process copper, at regional storage/RER S. Transformer primarily consists of steel and copper and some percentage of aluminum, porcelain [88]. Aluminum, production mix, at plant process is chosen for aluminum. It includes overall primary and secondary aluminum manufacturing processes according to worldwide production mix.

Tower is mainly manufactured of steel. Ecoinvent system process steel low-alloyed, at plant/RER S which spans entire steel manufacturing cycle is selected for steel material. Corresponding raw material consumption and waste, emission data are analyzed and included in life cycle inventory (LCI) subsequently for all processes. Emission analysis and LCI data are all based on latest June 2010 ecoinvent system process update. The material system process is based on average world production and European consumption mix based on EU steel plants. Alkyd paint, white, 60% in H₂O, at plant/RER S material is selected for

painting purpose. This alkyd dataset is based on one specific long oil alkyd commonly used in architectural paints of white color. Relevant material processes comprise overall manufacturing route of water-reducible alkyd resin production. Paint production is done by physical mixing of resin, water and pigments. All raw material consumption and emission analysis processes are based on latest June 2010 ecoinvent material process update.

Onshore turbines are erected on concrete foundations [82]. Foundation size and its material allocation entirely depend on geo-technical condition of the plant location. Likewise Elbehafen port Repower 5M installation, the foundation of the wind farm is assumed to consist of 1300 m³ concrete and 180 ton steel [81].

HAWT offshore wind plant, in turn, is installed in 30 m deep sea-water located around 30 km from shore. Unlike the onshore foundation, jacket type sub-structure together with legs, bracings and piles is used to support the offshore tower. Eight piles, each one with 50 m length and 1.07 m diameter, together combine to 315 ton. Jacket frame, sleeves and transition node weigh around 425 ton whereas other secondary structures like boat landings, J-tubes, anodes weigh 85 ton. Together, the full structure weighs 825 ton. In case of tripod structure, the total structural weight is supposed to become significantly higher, as mentioned in Repower study [83-84].

During construction, piles can be adapted to wide range of soil conditions. A transition piece exists between foundation and tower lessening the diameter of the foundation to the tower. Two boat landings are mounted on the jacket structure for easy maintenance purpose [83]. Turbines are mutually connected by cable lines, which are connected on high voltage (HV) transformer platform.

Cathodic protection system is opted for rust prevention while aluminum is allocated as active anode material. Aluminum electrolysis process continues through-out the plant life time. In this connection, aluminum electrolysis/RER S ecoinvent system process is selected for anode that includes anode production in its plant, transport of materials and disposal of the waste based on European process industries. Usually, paint repair and renewal activities become essential by every 10 years. Paint repair and anode renewal quantity are up-scaled, accordingly, from turbine erection time data.

Onshore wind farm needs storage area and proper extension of existing roads and highways as these farms are usually installed in remote places. Road connection between the farm and the city and within the farm is also necessary for maintenance purpose. Road works need gravel, HDPE, and polypropylene; all these materials are modeled based on equivalent ecoinvent process [90]. Onshore turbines are connected to the existing national distribution grid by a 10/60 kV transformer station; this transformer station is not included in the study. Total, there is 10 km of 10 kV cables for connecting 10 turbines to the existing transformer station. These cable lines carry substantial weight. Relevant data regarding the manufacturing of cables is up-scaled from a previous study on Italian wind turbine [90].

In offshore plant, a high voltage transformer is included to increase the voltage of transmission electricity. The transformer is set on a platform made of pile and jacket sub-structure close to the wind turbines. Corresponding jacket legs are interconnected by lattice girders and bracings. Each of the 6 piles is 50 m in length and 1 m in diameter and, as well, total they weigh 235 ton. In addition, there is a boat landing platform made of steel for maintenance purpose. Together

with jacket frame and boat platform, the entire transformer platform weighs 575 ton [83]. Transformer platform is set about 12 m above mean water level. On top of the transformer platform, a steel structure is assembled to house the high voltage transformer. High voltage transformer mainly includes steel, copper, transformer oil and some little fraction of other materials. Transformer oil also carries a significant weight. LCA databases contain data on all these materials. Steel structure is assembled onshore and transported to the offshore wind power plant as a single module. The module is placed on the transformer platform by means of a floating crane. The high voltage transformer weight is collected from manufacturer's specification [88].

The jacket structures are furnished with cathodic protection in order to prevent corrosion while aluminum is allocated as active anode material. This protection implies that aluminum is consumed throughout the lifetime. Other than this, 34 km long cable is considered in offshore LCA study, which is modeled based on the data provided by Norwegian input-output table [89].

Transportation to site includes transportation by light goods vehicle, lorry, rail freight and sea freight by barge tanker. A 'tkm' unit is used for transportation which indicates transportation of 1 ton goods over 1 km. Erection of wind farm includes erection of foundation, tower, nacelle, blade and other transmission, distribution components for all onshore and offshore turbines in the plant; this includes craning and other construction work at site. Turbine erection and foundation construction activities mainly require fuel (diesel) and medium voltage electricity. Welding, metal cutting activities also need to be performed in this stage.

In turn, operation and maintenance activities include change of lubrication oil and transport to and from the turbines. Turbine rotating parts will usually go through high level of wear and tear. Consequently, change of oil and lubrication of gearbox, generator, etc. are accounted in the study. Paint repair and renewal of active anodes to cathodic protection will be carried out at transformer station in 10 year frequency.

Onshore transport is carried out by passenger car and light goods vehicle, while, at sea, barge tanker and boats are used. A technician must carry out inspection of turbines and cables four times a year. Hence, 22000 personkm transport has been included by passenger car in the project model. 2000 tkm transportation by 3.5-16 ton lorry also has been allocated for lubrication oil and other maintenance freightage [9, 90].

It is estimated that offshore wind farm will be inspected eight times in a year. Sometimes it will be carried out by boat, sometimes by barge. As there is no ecoinvent process based on boat transport, transportation by barge has been assigned for all sea passage. Regular inspection of the cables of the offshore wind farm is not included in the study and is assumed to be insignificant. Dismantling and land-filling activities includes craneage after dismantling and transportation of dismantled materials from erection place to the final disposal site (by vessel at sea and by lorry and light goods vehicle in onshore).

3.2.4 Life cycle inventory of VAWT farm

For vertical axis machine, all key material and process data are congregated from corresponding documentations of Sandia National Laboratories [74-75]. VAWT turbine is built on two aluminum extruded, uniform thickness blade that are rigidly attached to a central tower. Contrary to HAWT, Darrieus turbine does not have a well-defined nacelle. Nevertheless, all of the major power train components, e.g., shaft, gear, generator, transformer and disc brake system are used in VAWT. The Sandia machine possesses one vertical, low speed and another horizontal, high speed shaft through an enclosed gearbox. The power train components are predominantly made on steel with a small percentage of brass, synthetic rubber, etc. Generator and transformer are modeled according to environmental product declaration (EPD) data of the manufacturer. Generator mainly consists of steel and copper while transformer primarily consists of steel, copper, aluminum and porcelain [87-88]. Relevantecoinvent processes are considered for these materials.

A large diameter, thin-wall steel tube tower transmits torque from blades to the revolving shaft. The full structure is supported by three guy cables attached to the top of the VAWT and by anchorage at ground level. Cable bridge strand manufactured of steel material is chosen for guy cable to maintain structural rigidity. Ancillary cable data is assumed likewise HAWT study. Relevant roads and establishment works are ensured as well.

During operation phase, lubrication oil is necessary to reduce wear and tear and for smooth, tranquil operation of machine parts. The components are lubricated, accordingly, once in every 6 months. Additionally, guy-cable tension and lubrication leaks need to be inspected twice in a year. Other life cycle processes, e.g., transportation by lorry, rail and sea-freight, consumption of medium

voltage electricity, steel arc welding, burning of diesel fuel during erection phase activities are modeled likewise onshore farm.

Overall life cycle inventories of 50 MW horizontal axis onshore, offshore and vertical axis wind farms are highlighted in Table 3.1 based on afore-mentioned turbines.

Table 3.1
Life cycle inventory of wind farms [74-77, 79-91]

Sub-assembly and process	Material with corresponding process ¹	Unit	HAWT onshore	HAWT offshore	VAWT
Rotor	Steel	ton	710	710	
	Glass fiber	ton	420	420	
	Epoxy	ton	160	160	
	Aluminum	ton			860
Nacelle (In case of VAWT it includes drive and power train components)	Steel	ton	2050	2050	1030
	Lubricating oil	ton	630	630	630
	Cast iron	ton	470	470	
	Copper	ton	190	190	
	Aluminum	ton	105	105	
	Lead	ton	95	95	
	Epoxy resin	ton	60	60	
	PVC	ton	47	47	
	PET	ton	40	40	
	Glass fiber	ton	32	32	
	ABS	ton	10	10	
	Ceramic tiles	ton	6.5	6.5	
	Silica sand	ton	3	3	
	Brass	ton			1.5
	Rubber	ton			0.7

Tower	Steel	ton	3450	2000	1480
	Alkyd paint	ton	20	20	
	Aluminum	ton		3.75	
Foundation	Concrete	m3	13000		
	Steel	ton	1800	8220	660
	Alkyd paint	ton		23	
	Aluminum	ton		7	
Ancillary	Aluminum	ton	37	44	37
	Brick	ton	14		14
	Cement	ton	10		10
	Copper	ton	12	33	12
	Gravel	ton	22		22
	HDPE	ton	5		5
	Polybutadiene	ton	24	19	24
	PVC	ton	28	29	28
	Steel	ton	2	830	2
	Alkyd paint	ton		6	
	Glass fiber	ton		1.2	
	Transformer oil	ton		48	
	Kraft paper	ton		1.5	
Additional life cycle processes	Diesel	MJ	18050	22050	18050
	Electricity, medium voltage	MWh	1700	2650	2300
	Transport, barge tanker	tkm ¹	30000	146000	30000

Transport, freight, rail	tkm	5000	5000	5000
Transport, lorry 16-32t	tkm	8200	8200	8200
Transport, lorry 3.5-16t	tkm	2000		2000
Transport, passenger car	personkm	22000	1000	22000
Welding, arc, steel	km	0.235	0.395	0.112
Wire drawing, copper	kg	13000	93000	
Tap water	l	1000000	1000000	1000000

¹ Material corresponding process includes all cradle to grave processes of the individual material starting from raw material extraction to material processing in primary, secondary and tertiary phases and then operation, maintenance life of the product and final end of life process.

² tkm indicates tonne.km

3.3 Results and discussion

3.3.1 Life cycle inventory results of three wind farms

Life cycle inventory (LCI) includes all inventory flows from and to the nature for individual wind farm. Major life cycle inventories (LCI) of three wind farms are calculated here based on 'single score' indicator method with a functional unit (FU) of 50 MW. As seen in Table 3.2, 'single score' based weighted LCI results are compartmentalized according to 1% cut-off point which delivers three categories of emission and consumption out of six; namely, emission to air, emission to water and raw material consumption. The emission to soil or other non-material emissions are insignificant; and, hence, are excluded due to 1% cut-off criteria.

3.3.2 Comparison between HAWT onshore and offshore farms

It is very important to determine how the impact values of tower, foundation and ancillary components differ between HAWT onshore and offshore farms. Accordingly, the LCA results of this section highlight the same. As evident in Fig. 3.2, the foundation and ancillary ecopoint values of HAWT offshore are much higher than HAWT onshore. Here, 1 Pt ecopoint indicates total LCA impact caused by 1 average European person over the time period of one year. Total, 3 damage and 11 individual impact eco-profiles are sorted sequentially from left to right in 'damage and impact categories' axis of Fig. 3.2. Within the damage categories, human health (HH) maintains the highest kilo ecopoint (kPt); whereas, within individual impact categories, respiratory inorganics (RI) emission comes up with the highest impact values in most sub-assemblies. RI emission basically happens due to the presence of steel and cast iron.

Table 3.2: Life cycle inventory of three wind farms (single score indicator method, FU = 50 MW, 1% cut-off point)

Substance	Emission category	Unit	HAWT onshore	HAWT offshore	VAWT
Total		Pt	3649138	4937099	1568893
Remaining substances		Pt	412918.2	536692.8	158847.7
Particulates, < 2.5 um	Air	Pt	334360.6	511728.8	165033.8
Particulates, > 2.5 um, and < 10um	Air	Pt	295711.1	465332.6	142094.9
Chromium	Air	Pt	178703.6	305651.8	71623.13
Nitrogen oxides	Air	Pt	164558.3	212552	77647.96
Carbon dioxide, fossil	Air	Pt	120311.8	154527.7	67480.09
Sulfur dioxide	Air	Pt	117754.4	155698.6	66680.08
Cadmium	Air	Pt	69553.11	77791.19	7983.637
Zinc	Air	Pt	47746.49	71421.02	19459.12
Oil, crude, in ground	Raw	Pt	322624.2	364377.3	292833.5
Gas, natural, in ground	Raw	Pt	278358.2	370187.5	122233.4
Nickel, 1.98% in silicates, 1.04% in crude ore	Raw	Pt	233583.6	400059.9	93891.38
Transformation, to arable, non-irrigated	Raw	Pt	59339.38	131387.1	6170.156
Transformation, from arable, non-irrigated	Raw	Pt	-59291	-131280	-6165.17
Arsenic, ion	Water	Pt	589192.8	730506.2	171796.6
Cadmium, ion	Water	Pt	392723.1	460275.5	79674.07
Copper, ion	Water	Pt	49124.35	53638.63	13785.27
Nickel, ion	Water	Pt	41866.08	66550.67	17823.86

Fig. 3.2: LCA assessment of tower, ancillary, foundation sub-assemblies (FU = 50 MW)

The remaining processes are also important contributors. Impact category respiratory inorganics (RI) is an important constituent of damage category human health (HH) which causes it to be of higher value. The other impact categories with their base material and process contributors are discussed in detail in Section 3.4. Other major impact contributors include eco-toxicity (E), minerals (M), fossil fuels (FF), carcinogens (C) and climate change (CC), etc. Here, 1 kPt eco-impact value indicates the LCA impact caused by 1000 average European persons over the time period of one year.

As seen from Table 3.1, an onshore tower needs almost 75% more steel than its offshore counterpart. Quantities of other materials do not vary significantly. Hence, an onshore tower carries 70% more kPt in comparison to offshore one. In turn, offshore foundation comes up with 1667.78 kPt which is around 3.7 times that of onshore foundation. The reason behind this lies in material information where an onshore foundation uses only 22% steel of its offshore complement. Other than these, great difference of kPt values lies in ancillary sub-assemblies. Ancillary components include parts and items aside from turbine structure that are necessary to distribute the generated power to the national grid. It is evident from Fig. 3.2 that HAWT offshore ancillaries exhibit around four times more kPt than HAWT onshore. Reasoning behind this again lies in additional inventory of high voltage transformer and its platform.

Next, life cycle damage comparisons of all sub-assemblies of HAWT onshore and offshore are featured in Fig. 3.3. Nacelle and rotor of both turbines exhibit almost equal eco-profiles as there is no significant diversity in inventory between these sub-assemblies. Nacelle maintains the highest kPt out of all and, hence, carries the most deteriorating impacts, largely surpassing other sub-assemblies by distant difference. Next, large impact carrying sub-assemblies are: offshore foundation, onshore tower, onshore foundation and offshore tower, sequentially. As evident from the figure, a large impact carrying foundation comes with a low impact carrying tower and vice versa. Conversely, rotor ecopoint is less for both turbines due to lower material consumption. Overall, the HAWT onshore and offshore farm emit 3649, 4937 kPts of impact based on 'single score' definition.

Fig. 3.3: LCA assessment of all HAWT onshore and offshore sub-assemblies (FU = 50 MW)

3.3.3 Comparison between HAWT onshore, HAWT offshore and VAWT farms

This section demonstrates the compartmentalized life cycle damage and individual impact comparisons between HAWT and VAWT farms based on unit electricity generation (FU = 1 kWh). As evident from Fig. 3.4, in all damage and most of the impact categories (except FF, R, RO, OL), the VAWT farm appears to be environmentally superior to the HAWT ones with lower ecopoint (Pt) representation; followed by the HAWT offshore and HAWT onshore farm. Eco-profile of each unit electricity generation stands at 7.19×10^{-4} ecopoint (Pt) in VAWT farm while for HAWT offshore and onshore, it equals 0.001105 and 0.001225 Pt, respectively. Overall farm scale ecopoint values stand at 1569.01,

3649.14 & 4937.10 kPt for VAWT, HAWT onshore and HAWT offshore farm; respectively.

Fig. 3.4: LCA impact and damage assessment of wind farms (FU = 1 kWh)

Figure 3.4, in addition, exhibits onshore and offshore farm HH damage values at 1.72 and 1.54 times of VAWT farm. Similarly, EQ onshore and offshore values are 1.27, 1.14 times of VAWT. Similarly, VAWT Farm also has the lowest Re value. These results are major findings for VAWT technology as these establish it as

being the most environment friendly wind turbine technology within all categories. VAWT technology, in addition, offers lesser weight, simpler foundation and less maintenance cost. Besides, as Paraschivoiu [92] mentioned, its power co-efficient can be greatly enhanced by introducing drag reducing airfoils. With all these merits coupled with a low imperative ecopoint, vertical axis turbine can be an alternative green electricity source of commercial scale in coming future. Lastly, within HAWT farms, HAWT offshore technology proves its dominance over onshore HAWT based on both life cycle eco-imprint and energy intensity perspective. This is mainly because of higher capacity factor of offshore farm. As it is mentioned in Section 3.2.1, the capacity factor of HAWT offshore farm is 50% more than its onshore counterpart. However, if the capacity factor stands the same for both farms, the offshore farm will have much more impact than onshore farm for per unit electricity generation (kWh).

3.3.4 New material and process with low LCA ecopoint

As mentioned in previous sections, the present LCA study, at its very core, endeavors on proposing new materials and process routes that will eventually diminish life cycle eco-impact values without compromising the structural needs and functional requirements. Hence, the materials that are used in larger quantity in wind farms are sorted out. A cut-off criterion of the materials with minimum 1% consumption (by weight) is used for this purpose to explore the most impact and damage contributing ones. Accordingly, the materials are sequentially arranged in terms of their life cycle impacts, as shown in Fig. 3.5. 'Copper, at regional storage'ecoinvent process which comprises all life cycle processes starting from material extraction to transportation to regional copper

storage exposes most adverse effect to the surroundings with distant difference amongst all the materials (see Fig. 3.5).

Fig. 3.5: Life cycle damage assessment of ecoinvent process (FU = 1 kg)

As seen from process based LCA analysis, total impact, HH and EQ damage assessment value of 1 kg copper process attend to be 1.82, 2.94 and 3.19 times of combined respective impact and damage assessment values of all other materials which are consumed by the quantity of at least 1% (of whole weight) in these farms. Steel is the most dominantly used material in all three farms. Nevertheless, in comparison to steel, copper causes 27.45 times total life cycle impact. Next, environmental impact intensive materials are epoxy resin, aluminum and lead, sequentially; with impact values of 3.28, 3.04 and 1.92 times that of steel. Glass fiber and steel material exhibit comparable eco-indicator points. Cast iron is better than GFRP and steel and, obviously, concrete is the best

of all materials under study, as shown in Fig. 3.5. Concrete is 43 times and 1979 times life cycle impact-savvy as compared to cast iron and copper, respectively.

Fig. 3.6: Wind farm LCA processes by life cycle impact (FU = 50 MW, cut-off point = 1%)

In the consequence of per kg process analysis, companion overall farm scale impact findings are shown in Fig. 3.6 which again manifests the substantial effect of copper, though it is used at much lower quantity in all three farms (see Table 3.1). Comparatively, ‘concrete, normal, at plant’ process leaves very negligible impact in HAWT onshore farm.

To reduce total wind farm impact, it is of utmost importance to replace copper before all else. As seen in Table 3.1, HAWT onshore and offshore farms experience usage of copper in cable, generator and transformer; whereas in VAWT farm, it is only used in cable. In generator and transformer, copper is

fundamentally used as winding material. In all these components, copper is chosen due to its high electrical conductivity and low price. Better substitutes of copper can be explored from appropriate material selection database [93]. For this, a limit based search criterion is implemented in CES, as follows:

$$\rho_{elr} \leq 3 * 10^{-8} \Omega.m \quad (3.1)$$

$$C_m \leq 5 \text{ USD/kg} \quad (3.2)$$

where ρ_{elr} and C_m indicate electrical resistivity and per unit material cost, respectively. Based on limiting criteria of Eqs. (3.1) and (3.2), Fig. 3.7 highlights various competitive materials with their fatigue strength and electrical resistivity properties.

Aluminum alloy grade T7451 with its 217 MPa fatigue strength and only 10.99% ecopoint contribution of that of copper appears as a strong candidate for replacement of the later; however, it carries electrical conductivity of $2.8 * 10^7$ S/m, which is 36% less than usual Cu cables (C101XX grade). Hence, cross-sectional area of Al conductor needs to increase to attain same current carrying capability. As wind turbine transmission cable, and windings of transformer, generator has no concern with increasing volume, Al alloy T7451 can be conveniently applied in these applications. In addition, density of Al alloy T7451 is one-third of copper and fracture toughness stands at $27.7 \text{ MPa}\sqrt{m}$ which is well above (54% higher) the standard design guideline value of $18 \text{ MPa}\sqrt{m}$ [93]. Hence, in addition to almost 9 times LCA impact minimization and superior mechanical properties, simultaneous mass and cost minimization can be achieved by replacing the Cu alloy with particular Al grade. Its other notable characteristics include its non-flammability, significant draw-ability and,

excellent behavior under exposure to sun, fresh water or ocean environment which are lucrative from electrical installation point of view.

Other replacements can be ensured with introduction of low LCA impact based monolithic and sandwich composite blade design. Particularly, enhanced usage of sandwich material coupled with low LCA imprint based natural core section can improve the blade design from both structural and life cycle perspective. But, because the LCA databases are still in their infancy with regards to impact categorization of diverse types of sandwich and composite materials, this substitution remains out of the scope.

Next, a concrete tower design comparing to a presently used steel tower can greatly decrease the life cycle imprint in the expense of structural robustness.

Fig. 3.7: Selection of substitute material for copper [violet and red points indicate Al and Cu material, respectively]

Figure 3.8 demonstrates the life cycle impact reduction in all farms with the introduction of aluminum in replacement of copper which ensures almost 30%, 25% and 7% impact reduction in HAWT onshore modified, HAWT offshore modified and VAWT modified design, respectively; in comparison to their corresponding counterparts. This is to mention that in current decades when, worldwide, scientists are closely examining the ways to ensure an environment-friendly design with more sustainability and lower eco-profiles, the reduction of LCA impact values in all sorts of future wind farms through newer generation green turbines will lead the way to the next dimension.

Fig. 3.8: LCA Impact comparison between existing and modified design (FU = 1 kWh)

3.3.5 LCA process contribution in overall farm scale

At this stage, the study focuses on overall farm scale impact contributing processes that carry individual process based LCA studies in both existing and modified designs. As featured in Fig. 3.8, impact categories like R, RI, OL, A/E and LU turn out with minor ecopoints in all farms; hence, only major adverse effect carrying impact categories are followed up in this analysis. Fig. 3.9, accordingly, represents life cycle processes that intensely contribute to carcinogenic emission. Carcinogenic substances are some cancer triggering emissions that include dioxin, benzene, kepone, asbestos, amongst many others. From Fig. 3.9 it is vivid that 'copper, at regional storage' ecoinvent process contributes the maximum carcinogenic impacts in the existing design, followed by 'steel, low-alloyed, at plant' process. Total carcinogenic impact arises to be almost one third of all onshore and offshore wind farm eco-impacts; whereas, in modified design, its contribution reduces to around one-seventh in both onshore and offshore farms. Same phenomenon continues in VAWT. However, as copper amount is significantly lower in VAWT, steel leads the value in current design.

Fig. 3.9: LCA processes by carcinogen impact contribution (FU = 50 MW)

Wind farms additionally emit significant RI substances. HAWT offshore issues 1302 kPts while onshore and VAWT farms emit 879, 434 kPts, respectively. Within the processes, steel contributes most followed by copper, epoxy resin and cast iron in existing designs. In offshore farm, remaining processes, altogether, exude large ecopoints; but, on individual level, they do not contribute 3% impact which is the cut-off point for this particular analysis. Additionally, VAWT farm does not witness any consumption of epoxy resin or cast iron. Hence, equivalent ecopoint columns demonstrate zero kPt contribution (see Fig. 3.10). Comparatively, in modified designs, RI emission diminishes by around 10% in all plants due to lower copper usage.

Fig. 3.10: LCA processes by respiratory inorganics impact contribution (FU = 50 MW)

Fig. 3.11: LCA processes by climate change impact contribution (FU = 50 MW)

Atmospheric green house gases (GHG), e.g., water vapors, CO₂, CH₄, NO_x and ozone layer absorb some part of solar thermal radiation that increases the global

temperature and, ultimately, affects to climate change effect. As shown in Fig. 3.11, steel and aluminum based processes outperform the rests in all farms in climate change category. Other contributing processes, rank wise, include concrete, epoxy resin, glass fibre, disposal activity, etc. Copper does not contribute to climate change. Hence, replacement of copper does not bring any change in wind farm's impact on climate change.

Fig. 3.12: LCA processes by ecotoxicity impact contribution (FU = 50 MW)

Next, in ecotoxicity emission category, steel and copper processes deliver major contributions. Total ecotoxicity emission of existing HAWT onshore, HAWT offshore and VAWT farm is equivalent to 471, 691, 162 kPts, respectively, as seen from Fig. 3.12. Higher ecotoxicity emission means the wind farms possess substantial potentiality to destabilize the natural ecosystem, bio-diversity and bio-chemistry of living organisms. Replacing the copper materials hereby

reduces the ecotoxicity imprint by almost 25% in onshore and offshore farm and 5% in VAWT farm.

Fig. 3.13: LCA processes by minerals impact contribution (FU = 50 MW)

Wind farms also significantly contribute in mineral and fossil fuel depletion categories. ‘Steel, low alloyed’ process bears the greatest impact in mineral depletion category followed by ‘copper, at regional storage’. Other processes are not as significant as these ones. Within the three farms, VAWT has the lowest contribution with total 157.92 kPts, as seen from ecopoint columns of Fig. 3.13. In turn, in fossil fuel impact category, aluminum and lubricating oil also emits major ecopoints other than steel process, as shown in Fig. 3.14. Next, distinguishing contribution comes from ‘epoxy resin, liquid, at plant process’ (80 kPt). Replacement of copper does not improve fossil fuel impact values; but mineral impacts reduce by around 30% in both onshore and offshore farm and almost 5% in VAWT farm.

Fig. 3.14: LCA processes by fossil fuels impact contribution (FU = 50 MW)

3.3.6 Life cycle impact of major power generation technologies

Finally, the life cycle impacts of all major power generation technologies are compared in Table 3.3. Ecoinvent per kWh power generation process data are used for all technologies except wind power technologies [91]. Modified wind turbine plants are included in the analysis in addition to the existing ones. Rank wise plant LCA impact sequences demonstrate that fossil fuel based technologies can never compete with wind turbine and other renewable energy technologies. Out of fossil fuel technologies, ecoinvent oil based plant process emits worst LCA impact followed by diesel, natural gas and coal based plant. Photovoltaic (PV), biomass and nuclear plants are sequentially better than fossil fuel technologies. The best performance is, however, demonstrated by hydro-kinetic power plants. Wind farms compete better with all plants except the hydro-kinetic ones. And, out of all wind farms, VAWT farm holds more promises chased by

offshore HAWT and onshore HAWT farm. In kilo ecopoint (kPt) basis, VAWT farm issues 1.55 times impact of hydroelectric one which may even further reduce with an offshore VAWT technology due to its higher capacity factor, as experienced in offshore locations. Overall per unit electricity generation eco-profile is presented in Table 3.3.

Total ecopoints of all these farms are given in first column of Table 3.3 where every plant is sequentially arranged, with all values within the column indicating that much times of the lowest value of that column. Thus an oil plant contributes almost 154 times eco-impact of the hydrokinetic plant to generate every 1 kWh of electricity. Within all plants, highest carcinogenic effect comes out of PV plant, highest respiratory organics emission effect comes out of biomass plant, highest respiratory inorganics emission effect comes out of coal plant, highest climate change effect is caused by diesel plant, highest radiation effect is generated by nuclear plant; highest land usage, ozone layer depletion and acidification/eutrophication effects are caused by oil plant while highest ecotoxicity emission and fossil fuel effect are contributed from diesel plant; amongst many other findings.

Within the two most leading renewable technologies (solar, onshore wind turbine), solar farm exhibits 5 times more impact of onshore wind farm which rises to almost 8.7 times in comparison to onshore VAWT farm for every kWh of electricity generation. In view of this, wind power is cleaner, greener and more environment-friendly than solar PV technology. Accordingly, wind power installation should be prioritized to combat various scenarios and environmental issues like fossil fuel depletion, global warming, climate change and ozone layer depletion.

Table 3.3

Life cycle impact assessment of major power generation technologies (FU = 1 kWh) [91]

Impact category	Total	C	RO	RI	CC	R	OL	E	A/E	LU	M	FF
Oil	153.74	364.77	155.02	89.84	261.74	30.98	1376664	9234.38	336.88	114.59	5.67	556.88
Diesel	142.91	517.71	285.75	38.71	305.24	0.00	450.78	12938.11	151.53	0.00	0.00	626.06
Natural gas	110.54	378.77	178.54	46.51	178.95	5.25	5032.57	271.79	94.00	2.42	1.00 ^a	478.38
Coal	68.39	85.09	35.90	98.96	289.30	0.00	143.19	637.35	291.94	0.00	0.00	53.70
PV	13.47	1467.72	29.80	5.42	16.11	30.38	138230	2923.66	11.40	10.36	51.87	22.86
Biomass	9.18	1.00 ^a	16253.5	11.10	12.45	0.00	1.00 ^a	1.00 ^a	55.98	0.00	0.00	1.39
Nuclear BWR	8.95	1347.47	3.31	4.05	2.85	2586.93	733044	1326.06	3.41	3.44	6.71	2.87
Nuclear PWR	7.73	1254.00	2.90	3.58	1.43	1809.89	6230.51	1259.81	2.42	2.66	6.93	2.44
HAWT onshore	2.65	252.46	2.89	1.50	2.25	2.80	5462.47	1037.99	1.78	1.26	18.12	2.76
HAWT onshore modified	1.86	77.65	2.80	1.35	2.26	2.81	5473.94	750.81	1.59	1.15	12.35	2.76
HAWT offshore	2.39	203.46	2.48	1.48	1.94	2.61	4615.21	1015.19	1.54	1.21	18.17	2.31
HAWT offshore modified	1.80	72.98	2.41	1.37	1.94	2.61	4616.17	800.81	1.39	1.13	13.86	2.31
VAWT	1.55	85.11	2.56	1.01	1.82	3.51	6188.29	489.72	1.17	1.01	10.03	2.57
VAWT modified	1.49	10.68	2.55	1.00 ^a	1.82	3.51	6189.27	466.00	1.15	1.00 ^a	9.55	2.57
Hydro power	1.00 ^a	12.29	1.00 ^a	1.20	1.00 ^a	1.00 ^a	2693.69	182.62	1.00 ^a	8.52	3.29	1.00 ^a

^a indicates the minimum weighted value; other values indicate multiples of the minimum weighted value.

3.4 Summary

This chapter has compared LCI, LCIA and LCA outcomes of three 50 MW wind farms of both horizontal and vertical categories. For horizontal axis category, Repower 5M turbine is chosen whereas, for vertical axis category, Sandia 1.6 MW machine is used. Overall study is performed based on ISO standards 14040 and 14044. Eco-indicator 99 impact assessment method is used which calculates mid-point impacts by 11 categories; namely, carcinogens, respiratory organics, respiratory inorganics, climate change, radiation, ozone layer depletion, ecotoxicity, acidification/ eutrophication, land use, minerals and fossil fuels. Various onshore and offshore horizontal axis wind turbine sub-assemblies are compared with each other. Out of these, nacelle carries the highest ecopoint and, thus, appears as the highest impact contributing one; largely surpassing other sub-assemblies by significant impact values. A large impact carrying foundation, however, accompanies with a low impact carrying tower and vice versa. Next, per kg life cycle process of all three farms are compared with each other. Within per kg process scale comparison, 'copper, at regional storage' process exposes the most adverse effect to the surroundings with distant difference amongst all the materials. Steel is the most dominantly used material in all three farms; nevertheless, in comparison to steel, copper issues 27.45 times of total life cycle impact. Henceforward, copper is replaced with other material and life cycle impacts in all three farms have greatly reduced. Finally, all major renewable and non-renewable power generation sources are compared with each other which reveal the wind farms competing remarkably with other technologies. Within the two most leading renewable technologies, solar farm exhibits 5 times impact of onshore wind farm which rises to almost 8.7 times in comparison to onshore vertical axis farm for every kWh of electricity generation. Thus wind power performs better than solar power from LCA impact perspective.

Chapter 4

Material selection for horizontal axis wind turbine blade and tower

4.1 Introduction

This chapter focuses on optimal material selection for horizontal axis wind turbine blade and tower. In Chapter 3, vertical axis wind turbine is observed to possess lots of promises, viz., lesser weight, simpler foundation and lesser LCA impact, within all competitive wind turbine technologies. However, it's still not the mainstream wind power technology due to its lower energy conversion efficiency, dynamic instability and catastrophic failure problems; hence, only horizontal axis wind turbine material selection is focused here based on multiple constraints and compound objective based design principle. Ashby material selection approach is explored hereby with appropriate material index that matches well with wind turbine blade and tower material [90].

This chapter is mostly taken from author's published work [94], with permission from Elsevier.

Both onshore and offshore wind turbine are covered and 4 optimization objectives, namely, mass, cost, embodied energy and carbon footprint efficiency are set as design criteria. All of these objectives are minimized simultaneously to attain best blade and tower material from the available engineering materials. Finally, all proposed material are compared with existing commercialized design materials; both for blade and tower.

4.2 Blade material selection for horizontal axis wind turbine

4.2.1 Material indexing for blade material

Ashby approach, at the outset, translates the design necessities into performance objectives. In subsequent stages, it screens out unsuitable materials based on performance potentiality. Surviving materials are ranked, accordingly, to achieve the top ranked materials pertaining to an objective. Objectives are basically aims or targets that are achieved throughout the design exercise. The minimum or maximum level to which these objectives are achieved are dictated by constraints. Finally, the objective function takes a form of: $P = P(F, G, M)$ where F , G , M indicate functional, geometric and material parameters. For specific values of F and G , performance function, P can be improved by optimizing the appropriate material property combination M , which is termed as material index [95].

According to this approach, the functional constraints have to be attributed in early stage in correspondence to each objective which ultimately leads to formulate the performance function with specific material index. In this context, major design loads need to be identified according to wind turbine certification standards that will be used as optimal blade material selection constraints. As highlighted in IEC 61400 and GL 2010 standards, ultimate design cases for wind turbine encompass a wide range of categories, as follows:

- extreme loads (50 year return gust speed) in non-operational machine state
- 50 year return rising and failing gust
- 50 year return gust speed
- 50 year return direction change for steady hub-height wind speed, control system fault, protection system fault
- extreme wind shear
- start-up and shut-down cases with 1 year return rising and falling gust
- loss of load and grid loss which causes the rotor to accelerate until the braking system is brought into action
- clearance between blade and tower
- dynamic response
- effect of rain, hail, snow, ice, temperature, lightning and chemically active substances

Whereas from loading point of view, these loads are of many different variations; from stress analysis perspective on a typical horizontal axis wind turbine blade, all these can be categorized into two stress types – flapwise bending and edgewise bending stress. Hence, both flapwise and edgewise bending stresses will be used as design constraint of blade according to Ashby optimal material selection approach [96]. Another important blade design constraint relates with edgewise bending stiffness which encompasses the analysis based on the clearance between blade and tower.

As for the normal operation cycle of a wind turbine is concerned, the turbine blades usually remain oriented such that their wide faces remain roughly parallel with the hub-rotation axis and, in case of so-called *up-wind design*, their leading edge faces the wind. In other words, effective wind direction as experienced by the blades is in the rotational plane of the rotor although real-wind direction is orthogonal to it.

Accordingly, the flapwise bending load gives rise to out-of-plane deformation of the rotor blade in direction parallel to the axis of rotation while edgewise bending occurs in the plane of rotation. Accordingly, flapwise bending load is considered as the design constraint for spar material while edgewise load is considered for shell material design constraint. Additionally, edgewise bending stiffness is also used as another constraint for shell material to ensure the clearance between blade and tower.

A three blade horizontal axis wind turbine (HAWT) with blade length, R , average chord length, c , and average chord width, d , is considered herewith an optimal aerodynamic shape to determine material index for spar and shell, as shown in Figs. 4.1-4.2. Density of air and blade materials are denoted by ρ_a and ρ , respectively. Some structural assumptions are made in this concourse according to industry guidelines, which are as follows [97-98]:

- Average chord width (d) is one-fifth of average chord length (c).
- A spar with a square cross-section is used to increase the bending rigidity. Each side of the spar equals to 1/3 of the average chord length (c).
- Thickness of the spar equals to 1.5% of average chord length (c).
- Thickness of the shell equals to 1% of average chord length (c).

Blade length (R) is assumed to be a fixed variable; and chord length, average chord width and spar thickness are assumed to be free variables. Consequently, the spar mass, m_{spar} , can be estimated as such:

$$m_{spar} = \left\{ \left(\frac{c}{3} \right)^2 - \left(\frac{c}{3} - \frac{3c}{100} \right)^2 \right\} R \rho = 0.0189 c^2 R \rho \quad (4.1)$$

whereas shell mass, m_{shell} , is given by:

$$m_{shell} = \left\{ cd - \left(c - \frac{2c}{100} \right) \left(d - \frac{2c}{100} \right) \right\} R \rho = 0.0236c^2 R \rho \quad (4.2)$$

Equations (4.1) and (4.2) are the objective functions for spar and shell mass minimization.

Fig. 4.1: Schematic diagram of a blade

With reference to blade root, flapwise bending moment amounts to the product of thrust force per blade times $2/3$ of the radius [15]. So, for a conventional three bladed horizontal axis turbine, the flapwise moment is expressed as:

$$M_{flap} = \frac{2 \left(\frac{4\pi\rho_a R^2 V_1^2}{9} \right) R}{9} = \left(\frac{8}{81} \right) \pi \rho_a R^3 V_1^2 \quad (4.3)$$

In turn, the bending moment in edgewise direction at the root of a single blade can be derived from rotor torque [15]:

$$M_{edge} = \frac{\dot{E}_{extracted}}{3\Omega} = \frac{0.593 \left(\frac{1}{2} \dot{m} V_1^2\right)}{3\Omega} = \frac{0.067 \rho_a A_2 V_1^3}{\Omega} \quad (4.4)$$

where A_2 represents rotor swept area, Ω indicates rotational speed of turbine and ρ_a is the density of air.

Maximum flapwise stress due to bending at the root of blade can be expressed using beam bending theory as:

$$\begin{aligned} \sigma_{flap} &= \frac{M_{flap} C_f}{I_{flap}} = \frac{M_{flap} \left(\frac{0.33c}{2}\right)}{I_{flap}} = \frac{53.1 M_{flap}}{c^3} \leq \sigma_f \\ \Rightarrow c &= \left(\frac{53.1 \pi \rho_a R^3 V_1^2}{\sigma_f} \right)^{1/3} \end{aligned} \quad (4.5)$$

where C_f is the distance from the neutral axis to spar end in flapwise load direction, I_{flap} is the second moment of area of blade cross section at root and σ_f indicates fatigue strength of the spar material. Accordingly, from Eqs. (4.1) and (4.5), spar mass can be expressed as:

$$m_{spar} = 0.0189 c^2 R \rho = 0.0189 \left(\frac{53.1 \pi \rho_a R^3 V_1^2}{\sigma_f} \right)^{2/3} R \rho = \frac{A \rho}{\sigma_f^{2/3}} \quad (4.6)$$

where $A = 0.0189 R (53.1 \pi \rho_a R^3 V_1^2)^{2/3}$. Hence, for a given blade size R and wind velocity V_1 , the blade mass can be minimized from Eq. (4.6) by minimizing the index:

$$MI_{spar} = \frac{\rho}{\sigma_f^{2/3}} \quad (4.7)$$

Next, maximum edgewise stress due to bending at the root of the blade can be expressed as:

$$\sigma_{shell} = \frac{M_{edge} C_e}{I_{ed}} = \frac{M_{edge} \frac{c}{2}}{0.0025c^4} = \frac{13.4\pi\rho_a R^2 V_1^3}{c^3 \Omega} \leq \sigma_f$$

$$\Rightarrow c = \left(\frac{13.4\pi\rho_a R^2 V_1^3}{\Omega \sigma_f} \right)^{1/3} \quad (4.8)$$

where C_e is the distance from neutral axis to shell end in edgewise load direction, I_{ed} is the second moment of area of blade cross section at root.

Fig. 4.2: Shell geometry approximation of the turbine blade

Accordingly, from Eqs. (4.2) and (4.8), shell mass can be expressed as:

$$m_{shell} = 0.0236 \left(\frac{13.4\pi\rho_a R^2 V_1^3}{\Omega \sigma_f} \right)^{2/3} R \rho = \frac{B\rho}{\sigma_f^{2/3}} \quad (4.9)$$

where $B = 0.0236R \left(\frac{13.4\pi\rho_a R^2 V_1^3}{\Omega} \right)^{2/3}$. Hence, for a given blade size, R , and wind velocity, V_1 , shell mass can be minimized from Eq. (4.9) by minimizing the index:

$$MI_{shell,be} = \frac{\rho}{\sigma_f^{2/3}} \quad (4.10)$$

Additionally, edgewise bending stiffness can be expressed as:

$$S_{ed} = \frac{P}{\delta} = \frac{3EI_{ed}}{R^3} = \frac{0.0075Ec^4}{R^3} \quad (4.11)$$

where $\delta = PR^3/3EI_{ed}$; with δ indicating deflection due to edgewise bending load, P .

Accordingly, from Eq. (4.11):

$$c^2 = \sqrt{\frac{S_{ed}R^3}{0.0075E}} \quad (4.12)$$

Henceforward, from Eqs. (4.2) and (4.12), m_{shell} can be represented as:

$$m_{shell} = 0.0236 \sqrt{\frac{S_{ed}R^3}{0.0075E}} R\rho = 0.27 R^{5/2} S_{ed}^{1/2} \frac{\rho}{\sqrt{E}} \quad (4.13)$$

Hence, for a given blade size, R , the shell mass can be minimized by minimizing the index:

$$MI_{shell,ed} = \frac{\rho}{\sqrt{E}} \quad (4.14)$$

Similarly, from Eqs. (4.7), (4.10) and (4.14), material index for blade carbon footprint, embodied energy and price minimization can be achieved by replacing ρ with ρC_{CO_2} , ρC_{EE} and ρC_{CO} whereas C_{CO_2} indicates total carbon dioxide produced and released into the atmosphere as a consequence of per unit material production (kg/kg), C_{EE} indicates energy content of the material (MJ/kg) and C_{CO} indicates material cost (USD/kg).

4.2.2 Spar material selection through normalized surface and weight factor approach

It can be inferred from material indices that selected objectives can be met with either lower values of the nominator or higher values of the denominator or both. That is, the optimal materials will be the one with lower density, carbon footprint, embodied energy and cost and with higher fatigue strength properties. At such phenomenon, the problem intrinsically turns into a conflicting objective problem as materials with lower mass usually have lower fatigue strength, higher carbon footprint, and as well as higher per unit price. In this context, selecting an appropriate blade material needs trade-off between conflicting objectives. Hence, six normalized surfaces (normalized with respect to GFRP) are plotted in relation to the aforementioned material indices in this section.

In essence, the objective of this study is to explore a better substitute material in comparison to the presently used ones. Today's multi-megawatt wind turbine blades are mostly made of glass fiber reinforced polymer matrix (GFRP) composite. In consequence to that, material index based surfaces are plotted with relative to glass fiber reinforced epoxy. Accordingly, spar mass made of an alternative material differs from the one made of glass fiber reinforced epoxy by the factor:

$$\frac{m_{new,sp}}{m_{GFRP,sp}} = \frac{m_{1,sp}}{m_{2,sp}} = \frac{A\rho_1}{\sigma_{f_1}^{2/3}} * \frac{\sigma_{f_2}^{2/3}}{A\rho_2} = \frac{212\rho_1}{\sigma_{f_1}^{2/3}} \quad (4.15)$$

where, 1 and 2 indicate substituted spar material and existing GFRP material, respectively. Similarly, spar carbon footprint, embodied energy and total cost made from an alternative material differ from one made of glass fiber reinforced epoxy by the factor:

$$\frac{CO_{2,new,sp}}{CO_{2,GFRP,sp}} = \frac{CO_{2,1,sp}}{CO_{2,2,sp}} = \frac{AC_{CO_2,1}\rho_1}{\sigma_{f_1}^{2/3}} * \frac{\sigma_{f_2}^{2/3}}{AC_{CO_2,2}\rho_2} = \frac{27.02\rho_1 C_{CO_2,1}}{\sigma_{f_1}^{2/3}} \quad (4.16)$$

$$\frac{EE_{new,sp}}{EE_{GFRP,sp}} = \frac{EE_{1,sp}}{EE_{2,sp}} = \frac{AC_{EE,1}\rho_1}{\sigma_{f_1}^{2/3}} * \frac{\sigma_{f_2}^{2/3}}{AC_{EE,2}\rho_2} = \frac{1.89 * 10^{-6}\rho_1 C_{EE,1}}{\sigma_{f_1}^{2/3}} \quad (4.17)$$

$$\frac{CO_{new,sp}}{CO_{GFRP,sp}} = \frac{CO_{1,sp}}{CO_{2,sp}} = \frac{AC_{CO,1}\rho_1}{\sigma_{f_1}^{2/3}} * \frac{\sigma_{f_2}^{2/3}}{AC_{CO,2}\rho_2} = \frac{10.5C_{CO,1}\rho_1}{\sigma_{f_1}^{2/3}} \quad (4.18)$$

Material properties like density, fatigue strength, per unit carbon footprint, embodied energy and cost of glass fiber reinforced epoxy are taken from *Cambridge Engineering Selector (CES) 2013* database [93]. In addition to the existing material indices, for safe operation, further limiting criterion on fracture toughness (K_{Ic}) and fatigue strength (σ_f) are considered with the following minimum values specified:

$$\sigma_f \geq 150 \text{ MPa} \quad (4.19)$$

$$K_{Ic} \geq 15 \text{ MPa} \sqrt{m} \quad (4.20)$$

Based on Eqs. (4.15)–(4.20), six normalized surfaces are plotted, as evident in Figs. 4.3–4.8. Fig. 4.3 highlights 10 materials that exhibit more competitive mass and carbon footprint with respect to reference material (GFRP composite). Out of these, PEEK/IM carbon fiber material designs the same size wind turbine spar with only 25% of current weight and, simultaneously, it reduces carbon footprint by 50%. Low alloy steel 0.42C 300M provides 50% carbon savings but increases the weight by nearly 80%. Similarly, beryllium alloy grade I-250 ensures 30% mass savings but it increases the spar carbon footprint by 13 times.

Fig. 4.3: Normalized mass and carbon footprint of spar material (with competitive materials inside the small box)

Fig. 4.4: Normalized mass and cost of spar material (with competitive materials inside the small box)

Fig. 4.5: Normalized mass and embodied energy of spar material (with competitive materials inside the small box)

Based on relative mass versus cost index based logarithmic plot, Fig. 4.4 features many materials from both composite and engineering alloys family better than current E-glass fiber reinforced epoxy material. For example, S-glass fiber reinforced epoxy material, on average, provides 60% mass and cost savings. Epoxy/high strength carbon fiber, on the other hand, presents nearly 80% mass and 75% cost savings. A variety of steel alloys, in turn, offer higher weight, sometimes up to 800%, with substantial cost reduction.

Figure 4.5 shows mass versus embodied energy correlation with respect to glass fiber reinforced epoxy composite. It is evident that though there are many materials available that can optimize either mass or embodied energy, only very few appear to be preferable when both the objectives are simultaneously applied. Beryllium grade I-2208 provides 50% weight savings, but increases per unit mass embodied energy by 1200%. It is to be noted that the Be alloys are expensive and have poor machinability and are

toxic in nature. Ferrous alloys, on the other hand, have less embodied energy (J/kg) during their production cycle. It is to be known that ferrous alloys, because of their high density, increase the inertial and gravitational forces considerably.

Fig. 4.6: Normalized carbon footprint and cost of spar material (with competitive materials inside the small box)

Figure 4.6 compares the spar materials based on carbon footprint and cost. It appears that cast iron based alloys perform the best in this category. Aluminum alloys, comparatively, provide lower cost and higher carbon footprint while cast iron, nodular graphite, BS 900/2 provides up to 85% carbon footprint and 95% cost savings.

In turn, Figure 4.7 provides almost a linear pattern between carbon footprint and embodied energy. Again steel and cast iron based alloys perform better than GFRP. Intermediate modulus (IM) carbon fiber reinforced poly ether ether kethone (PEEK) composites, which was previously exhibiting 80% mass savings with reference to GFRP,

now demonstrates 5 times carbon footprint emission and embodied energy consumption. It indicates profound conflict between the chosen objectives.

Fig. 4.7: Normalized carbon footprint and embodied energy of spar material (with competitive materials inside the small box)

Figure 4.8 is a log-log plot between cost and embodied energy objectives. Very few composite materials come out as competitive material in this stage when compared to GFRP. Similarly, aluminum alloys cannot perform well because of high embodied energy, though, on average, they offer nearly 90% cost saving with respect to the base material. Some tool steels perform similar to the base material. Another competitive material group comprises of glass fiber based polyamide materials which offers equivalent embodied energy with a massive 85% cost saving.

Fig. 4.8: Normalized cost and embodied energy of spar material (with competitive materials inside the small box)

One thing is evident from these plots that materials are exhibiting conflicting performances based on different objectives. The material which offers lower mass, may not performing equally well in other three objectives and vice versa. Therefore, materials with highest competitiveness are analyzed with a weight factor approach [95]. In weight factor approach, competitive objectives are initially assigned specific weightage (w_i) depending on their relative importance in design. Next the absolute values of the material indices (M_i) are scaled by either maximum or minimum property value [28]. Finally, all W_i s are summed up to get $\sum_{i=1}^n W_i$. The best material appears to be the one with highest $\sum_{i=1}^n W_i$.

$$W_i = w_i \frac{MI_i}{MI_{imax}} \text{ (if maximization of material index is design goal)}$$

or

$$W_i = w_i \frac{MI_{imin}}{MI_i} \text{ (if minimization of material index is design goal)}$$

For blade design, mass minimization emerges as the single most influential objective as all other objectives intensely depend on per unit mass of the blade material. Hence, mass minimization is assigned a higher weightage (50%). While, per unit carbon footprint, cost and embodied energy consumption minimization are assigned 20, 20 and 10% weightage, respectively. Though it is an arbitrary selection, its relative emphasis is supported by established norms of the design community and based on the commands of energy economics and corresponding manufacturing industry. As highlighted by Breton and Moe [99] and Dutton et al. [100], rotor mass is a potential design factor for further development of onshore and offshore wind turbine industry. There is a desire to develop longer blades to improve the turbine power rating as it goes square of the blade size [15]. But, the blade mass scales with increasing blade size. With increasing blade size, the system also emerges more vulnerable to fatigue as there are higher chances of having imperfections in them; also gravity load increases severely with higher length in addition to implied increase in centrifugal, braking and gyroscopic/yaw force.

Hence, the weight percentage of mass objective is set at a higher value. Next, from energy economics and investment payback criterion, blade material cost is an additional prime factor. In this regard, it is assigned a higher percentage of 20%. Next, addressing the scientific consensus on environmental concerns and its grim realities as well as acknowledging the overwhelming appeal of the environmentalists, carbon footprint minimization is considered a higher percentage in comparison to embodied energy factor [101].

Table 4.1: Spar material selection from material indices with respect to E-glass fibre epoxy composite

Material	Relative mass index, $\frac{212\rho_1}{\sigma_{f_1}^{2/3}}$	Relative carbon footprint index, $\frac{27.02\rho_1 C_{CO_2,1}}{\sigma_{f_1}^{2/3}}$	Relative cost index, $\frac{10.5C_{CO,1}\rho_1}{\sigma_{f_1}^{2/3}}$	Relative embodied energy index, $\frac{1.89 * 10^{-6}\rho_1 C_{EE,1}}{\sigma_{f_1}^{\frac{2}{3}}}$	$\sum_{i=1}^n W_i$
PEEK/IM carbon fiber	0.259	0.821	1.07	0.696	0.75
Epoxy/HS carbon fiber	0.3	0.659	0.393	0.728	0.71
Cyanate ester/HM carbon fiber	0.292	1.07	1.85	0.894	0.63
Epoxy/Aramid carbon fiber	0.359	0.872	0.69	0.759	0.59
Oak (quercus spp.)	1.52	0.0868	0.686	0.102	0.41
Lignumvitae	1.98	0.113	0.894	0.133	0.32
Alloy steel, AISI 9255	2.02	0.559	0.0722	0.645	0.31
Alloy steel, 0.40C 300M	1.87	0.502	0.102	0.58	0.26
Poly Propylene (42% glass)	1.29	1.11	0.22	1.15	0.19

Table 4.1 highlights the best competitive materials of normalized surfaces for weight factor based calculation. It is evident that some composite materials are exhibiting superior performance in comparison to engineering alloys. Out of these, unidirectional PEEK/IM carbon fiber and epoxy/HS carbon fiber, laminates top the list. PEEK/IM carbon fiber composite possesses high tensile strength (2.41 - 2.43 GPa) as well as high

fatigue strength (1.33 - 1.58 GPa at 10^7 cycles) which are very aspiring properties for spar application. On the other hand, within engineering alloys, most competitive ones are low alloy based AISI 9255 and low alloy based 0.40C 300M steels. These alloys possess fatigue strength of 720 and 840 MPa, at 10^7 cycles, respectively, which is lower than the aforementioned composite material properties. Accordingly, concerning the overall analysis, PEEK/IM carbon fiber, UD composite and epoxy/HS carbon fiber, UD composite are the best spar materials based on applied weightage factor.

4.2.3 Shell material selection through normalized surface and weight factor approach

Likewise spar material, another weight factor based analysis is performed for shell material from the data of CES plots. Herein, individual plots are drawn for each objective (mass, carbon footprint, embodied energy and cost) based on two material indices, as stated in Eqs. (4.10) and (4.14).

Fig. 4.9: Material index due to edgewise stiffness vs. material index due to edgewise strength for shell mass minimization objective

Relevant CES plots for four objectives are presented in Figs. 4.9-4.12 which illustrate almost linear nature of the material properties, i.e., most materials with high fatigue strength also carry high stiffness property.

Fig. 4.10: Material index due to edgewise stiffness vs. material index due to edgewise strength for shell carbon footprint minimization objective

Fig. 4.11: Material index due to edgewise stiffness vs. material index due to edgewise strength for shell embodied energy minimization objective

Finally, fracture toughness (K_{IC}) and fatigue strength (σ_f) based limiting criteria are applied, as specified in Eqs. (4.19) and (4.20). Table 4.2 highlights the weight factor calculation for shell material based on above-mentioned analysis. Herein, 50% weightage is put on mass based material indices; with each constraint based index (either edgewise strength based index or edgewise stiffness based index) getting 25% weight. Next, each constraint of carbon footprint and embodied energy is allocated 10% weight; thus accumulating the total weightage for each objective at 20%. Finally, material cost based indices is calculated with 10% weightage; with each constraint based index equally sharing the weight factor.

Fig. 4.12: Material index due to edgewise stiffness vs. material index due to edgewise strength for shell cost minimization objective

Table 4.2: Shell material selection from multiple constraint and compound objective based indices

Material	$\frac{MI_{imin}}{MI_i}$	$\frac{MI_{imin}}{MI_i}$	$\frac{MI_{imin}}{MI_i}$	$\frac{MI_{imin}}{MI_i}$	$\frac{MI_{imin}}{MI_i}$	$\frac{MI_{imin}}{MI_i}$	$\frac{MI_{imin}}{MI_i}$	$\frac{MI_{imin}}{MI_i}$	$\sum_{i=1}^n W_i$
	(based on $\frac{\rho}{\sigma_f^{2/3}}$)	(based on $\frac{\rho}{\sqrt{E}}$)	(based on $\frac{\rho C_{CO_2}}{\sigma_f^{2/3}}$)	(based on $\frac{\rho C_{CO_2}}{\sqrt{E}}$)	(based on $\frac{\rho C_{EE}}{\sigma_f^{2/3}}$)	(based on $\frac{\rho C_{EE}}{\sqrt{E}}$)	(based on $\frac{\rho C_{CO}}{\sigma_f^{2/3}}$)	(based on $\frac{\rho C_{CO}}{\sqrt{E}}$)	
Cyanate ester/HM carbon fiber	0.88	1	0.37	0.38	0.59	0.46	0.018	0.012	0.65
PEEK/IM carbon fiber	1	0.70	0.26	0.17	0.39	0.19	0.037	0.016	0.53
Low alloy steel AISI 9255	0.125	0.17	0.8	0.96	0.99	0.92	0.96	0.78	0.53
Carbon steel AISI 1340	0.12	0.16	0.77	0.99	0.99	0.98	0.84	0.73	0.52
Epoxy S glass fiber	0.36	0.33	1	0.81	1	0.63	0.064	0.036	0.52

It appears that, likewise spar, same group of material emerges as the best for shell. Out of this, high modulus (HM) carbon fiber based cyanate ester composite tops the list. Cyanate ester/HM carbon fibre possesses high Young's modulus (335 GPa) which ensures very high edgewise stiffness to maintain necessary tower to blade clearance distance during turbine operation time. Moreover, this material possesses very high fatigue strength of 1.31 GPa (at 10^7 cycles) whereas its yield strength stands at 2.19 GPa. It also carries excellent fracture property with mode I fracture toughness value of 50.6 MPa.m^{0.5}. The next material in list i.e. PEEK/IM carbon fiber composite, which is the best material for spar application, also holds strong basis as a competitive shell material with excellent properties. Besides other properties that are mentioned in spar material discussion, it possesses very high Young's modulus of 148 GPa. Once the density term is considered, the specific Young's modulus of this material will be extremely high in comparison to most of the engineering materials.

4.2.4 Final blade material for wind turbine

Same group of materials turn up as the best performing materials both for spar and shell study. Accordingly, few case-studies are presented herein with different blade materials on how much mass, carbon footprint, energy content and cost savings these can ensure in comparison to presently used blade materials. Vestas 90 m 3.0 MW turbine is chosen hereby as a representative wind turbine of the industry. Each blade mass of this turbine currently stands at 6.7 ton [102]. With PEEK/IM carbon fiber material in both spar and shell application, the blade weight stands at ~1750 kg; this is around 74% reduction in comparison to current mass. Simultaneously, PEEK/IM carbon fiber blade *cradle to gate* life cycle processes decrease energy consumption by 228 GJ (around 30% reduction) and atmospheric carbon dioxide emission by 9400 kg (around 17% reduction). All these benefits are achievable only with 7% increase in blade

material cost. In turn, with UD epoxy/HS carbon fiber, blade cost reduces by 60% with concurrent 70% mass, 34% carbon footprint and 23% energy consumption reduction.

As for onshore small scale wind turbine, some hybrid natural materials like oak (*quercus* spp.) and *lignumvitae* (l) enter into the design space with lower fatigue strength and lower fracture toughness limiting criteria. These materials outperform all remaining ones of all material families, except some composite ones. Thus, for low scale (<100 kW) onshore turbines, aforesaid materials can also be used for potential blade application in gentle windy regions. For offshore blade application, these small scale turbines are, however, undesirable; partially because of the increased cost of bottom support structure. Moreover, there are additional infrastructural hindrances in onshore locations as massive size turbines frequently face transportation related problems. Hence, small scale turbines are sought for in onshore locations; but not in offshore.

Offshore wind energy is the industry cluster where the turbine size limit is still a long way to reach. However, fatigue loadings and stress corrosion cracking appear more severe in offshore locations due to salinity effect, moisture absorption and thermal gradient issues between turbine top and bottom section. Particularly, atmospheric ambient oxygen often drastically affects the above-water environment even in comparison to immersive state that indubitably affects the offshore wind turbine blade material selection aspects. Thus, the moisture absorption permeability plays a very important role in offshore blade design.

As within the listed materials in Table 4.1 and Table 4.2, intermediate modulus carbon fiber/PEEK composite absorbs moisture at intensely high rate. Its 24 hour water absorption permeability value stands at 0.47% which signifies a 0.47% weight gain in each 24 hour for the sample immersed in distilled water at room temperature [93]. Rests

of the composite materials show much lower water absorption permeability, in the digit of second decimal, which makes them lucrative for offshore blade application. Apart from base materials used, appropriate protection systems need to be ensured in blade structure to ensure adequate durability and safety of the system.

4.3 Tower material selection for horizontal axis wind turbine

4.3.1 Material indexing for tower material

Likewise blade structure, wind turbine tower also endures highly turbulent aerodynamic loads and numerous fatigue producing stress cycles over entire design life of 20 years [15-16, 72]. Detailed of these wind turbine tower design loads are mentioned in Section 4.2.1 according to IEC 61400 and GL 2010 standards. Here, in the study, aerodynamic loads on wind turbine tower and tower top masses are considered as bending and stability/buckling design constraints, respectively, for wind turbine tower material selection. Aerodynamic loads on tower are of two types: i) one due to thrust force on rotor which acts as a cantilever beam bending scenario to the whole tower; ii) another is distributed bending load on whole length of tower. The second one is insignificant in comparison to first one. According to literature [115], the first load i.e. the cantilever bending load on tower can be presented as:

$$P_1 = \left(300 \text{ N/m}^2 \right) * \text{rotor swept area}$$

whereas, according to Uys et al. [116], the second load can be expressed as

$$P_2 = \left(6.5 \text{ KN/m} \right) * \text{tower length}$$

For Vestas V90 turbine, with rotor swept area of 6362 m² and tower length of 90 m [102],

$$P_1 = 1908.6 \text{ kN}$$

$$P_2 = 585 \text{ kN}$$

It is visible here that P_2 is 30% of P_1 ; hence, P_2 is not considered in the author's material selection analysis. Henceforward, a tapered tubular tower is considered with height, L , mean outer diameter, D , and mean wall thickness, t . Structural material density is represented by ρ , as highlighted in Fig. 4.13. The following assumptions about the tower design are made herewith [72]:

- Tower mean outer diameter, D , is the arithmetic mean of tower top and bottom
- Tower top outer diameter, D_{top} , is assumed to be 50% of tower mean outer diameter, D .

Fig. 4.13: Schematic diagram of tapered tower with hollow tubular cross-section

Tower is the heaviest component of a wind turbine structure. Hence, its mass minimization attracts due consideration. Tower mass can be expressed as:

$$m_{tower} = \pi D L t \rho \quad (4.21)$$

For transverse bending and buckling load based constraint, tower length, L , and mean wall thickness, t , are considered as fixed variable whereas mean outer diameter, D , is assumed as free variable. Hence, the bending stress, due to aerodynamic forces, on the wind turbine tower can be expressed as:

$$\sigma = \frac{M_{bend}C}{I} = \frac{FL * \frac{D_{top}}{2}}{\frac{\pi D_{top}^3 t}{8}}$$

$$\Rightarrow D^2 = \frac{16FL}{\pi \sigma t} \quad (4.22)$$

where C indicates average tower radius (i.e. the average distance between bending neutral axis to tower skin), F indicates aerodynamic force on tower and M_{bend} symbolizes tower base bending moment due to corresponding force. From Eqs. (4.21) and (4.22):

$$m_{tower} = \left\{ 16\pi(F)(L^3 t) \left(\frac{\rho^2}{\sigma} \right) \right\}^{0.5}$$

Hence, considering the tower as a beam structure for the transverse bending issue, its mass can be minimized by minimizing the index:

$$MI_{bend, mass} = \frac{\rho}{\sigma^{0.5}} \quad (4.23)$$

Next, bucking load for a tapered column can be expressed as [103]:

$$P = \frac{\pi^2 EI}{L_e^2} = \frac{\pi^3 E D^3 t}{12.48 L^2}$$

$$\Rightarrow D^3 = \frac{12.48 P L^2}{\pi^3 E t} \quad (4.24)$$

From Eqs. (4.21) and (4.24):

$$m_{tower} = \left(\frac{12.48PL^5t^2\rho^3}{E} \right)^{1/3} = \left\{ 12.48(P)(L^5t^2) \left(\frac{\rho^3}{E} \right) \right\}^{1/3}$$

Hence, the material index for mass minimization due to buckling can be represented as:

$$MI_{buckle,mass} = \frac{\rho}{E^{1/3}} \quad (4.25)$$

As evident from Eqs. (4.23) and (4.25), blade mass will be minimized with lower value of the material indices. Likewise blade study, density, ρ , can be replaced with ρC_{CO_2} , ρC_{EE} and ρC_{CO} to get appropriate material indices of carbon footprint, embodied energy, and cost minimization, respectively.

4.3.2 Tower material selection by coupling equation and weight factor approach

It appears from aforesaid discussion that, for each performance objective of tower material selection, there are two material indices which resembles an over-constraint based design (more constraints than free variable). Nevertheless, multiple constraints based equations for same objective should provide equal value as tower material ultimately can have only one total weight, cost, carbon footprint or embodied energy quantity. In this regard, available material indices need to be coupled to find only one performance measure for each objective. According to Eqs. (4.23) and (4.25), for mass minimization objective,

$$\frac{MI_{bend,mass}}{MI_{buckle,mass}} = C_c$$

where, C_c indicates coupling constant. Specifications of a Vestas V90-3.0 MW turbine are collected from manufacturer website to evaluate the coupling constant, C_c , of this turbine [102, 115]:

$$C_c = \frac{(12.48 * 2199.4 * 10^3 * 90^5 * 0.28^2)^{1/3}}{(16\pi * 0.96 * 10^6 * 90^3 * 0.28)^{0.5}} = 0.08$$

Similarly, for carbon footprint, cost and embodied energy consumption minimization, the material indices can be represented as:

$$\frac{MI_{bend,CO_2}}{MI_{buckle,CO_2}} = \frac{MI_{bend,CO}}{MI_{buckle,CO}} = \frac{MI_{bend,EE}}{MI_{buckle,EE}} = C_c \quad (4.27)$$

Fig. 4.14: Material index due to buckling vs. material index due to bending for mass minimization objective

As in the blade design, further constraints are imposed on the materials based on fatigue strength and fracture toughness, as mentioned in Eqs. (4.19) and (4.20). Above properties ensure that wind turbine tower materials will at least possess this minimum

level of respective values. Finally, a logarithmic plot of mass based indices is plotted in Fig. 4.14 based on Eqs. (4.26). The logarithmic form, $\log(MI_{bend, mass}) = \log(MI_{buckle, mass}) + \log 0.08$ is a family of straight parallel lines of slope 1 on a plot $\log(MI_{bend, mass})$ against $\log(MI_{buckle, mass})$ with the selection guideline corresponds to a value of $C_c = 0.08$. All materials on this selection line perform equally well for tower design. As evident from Fig. 4.14, only a small number of materials are left above the selection line. Consequently, the selection area can be expanded to the nearby materials, both right and left. Composite materials appear to out-perform other material groups in ensuring mass-minimal design. Out of these materials, cyanate ester/high modulus carbon fiber performs best. Comparing to the steel alloys, it offers around 7 times weight reduction. Titanium based alloys also compete better in comparison to other metal alloy families, but Ti based designs are very expensive.

Fig. 4.15: Material index due to buckling vs. material index due to bending for carbon footprint minimization objective

Figure 4.15 depicts carbon footprint based material indices for bending and buckling constraint analysis. In this category, cast irons and various steel grades excel other materials in ensuring lower carbon footprint. Epoxy/high strength carbon fiber, which is the best composite material of this category, leaves at least 6 times more carbon impact of cast iron, BS 900/2 material. Aluminum alloys also fail to ensure lower carbon footprint.

Fig. 4.16: Material index on buckling vs. material index on bending for cost minimization objective

Figure 4.16 shows the material indices for tower material cost optimization and, again, the cast iron based alloys dominate with lower price. Some aluminum and magnesium alloys also perform well. In turn, composite materials exhibit much higher price which makes it less desirable for tower application as wind turbine structure needs a very long tower to achieve a higher power output with increased air velocity.

Material selection chart for energy content objective is shown in Fig. 4.17 with the coupling constant line for the constraints of bending and buckling. Cast iron and carbon steel based materials outperform others in lower energy consumption. Composite materials do not perform well as their manufacturing process needs lots of energy consuming stages. Like previous, very few materials remain on above the selection gradient line. Hence, nearby competitive materials also need to be explored as a potential tower material.

Fig. 4.17: Material index due to buckling vs. material index due to bending for embodied energy minimization objective

It is evident from Figs. 4.14-4.17 that materials are exhibiting multi-faceted performance based on different objectives. A preliminary list of candidate materials is presented in Table 4.3 for further investigation with respective material indices. Minimum index values are shown at bold, as minimization of the competing objectives is the design goal and all other values are normalized with respect to minimum index value in each

individual objective category. Mass minimization is again accepted as the single most influential objective as tower is the heaviest component on a wind turbine and other objectives also depend on total tower mass. Accordingly, mass based index is assigned 50% weightage – 25% for bending based mass index and another 25% for buckling based mass index. The other objective measures are, as well, assigned equal weightage for two constraints. As both the constraints together, carbon footprint, cost and embodied energy consumption minimization are assigned 20, 20 and 10% weightage; respectively. Each competitive material weight factors are summed up consequently which is presented in Table 4.2. The material with highest value of the weight factor will be the best material based on weight factor definition in Section 4.2.

Table 4.3: Value of the material indices corresponding to bending and buckling load constraints

Material	$\frac{\rho}{\sqrt{\sigma}}$	$\frac{\rho}{\sqrt[3]{E}}$	$\frac{\rho C_{CO_2}}{\sqrt{\sigma}}$	$\frac{\rho C_{CO_2}}{\sqrt[3]{E}}$	$\frac{\rho C_{EE}}{\sqrt{\sigma}}$	$\frac{\rho C_{EE}}{\sqrt[3]{E}}$	$\frac{\rho C_{CO}}{\sqrt{\sigma}}$	$\frac{\rho C_{CO}}{\sqrt[3]{E}}$	$\sum_{i=1}^n W_i$
Epoxy/HS carbon fiber	0.045	0.301	0.791	5.18	1.2e7	7.9e7	1.93	12.6	0.62
Cast iron, BS 900/2	0.398	1.29	0.405	1.31	6.8e6	2.2e7	0.235	0.761	0.58
Steel alloy, AISI 9255	0.293	1.32	0.619	2.79	1e7	4.6e7	0.22	0.99	0.45
Steel alloy, AISI 1340	0.309	1.33	0.767	3.29	9.8e6	4.2e7	0.209	0.898	0.44
Steel alloy, 0.40C 300M	0.27	1.33	0.571	2.81	9.4e6	4.6e7	0.299	1.47	0.42
Steel alloy, AISI 420	0.308	1.32	1.57	6.71	2.5e7	1.1e8	0.742	3.17	0.22

Unidirectional epoxy/HS carbon fiber laminates arises as the best competitive material for tower application. Cast iron based nodular graphite alloy, BS 900/2, simultaneously, performs very well. This alloy possesses high yield strength of 750 MPa and high fatigue strength of 325 MPa at 10^7 cycles. It has significant ductility and it is very strong in compression. On top of all, its carbon footprint and embodied energy content values are remarkably low comparing to other ferrous alloys. Though steel price is increasing worldwide and composite price is decreasing at the same time; still cast iron, nodular graphite, BS 900/2 alloy holds lots of promises for wind turbine tower structure. Its per unit cost is only 12% of epoxy/high strength carbon fiber, UD composite composite. However, in comparison to epoxy/high strength, it provides a much heavier design (8.67 times) with the same geometry and same loading condition.

More steel based alloys will appear to be superior once the weight percentages of the objectives are changed. Down the perspective, assuming the REpower 5M turbine (designed for both onshore and offshore) tower as originally made of AISI 316 steel (which is one of the most widely used steel grade worldwide), overall mass, carbon footprint, embodied energy and cost reduction occurs by around 6%, 81%, 80% and 90% respectively with cast iron, nodular graphite BS 900/2 being considered as candidate material [104]. Epoxy/HS carbon composite provides even more weight savings (almost 78%) with simultaneous carbon footprint and embodied energy reduction but at a moderately higher cost (almost 67% increase). Same material will be applicable for offshore tower design once the moisture absorption permeability is considered.

4.4 Summary

Today engineering designs are increasingly driven by product functionality, usability, mass, cost and environmental sensitivity. An improper choice of material adversely

affects the performance, profit margins and undermines the long-held reputation in the presence of any accidents. Hence, engineers are always on a lookout to streamline their product lifecycle. In this chapter, we explored wind turbine blade and tower material selection strategies based on multiple constraints and conflicting objective based principle. Blade material is optimized based on both flapwise and edgewise loading whereas tower material is optimized based on both transverse bending and axial buckling.

Appropriate mass based material indices are determined from aerodynamic and structural performance equations; and, these material indices are translated into carbon footprint, cost and embodied energy minimization indices. Wind-turbine is a fatigue-prone machine; hence, fatigue strength criterion is used in material indices rather than yield failure principle. Limiting criteria of ultimate fatigue strength and mode I stress intensity factor are used, both for onshore and offshore wind turbine blade and tower, to ensure structurally superior materials. Mode I stress intensity factor is usually the most critical one within all stress intensity factors and, hence, it is a critical design parameter for any engineering structure. For offshore designs, another criterion of moisture absorption permeability is evaluated for competing materials.

At the end, both optimized blade and tower materials are compared with existing material and it showed considerable material, carbon footprint, cost and embodied energy efficiency are attained with newer materials. Novelty of the work includes material selection for wind turbine blade and tower from material indexing based on proper aerodynamic loading condition. Further to that, here materials are optimized based on four competing objective rather than only focusing on structural weight minimization, as been conducted in previous study [35].

Chapter 5

Fracture characterization of glass fiber reinforced polymer matrix composite laminates under biaxial loading

5.1 Introduction

This chapter presents bi-axial fracture response of cross-ply glass fiber reinforced epoxy matrix composite laminate. The chapter first introduces an optimal cruciform geometry design work based on finite element (FE) simulation. Next, it characterizes glass fiber composite material under different mechanical testing. Next, it reports the results on biaxial experimentation of GFRP material in presence of various crack profiles oriented at different angles. Finally, it analyzes the biaxial results in comparison to uniaxial ones. Detailed experimental processes include determination of equivalent loaded area from uniaxial testing along each longitudinal and transverse direction of cruciform specimen and comparing the findings with results of tensile specimen failure under uniaxial loading. Next, notch-sensitivity behavior of glass fibre reinforced plastic material is studied under uniaxial loading. Afterwards, interlaminar and translaminar mode I fracture toughness properties of GFRP material are determined experimentally before biaxial testing. Optimized cruciform specimen is analyzed simultaneously by finite element simulation to numerically calculate mode I and mode II fracture toughness.

Geometric factors relevant to cruciform shape are determined to calculate relevant fracture toughness properties. Final geometry factors match well with experimental test results and accordingly mode I and mode II fracture toughness values are determined.

5.2 Biaxial test frame design

The biaxial test frame was developed in collaboration with Instron Singapore Pte Ltd. It consists of a primary (axial) axis load carrying capacity up to 250 kN with secondary (transverse) axis actuator supports up to 10 kN. The secondary axis load frame sits upon the primary axis uniaxial load frame and is mechanically attached to the uniaxial test frame. Complementary actuator load control is electronically interlocked with primary axis actuator by a constant load ratio principle. A schematic view of the test setup is shown in Fig. 5.1.

The right slider of secondary actuator is designed to allow tension and compression of the specimen along the transverse direction [see Fig. 5.1]. In order to suppress the out of plane buckling of the cruciform geometry under compressive loading, anti-buckling guides are installed. Secondary actuator frame provides uniaxial tension and compression force in secondary axis through a constant load controlling mechanism in compliance with Instron 5985 uniaxial test machine. A pneumatically controlled cylinder compresses inlet air and drives a piston to generate force which in turn generates necessary displacement of the secondary frame. As the frame starts moving, it transmits a tensile or compressive force on the biaxial specimen in transverse direction.

Before starting the experiment, biaxial test-frame needs to be securely fastened to Instron 5985 machine by compliant support mechanism. A cable support is assured at the rear end of biaxial set-up to maintain proper alignment of the specimen between

two secondary sliders. One end of the cable is fastened to 20.5 cm long pneumatic cylinder while another end is annexed to stationary standing frame. It is crucial to ensure proper alignment of the overall setup as well as alignment between the setup and specimen centre before commencement of the experiment to reduce any unnecessary movement of the secondary frame that might transmit bending effect to the specimen and, hence, adversely contribute to applied net force.

Fig. 5.1: Schematic diagram of experimental biaxial test-frame

Before conducting any experiment, real-time responsiveness of the secondary frame and its conformity with primary loading axis is investigated. To check this, compressor air pressure is elevated first to 0.2 MPa and load ratio is set at an arbitrary value. Small loads are applied consequently in the biaxial system and relationship between axial

force and transverse force increment is carefully traced. Once the system starts to respond to the set load ratio, the cylinder pressure is increased to 0.6 MPa to ensure prompt responsiveness of the secondary system (see Fig. 5.2).

Fig. 5.2: (a) Air compressor (JUN-AIR Model OF302) to drive the piston of pneumatic cylinder; (b) pressure gage

5.3 Optimal design of cruciform geometry

In line with the development of biaxial test-frame, it is significantly important to propose an optimal specimen geometry that will eventually match with the system fixture and loading constraints. Finite element simulations are used in this pursuit to reduce the number of design variables and thus to achieve an optimized geometry. Ten different cruciform shapes are studied in this connection, out of which four representative ones are presented here with key results [see Fig. 5.3]. The objective and essential feature of this analysis is to attain far-field stress uniformity for representative specimen geometries within central location of 100 mm x 100 mm area under different loading conditions.

In several earlier studies of cruciform specimen optimization, different specimen designs were explored with various shapes and profiles [46, 48-49]. These include arrangement of wedge grip alignment holes for precise alignment of wedge grips, tapering of loading arm width between wedge grip and gage section to increase the likelihood of failure in central gage area, increasing of loading arm width in wedge grip to the full dimension of test fixture to lower the uniaxial stress and to diminish the failure probability in loading arms; evaluation of corner radius between adjoining arms to reduce stress concentration; CNC milling in gage section to form a narrower cross-section and thus to ensure a stress-riser in central section; usage of thicker laminates in loading arms to ensure failure to occur in biaxially loaded gage section and arrangement of additional corner rounding in cruciform specimens.

This study, however, plans to study the biaxial response of composite materials in presence of various cracks. Accordingly, it is less challenging to achieve a gage section failure in this study. Therefore, simulation runs with tapered specimen or resorting to milling towards the central zone is not pursued here. But another challenging issue arises here as the study deals with crack and imperfections: it appears difficult to achieve uniform remote stress with the gage section because of stress singularity issues carried by notch. A uniform far-field stress could be attained easily in the central region of a neat specimen; but it is, nevertheless, difficult to achieve in the presence of imperfection. This problem is compounded by the fact that, in actual experiment, boundary conditions of specimen holding regions do not remain ideal. Respective to that, it is planned to study cruciform geometries by a coefficient of variance (COV) calculation on stress values. This follows in calculation of mean and standard deviation of stresses along defined rings of mesh elements near the boundary zones of gage

section and ultimately optimization of critical specimen geometry based on these COV values. For simulation runs, specimens with following attributes are selected:

1. Cruciform specimen with straight and rounded corner fillet
2. Cruciform specimen with inclusion of corner rounding in addition to corner fillet

An illustrative diagram of four representative geometries is highlighted in Fig. 5.3. These geometries will sequentially be referred as Geometry A, B, C and D, in subsequent sections. In Geometry A, rounded corners are ensured with corner fillet radius of 25 mm. In Geometry B, the corners between adjoining arms are straight to minimize the specimen fabrication cost. In Geometry C, a corner rounding is used with curvature radius of 30 mm while in Geometry D, a curvature radius of 15 mm is used at the end of tapered arm section. All geometries are 320 mm long and 200 mm wide to match with experimental biaxial frame.

The uniaxial properties of GFRP composite material are experimentally measured following several ASTM standards [105-107] and these are listed in Table 5.1.

Geometry A

Geometry B

Geometry C

Geometry D

Fig. 5.3: Various specimen geometries for finite element simulation

Table 5.1: Material properties of GFRP unidirectional lamina

Young's modulus in fiber direction, E_{11}	Young's modulus in other than fiber direction, $E_{22} = E_{33}$	Poisson's ratio between fiber and other direction, $\nu_{12} = \nu_{13}$	Shear modulus between fiber and other direction, $G_{12} = G_{13}$	Shear modulus between transverse and out-of-plane direction, G_{23}
46 GPa	13 GPa	0.26	7 GPa	4.48 GPa

Finite element (FE) simulations are performed by commercial FE software ABAQUS. 2D planar deformable shell structure is modeled for all geometries. Material definition comprises of GFRP 17500 grade material with 0.8 mm through-thickness $[0/90]_s$ cross-ply lay-up.

Fig. 5.4: Transverse stress distribution for 0° crack: (a) Geometry A; (b) Geometry B; (c) Geometry C; (d) Geometry D (original specimen boundaries are at red color, displacements are multiplied by 11 times for visualization purpose)

A through thickness crack of 45 mm length is analyzed in all geometries. An optimum specimen geometry with 45 mm crack is undeniably favorable for reduced crack lengths too, as corresponding COV values decrease along the ring of elements defining the far-field zone in those cases. The crack orientation angle over the simulation varies from 0° to 45° . As symmetric cross-ply laminate is used in the simulation runs, range of orientation angles from 0° to 45° ensures a study resembling crack orientation from 0° to 90° due to inherent symmetry situation in case of 45° crack. Similarly, transverse load is set at 10 kN which is the maximum load capacity that transverse actuator can transmit.

Fig. 5.5: Transverse stress distribution for 45° crack: (a) Geometry A; (b) Geometry B; (c) Geometry C; (d) Geometry D (original specimen boundaries are at red color, displacements are multiplied by 11 times for visualization purpose)

The optimal specimen designed with this higher load setting ensures that the specimen will be assuredly superlative for any lower load ratings. Thus selecting some extreme limits help reduce the number of variables for succeeding simulation runs. Through-

thickness blunt cracks (0.5 mm curvature radius) of 45 mm length and 1 mm width are included in the centre of the specimen. A boundary condition is applied to bottom end of axial direction by constraining all translational and rotational movements. Another boundary condition is applied to the left end of the transverse direction by constraining all movements except the axial translation. Shell edge loadings of 125 and 100 kN/m are applied to the top and right gripping zone of primary and secondary axis, respectively. 8-node, plane stress, thin shell, reduced integration, quadratic order element (S8R5) is chosen for discretizing the geometry. Plane stress, thin shell element matches well with real cruciform shape profile. Reduced integration needs less computation time and resources in comparison to full integration. Quadratic element increases accuracy in comparison to linear element.

Fig. 5.6: Axial stress distribution for 0° crack: (a) Geometry A; (b) Geometry B; (c) Geometry C; (d) Geometry D (original specimen boundaries are at red color, displacements are multiplied by 11 times for visualization purpose)

A mesh convergence study is performed duly and critical element shapes, meshing technique and meshing algorithm are chosen accordingly. Two circles are drawn by face partitioning encircling the crack tips to ensure uniform mesh element shapes within the whole circular zone. A very fine mesh is ensured at crack tip; for which local seeding is done in every 0.2 mm for mesh element generation.

Finally, developed longitudinal and transverse direction stress profiles over the gage area of all specimens are presented in Figs. 5.4-5.7. Corresponding displacements are multiplied 11 times of actual displacement for enhanced visualization. From stress profiles it is visualized that crack-tip stress concentration effect only spreads to a narrow zone. Thus, far-field stresses over all specimens show relative uniformity.

Fig. 5.7: Axial stress distribution for 45° crack: (a) Geometry A; (b) Geometry B; (c) Geometry C; (d) Geometry D (original specimen boundaries are at red color, displacements are multiplied by 11 times for visualization purpose)

Crack-tip zones are also the regions where the highest stresses develop; this indicates the biaxial failure initiating from the gage section of the test frame, as vivid from Figs. 5.8–5.9.

In neat specimens, attaining biaxial failure in the gage zone of cruciform specimens poses a significant challenge, as described earlier. However, in crack based optimal specimen design, far-field stress uniformity needs to be statistically evaluated as crack carries stress singularity issues and a successful cruciform geometry will be the one which ensures a uniform far field stress around the test area of interest prompting the failure to occur in biaxially loaded zone.

Fig. 5.8: Crack tip stress profile for 0° crack for Geometry A

To evaluate uniformity, mean and standard deviation of respective far-field stresses are determined. In horizontal direction, far-field element ring is taken at 25 mm distance from the crack tip and, in vertical direction, it is 40 mm away from the crack line. A smaller value of standard deviation about the stress values indicates a better uniformity.

Fig. 5.9: Crack tip stress profile for 45° crack for Geometry A

Fig. 5.10: Co-efficient of variance of far-field stresses in axial direction (A, B, C, D indicate various specimen geometry with white and grey columns indicating 0° and 45° crack, respectively)

A low coefficient of variance (COV) is an even better indicator of uniformity as it offers a ratio between standard deviation and mean stress. It appears from Fig. 5.10 that Geometry D carries the highest COV value for 0° crack both for primary load-frame and secondary load-frame axis stress; this can be attributed to the wider corner rounding of

this geometry very near the gage zone. For 45° crack also, its COV value is measurably higher; as evident in Fig. 5.11. Geometry C shows relatively lower COV values both for 0° and 45° crack in comparison to Geometry D. This can be implicated due to the presence of a relatively smoother corner fillet section. Geometry A, however, maintains the best horizontal and vertical stress uniformity in comparison to other geometries for both crack orientations. The COV values for 45° cracks are mostly higher than 0° cracks for all simulation runs which are not unexpected for an angle crack. It is also noteworthy for Geometry B that some loads are bypassing the central section towards the perpendicular arms which contribute to its higher stress variation in comparison to Geometry A.

Fig. 5.11: Co-efficient of variance of far-field stresses in transverse direction (A, B, C, D indicate various specimen geometry with white and grey columns indicating 0° and 45° crack, respectively)

Based on the simulation runs, Geometry A is found to possess better stress uniformity based on dimensional, loading and boundary constraints of the experimental test frame.

Above section has highlighted the key steps in optimizing a planar cruciform specimen geometry for biaxial characterization of fibre reinforced composite material. Final geometry is achieved with a corner rounding of 25 mm at the intersection of two perpendicular arms with a constant thickness throughout the body, as shown in Fig. 5.12. Provision of some supporting mechanisms is ensured in the grip zone to rigidly fix the specimen in load frame fixtures. An optimized geometry is thus far reached from simulation studies and material characterization of the composite laminate is studied subsequently.

Fig. 5.12: Final optimum composite specimen geometry

5.4 Biaxial experimentation

5.4.1 Determination of equivalent loaded area

During experimental phase, it is of foremost importance to determine the actual loaded cross-sectional area in gage section of cruciform specimen before starting any biaxial testing. Prime reason behind this lies in the presence of non-uniform cruciform geometry because of which significant portion of the load bypasses from the gage section towards the adjoining arm direction. If not carefully analyzed, this biaxial load sharing significantly affects the experimental analysis and results. Hence, cruciform specimen failure strengths are compared with uniaxially loaded specimen failure strengths on the basis that both specimens are supposed to fail once the applied stress surpasses the critical strength value of particular laminate sequence. In a typical uniaxially loaded specimen, gage area is known and applied ultimate load can be identified from the test data; thus uniaxial tensile failure strength can be determined according to ASTM standard D3039/3039M [105]. Cruciform specimen can be assumed to fail once the same critical stress is developed for similar laminate sequence. Thus the equivalent loaded area can be calculated directly from the failure strength as applied load is directly retrieved from test data recording software. Based on equivalent loaded area in longitudinal and transverse direction, the biaxial load ratio between the two directions can be achieved from applied ultimate load factor which is crucial for a non-symmetric loading condition like the present experimental setup. Quantifying the loaded area in this procedure also eliminates the need for strain instrumentations in neat biaxial specimen failure study.

Uniaxial tensile loading on standard and cruciform specimens are performed accordingly to determine the equivalent loaded area in both axial and transverse directions of specimen. Standard tensile test specimens are prepared following ASTM

test standard D3039/3039M [105]. Four ply symmetric cross-ply laminates with $[0^\circ/90^\circ]_s$ sequence are prepared by hand layup process. Laminates are then end tabbed using 8 ply GFRP laminate of 40 mm length and 25 mm width in top and bottom of each gripping end. Next, laminates are cured in Binder[®] convection oven (model no. Binder[®] ED 240) at a temperature of 130° C for two and half hours according to manufacturer's suggested curing cycle. Final uniaxial tensile test sample gage section is of 120 mm long, 25 mm width and 0.8 mm thick whereas each of two 40 mm long two end tab region thickness equals to 4 mm. Uniaxial testing is performed on an Instron 5569 machine under displacement-control loading with a crosshead speed of 1 mm/min. Test machine has an accuracy of 3% of indicated load and all important test data are recorded simultaneously.

(a)

(b)

Fig. 5.13: GFRP sample with $[0^\circ/90^\circ]_s$ laminate sequence: (a) specimen after failure under uniaxial tensile test; (b) stress-strain diagram of test specimens

Fig 5.14: Failed samples after uniaxial tensile failure (left sample is failed by horizontal direction lading; right one failed by transverse direction loading)

Failed specimens and corresponding stress-strain responses are shown in Fig. 5.13. Test samples show an initial linear stress-strain pattern up to 1% strain followed by non-linearity due to matrix material and final catastrophic failure (typical composite material failure behavior) occurs at strain value of 2.5%. The linear slope of 10 GPa is indicative of Young's modulus of cross-ply GFRP laminate and average ultimate failure strength equals to 288 MPa. Next, cruciform samples are prepared according to the cruciform geometry of Fig. 5.12. First, an aluminum mould sheet is prepared for blanking purpose by Hitachi CB75F 230 volt band saw cutting machine. Individual laminas are cut in specified dimension with aluminum mould from GFRP pre-preg roll

after defrosting overnight. Cross-ply $[0^\circ/90^\circ]_s$ laminates are prepared by hand layup and cured in convection oven according to the recommendation of GFRP roll manufacturer. This results in rectangular composite sheets of about 325 mm length, 205 mm width and 0.8 mm thick. Laminates are then machined to cruciform shape by high pressure water jet cutting (~ 650 MPa) to maintain all critical dimensions of the plan form with minimum damage to constituents of composite laminate.

Neat cruciform test specimens are then uniaxially tested in a 250 kN Instron 5985 uniaxial test machine to determine the equivalent loaded area. Two specimens are tested under uniaxial tensile loading along longitudinal and transverse direction of specimen. Final failed samples are shown in Fig. 5.14. Both failures occur in gage zone and the cracks proceed towards the boundary region of the specimen. Average failure loads for longitudinal and transverse direction tensile test measured to be 14.58 and 17.6 kN, respectively.

It can be assumed that ultimate tensile failure strength for both uniaxial test sample and cruciform specimen are same. As seen previously, average uniaxial tensile specimen failure stress was 288 MPa. Hence, equivalent loaded area in specimen longitudinal direction stands at,

$$A_{equiv. y} = 5.063 * 10^{-5} \text{ m}^2 \quad (5.1)$$

Similarly, equivalent loaded area in specimen transverse direction stands at,

$$A_{equiv. x} = 6.11 * 10^{-5} \text{ m}^2 \quad (5.2)$$

whereas transverse and longitudinal directions of the specimen are represented as x and y directions, respectively. From equivalent loaded area principle, individual axial and transverse direction stresses also can be calculated:

$$\frac{\sigma_x}{\sigma_y} = \frac{P_x}{P_y} * \frac{A_{equiv. y}}{A_{equiv. x}}$$

$$\frac{\sigma_x}{\sigma_y} = 0.83 * \frac{P_x}{P_y} = 0.83 * \lambda \quad (5.3)$$

where P_x , P_y stand for applied load in transverse and longitudinal direction of specimen, respectively; and λ indicates load ratio. Different biaxial load ratios will be applied in actual biaxial experimentation by changing the value of λ . The axial to transverse load ratios ranges from 1:0, 1:0.4, 1:0.8, 1:1. 1:0 load ratio indicates a uniaxial test case. With changing load ratios, stress ratios also vary from 1:0, 1:0.333, 1:0.667, 1:0.833.

5.4.2 Evaluation of notch sensitivity characteristics of GFRP laminate under uniaxial tensile loading

Generally, with increasing number of plies, GFRP composite material increasingly follows linear elastic fracture mechanics (LEFM) behavior and exhibits enhanced notch sensitivity [108]. Thus, critical stress intensity factor of the material plays a decisive role in the failure of such composite structures. However, with decreasing number of plies (say, in the range of 4, 8), same material demonstrates very wide range of behavior; some appearing as notch sensitive while others as notch insensitive [108]. Hence, notch sensitivity of uniaxial 4 ply cross-ply GFRP samples needs to be evaluated experimentally. Uniaxial composite samples are prepared accordingly by oven curing process and centre notches are machined in a CNC milling machine (model no. Mazak FJV-250) using special carbide drill bit (part no. CA-4270-5-1.0). 1 mm wide through

thickness central cracks of 6, 8, 10 and 12 mm length are machined in samples of ~24 mm width consequently. Tensile tests are performed according to ASTM D3039/D3039M standard in an Instron 5569 universal test machine under displacement control loading (with a crosshead speed of 1 mm/min) [105]. Associated ultimate failure loads of specimens with different crack lengths are highlighted in Fig. 5.15.

Fig. 5.15: Notch strength of [0/90]_s cross-ply GFRP composites: depicting notch sensitivity response

There are two ways in which a material with inherent stress-riser can fail – one is a notch-insensitive manner whereby net-section stress reaches the uniaxial failure strength and another is notch-sensitive manner whereby local stress-singularity effect near the crack tip governs the failure characteristics [109]. Net-section stress in a sample of width w containing a central crack or notch of length $2a$ is given by:

$$\sigma_{net} = \left(1 - \frac{2a}{W}\right) \sigma_t^{\infty} \quad (5.4)$$

where σ_t^∞ indicates failure strength of neat specimen under uniaxial tensile loading. In Fig. 5.15, net section stress given by Eq. (5.4) is plotted against the non-dimensional crack length to width ratio ($2a/W$). It shows that the GFRP cross-ply laminate behavior is far away from the ductile notch-insensitive response under uniaxial tensile loading i.e. specimen failure is governed by crack length.

5.4.3 Determination of fracture toughness properties of GFRP laminate

As the composite material is notch sensitive under uniaxial tensile loading, it is important to determine the fracture toughness properties of GFRP laminate before evaluating material failure under biaxial loading. There are 3 different modes associated with crack based failure – mode I (opening mode), mode II (sliding/in-plane shearing mode), mode III (tearing/out-of-plane shearing mode). Depending on various loading condition, specimen geometry and crack orientation, different modes of failures dominate either individually or collectively. In plane stress problems like the present specimen geometry, mode I, mode II and mixed mode (mode I/mode II) fracture toughness play crucial role. Mode I property is specially important in this case as this is usually the lowest one; hence, the most critical.

In orthotropic composite materials, mode I stress intensity is calculated by two opening modes; one is interlaminar opening mode which covers propagation of crack between two laminae within the same laminate; and another is translaminar opening mode that focuses on crack propagation in through thickness direction in the presence of an in-plane force. Out of the two, interlaminar opening mode is more critical as it carries a much lower critical stress intensity factor in comparison to translaminar opening mode. However, in an in-plane biaxial testing (likewise present cruciform based one),

translaminar opening mode has higher probability to occur due to loading condition. Accordingly, both interlaminar and translaminar mode I (crack opening mode) critical stress intensity factors $K_{IC,I}$ and $K_{IC,T}$ are determined herein.

Fig. 5.16: Load versus extension plot for specimens under double cantilever beam bending test

Interlaminar fracture toughness is determined by double cantilever beam (DCB) bending testing according to ASTM standard D5528 [110]. The specimens are prepared using 24 plies. A non-adhesive Polytetrafluoroethylene (PTFE) film of 0.05 mm thickness is inserted in mid plane (after 12 plies) during layup to serve as a starter pre-crack during experimentation. PTFE film is 63 mm long and 25 mm wide and is placed at one end of DCB specimen. Special type of PTFE film is used that avoid the problems associated with folding and crimping during actual experimentation. Eventually

specimens are cured in convection oven following the supplier's curing cycle. Final specimen dimensions are 125 mm * 25 mm * 4.8 mm. Three specimens are prepared accordingly and all are scrubbed with sandpaper followed by acetone washing before the experiment.

Two piano hinges are also attached in top and bottom section of the PTFE film end of each specimen with Hysol® EA 9309.3NA epoxy paste adhesive from Henkel Corporation. EA 9309.3NA is a two-part paste adhesive and is famous across the industry because of its high shear and peel strength between different combination of materials (e.g., between metal and composite). Two parts of this adhesive require well-mixing in a correct weight ratio before applying between piano hinges and specimen surface. Curing is performed at the end for 2 days at 77°F/25°C temperature. Herein, the challenge is to firmly join the piano hinge to laminate surface and, in the same time, not constraining the rotational movement of the hinge which will change the boundary condition during DCB experiment.

The specimens are marked in thickness direction in every mm with a permanent marker to understand the crack behavior under applied load. A displacement control loading of 1.0 mm/min is applied by clamping one end of each of the two piano hinges in a properly calibrated Instron 5569 machine. During experiment, delamination growth occurred in a slow stable extension mode. Load versus extension data are presented in Fig. 5.16 for all three samples. Finally, modified beam theory (MBT) is used to calculate the fracture toughness. MBT offers the most conservative values of $G_{IC,I}$ and, hence, it is used in the experiment. Mode I interlaminar fracture toughness of GFRP cross-ply composite laminate are determined for the samples based on the expression:

$$G_{IC,I} = \frac{3P\delta}{2W(a + |\Delta|)} \quad (5.5)$$

where P indicates load and δ indicates displacement. Accordingly,

$$G_{IC,I} = 0.90 \frac{\text{kN}}{\text{m}}$$

This can be converted to mode I stress intensity factor by using the relation $G_{IC,I} = K_{IC,I}^2/E$. Consequent mode I stress intensity factor stands at:

$$K_{IC,I} = 4.6 \text{ MPa}\sqrt{\text{m}}$$

For determining translaminar mode I stress intensity factor, linear elastic fracture mechanics (LEFM) principle is employed. Based on this, critical translaminar stress intensity factor of a 0° central crack in an infinite plate can be expressed in terms of ultimate failure stress (σ_f) as:

$$K_{IC,T} = \sigma_f \sqrt{\pi a} \quad (5.6)$$

A geometric correction factor is introduced in this closed form equation to comply with a finite plate boundary condition. The final expression of stress intensity factor follows as [117]:

$$K_{IC,T} = \sigma_f \sqrt{\pi a} \left[\frac{1 - \frac{a}{w} + 0.326 \left(\frac{2a}{w} \right)^2}{\sqrt{1 - \frac{2a}{w}}} \right] \quad (5.7)$$

Depending on finite plate equation, respective $K_{IC,T}$ values corresponding to various respective crack dimensions can be directly calculated from fracture failure stresses, as highlighted in Fig. 5.15. The fracture failure stress values are already presented in Fig. 5.15. Translaminar Mode I critical stress intensity factor for GFRP $[0/90]_s$ laminate is determined accordingly; which equals to $15.02 \pm 1.2 \text{ MPa}\sqrt{\text{m}}$.

5.4.4 Applicability of digital image correlation (DIC) system

Digital image correlation (DIC) system is used in biaxial experiment for full-field strain measurement. DIC is a non-contact strain measurement technique which offers the possibility to determine displacement and deformation field in the surface of an object only by comparing the test images before and after the deformation. Herein, the specimen surface needs to have sufficient distinct features that allow DIC to calculate deformation between a set of pixels. If the surface does not have distinguishable feature from point to point, it is painted or coated with spray to achieve a surface finish that eases the traceability of the point. A 12.4 Megapixel Sony α 700 digital single-lens reflex camera is used in the experiment due to its high image resolution (up to 4288*2566 pixels) and quick image frame grabbing feature (up to 5 frames per second). Around 40 test images are taken for each specimen during the testing to accurately capture the crack initiation and propagation behavior without any time lapse. A spray canister is used to apply speckle points on the specimen based on which DIC software will calculate the displacement and strain. A light source is used to illuminate the speckle points within the gage area. To eliminate picture grabbing problems associated with vibration, the camera is set securely on a tripod and a controller is used to remotely control the shutter function.

Vic-2D software from Correlated Solutions Inc. is used to obtain the displacement and strain patterns from the processed images. Before introducing the DIC for biaxial experimentation, a validation study is performed first to compare the response of DIC with strain gage based measurement system. For this, GFLA-3-350-70 strain gages (with gage length 3 mm and gage factor 2.07), supplied by Tokyo Sokki Kenkyujo Co. are mounted on specific location of three composite specimens. Strain measurements are

recorded with National Instruments data logger. Strain gage position is shown in Fig. 5.17 (a) and axial direction strain pattern (ϵ_{yy}) over the whole gage area is presented in Fig. 5.17 (b). The specimens are loaded uniaxially until the failure and final response from the strain gage and DIC are noted; as highlighted in Table 5.2.

Fig. 5.17: Strain measurement under uniaxial testing of GFRP specimen with crack by (a) strain gage; (b) DIC analysis

Table 5.2: Strain results measured by strain gage and DIC software		
Strain gage readings in axial direction, ϵ_{yy} (microstrain)	DIC reading in axial direction in same location, ϵ_{yy} (microstrain)	Difference between Strain gage and DIC reading
7605	6850	~ 11.03%
6892	6325	~ 8.96%
9064	8300	~ 9.21%

From two strain measurement techniques, it is understood that DIC system correlates well with strain gage readings within experimental scatter. The differences between respective reading values are less than 12%.

5.4.5 Biaxial testing on GFRP sample

5.4.5.1 Experimental description

During biaxial experimentation, glass fiber composite specimens are securely affixed to the test frame fixtures by fastener supporting mechanism. Specimens are ensured not to possess any twisting motion while mounting. A displacement control loading of 1 mm/min crosshead speed is maintained during testing. A load control ratio is set as an input to Bluehill 3 software at the outset of experiment. Instron 5985 uniaxial machine pulls the specimen in axial direction by predefined displacement loading rate and associated load sensor sends sequential feedback to the system about grip area load reading in every 0.01 second.

Fig. 5.18: Optimal cruciform specimen, as firmly secured before starting of experiment

The feedback information is used to apply a transverse load to the cruciform specimen by the pre-assigned load ratio. Transverse grip load cell follows the load reading in x axis direction every 0.01s and Bluehill 3 software tabulates all readings at successive

interval. A test stop criterion is set at either 40% drop in axial force or transverse force value reaching a limit of 9.75 kN. Fig. 5.18 shows a cruciform specimen firmly fixed to the axial and transverse direction of the test frame.

To generate a fracture failure boundary on K_I , K_{II} space, it is crucial to perform a failure test under equi-biaxial tensile loading (with stress ratio 1:1). However, because of loading constraint of 10 kN in transverse direction, achieving a biaxial failure under true equi-biaxial loading is elusive. Still, during initial part of experimental phase, it is highly endeavored to achieve a failure under 1:1 loading condition. In this case, most of the failures occurred in the grip area and to eliminate this problem, grips are strengthened by including end tabs manufactured with GFRP composite prepregs.

Both GFRP end tabs and main laminate are prepared by hand up and succeeding oven curing process. Main laminate consists of $[0/90]_s$ sequence whereas GFRP end tabs is built on $[0/90]_{2s}$ sequence. Total 8 end tabs are prepared for each specimen each one of which consisted of 8 ply $[0/90]_{2s}$ laminate. Thus the grips are reinforced with 20 layers instead of 4. Finally, through-thickness centre notches of 40 mm and 45 mm length with 0° , 45° orientations are introduced in all specimens by CNC milling. Proposed test matrix for studying biaxial material response of cross-ply GFRP laminate is listed in Table 5.3. Two specimens are experimentally evaluated for each of the test cases.

Table 5.3: Test matrix for biaxial failure response of $[0/90]_s$ GFRP composite material

Crack length	40 mm		45 mm	
Crack orientation (with respect to transverse direction)	0°		45°	
Transverse to axial load ratio	0:1	0.4:1	0.8:1	1:1
Transverse to axial stress ratio	0:1	0.333:1	0.667:1	0.833:1

5.4.5.2 Load versus displacement plots of biaxial testing

Figures 5.19-5.22 highlight load versus displacement plots for 40 mm 0° , 40 mm 45° , 45 mm 0° and 45 mm 45° crack, respectively. The axial and transverse load and axial displacement data are taken from the test recorder whereas transverse displacement is calculated from displacement loading rate multiplied by time. All pre-cracked GFRP 17500 grade cruciform specimens experienced sudden catastrophic failure after maintaining a straight gradient up until the break point. This resembles brittle failure behavior of the laminate under biaxial testing with sudden drop in load bearing property after the crack starts to propagate.

Fig. 5.19: Load-displacement plot for 40 mm 0° crack under biaxial tension-tension loading

Fig. 5.20: Load-displacement plot for 40 mm 45° crack under biaxial tension-tension loading

Fig. 5.21: Load-displacement plot for 45 mm 0° crack under biaxial tension-tension loading

Fig. 5.22: Load-displacement plot for 45 mm 45° crack under biaxial tension-tension loading

Fig. 5.23: Crack propagation direction under biaxial tension-tension loading – (a) for 0° crack; (b) for 45° crack

The crack propagation direction of two samples (one with 0°, another with 45° crack) is shown in Fig. 5.23. Similar propagation direction is observed for all specimens tested with varying load ratio and crack length.

5.4.5.3 Failure surface on the basis of axial and transverse stress

Transverse failure stresses are plotted against axial failure stresses in Fig. 5.24 for 0° crack for set load ratios. It appears from ultimate failure stresses that in the presence of transverse tensile load, specimen can withstand a higher amount of axial tensile stress in comparison to uniaxial test case; as illustrated in Figs. 5.24, 5.26-5.27. Dotted line indicates failure stresses of 90° crack based samples; derived by applying equivalent area principle based on failure stresses of 0° crack [see Fig. 5.24].

Fig. 5.24: Transverse versus axial stress diagram of biaxially loaded [0/90]_s GFRP cruciform specimen with 0° cracks of varying dimension [dotted line indicates failure stresses of 90° crack based samples]

From 1:0 to 1:1 load ratio, axial stresses rise by around 20% for both 40 mm and 45 mm 0° cracks whereas the transverse stresses increase from 0 MPa (for 1:0 load ratio) to 138 MPa and 126 MPa in 1:1 load ratio for 40, 45 mm crack, respectively [see Fig. 5.27]. The failure surface appears a linear pattern for axial to transverse load ratio starting from 1:0 to 1:1 load condition for 0° crack. Other load conditions in axial to transverse ratio from 1:1 to 0:1 is not tested due to loading constraint in transverse direction which can only apply up to 10 kN load. It is understood that with increasing transverse to axial load ratio more than 1:1, crack closure force will be more dominating rather than the crack opening forces. Hence, for transverse to axial load ratio more than 1:1, cross-ply GFRP laminate is likely to fail at significantly higher load than 10 kN.

Fig. 5.25: Transverse versus axial stress diagram of biaxially loaded [0/90]_s GFRP cruciform specimen with 45° cracks of varying dimension

Gradient line between these two loads is always positive within 1:0 to 1:1 load ratio; this indicates that for 0° crack and for a symmetric cross-ply specimen, biaxial tensile failure will always occur at higher stresses than uniaxial failure, i.e., material will attain higher load bearing capability under tension-tension loading space [see Fig. 5.20]. Similar results are witnessed in several other studies where circular holes are cut in graphite/epoxy laminate plates of different orientations. Everywhere, the ranges of strength reduction ratio of notched biaxial to un-notched uniaxial strength appear higher than the corresponding uniaxial loading values [51-53].

Both 40 mm and 45 mm crack show comparable ultimate failure values of axial and transverse stress. But stress values associated to 40 mm crack are higher in comparison to 45 mm because of shorter crack length dimension which is expected due to associated changes in load-bearing capacity with lower crack length. An increase in crack length by 12.5% reduces both axial and transverse failure stress by 7% which signifies a significant decrease in material strength.

Fig. 5.26: Axial stress versus transverse-axial stress ratio for biaxially loaded $[0/90]_s$ GFRP specimen with varying crack profiles

Fig. 5.27: Transverse stress versus transverse-axial stress ratio for biaxially loaded $[0/90]_s$ GFRP specimen with varying crack profiles

Next, biaxial tests are repeated for cracks oriented at 45° angle with respect to transverse direction. Ultimate axial and transverse failure stresses are highlighted in Fig. 5.25. As in the case of 0° crack, 45 mm cracks also failed at lower load in comparison to 40 mm ones. In another scenario, axial load increases by about 15% from 1:0 (uniaxial) to 1:1 load ratio (equi-biaxial), both for 40 mm and 45 mm crack; as it is vivid in Fig. 5.26. Transverse stress also rises with increasing of its share in the load ratio, as highlighted in Fig. 5.27. Ultimate axial and transverse failure stresses corresponding to angle cracks are always higher than 0° cracks in all load cases. This is an expected phenomenon in case of an angle crack whereby it actually experiences a mixed mode fracture due to the presence of both crack opening and sliding modes.

Finally, as the gradient line between two axial and transverse stresses always appears positive; this necessitates a higher load for specimen failure under biaxial tensile-tensile loading in comparison to uniaxial case. This is attributed to crack closure mechanism brought by transverse tensile loading in experimentation, as highlighted in Fig. 5.28. The greater the transverse force, the stronger crack closure force is experienced by the laminate. Thus, corresponding axial load rises even higher to open the crack and thus to initiate failure [see Figs. 5.26-5.27].

Fig. 5.28: Behavior of crack under (a) transverse tensile stress; (b) transverse compressive stress

Cracks oriented at 45° create a symmetric condition between two adjoining arms in this series of testing. Hence, the failure results associated with load ratio of 1:0, 1:0.333, 1:0.667 can be applied for other reciprocating load ratios of 0.667:1, 0.333:1 and 0:1. Thus, a complete first quadrant tensile-tensile failure plot is achievable in a 45° crack without testing the load cases in which the transverse stress exceeds the axial stress; as illustrated in Fig. 5.25. Ultimate failure boundary for 45° crack represents almost a linear pattern like 0° crack boundary.

5.4.5.4 Determination of geometrical factor relevant to cruciform specimen for fracture characterization

In this section, it is evaluated whether crack is playing a major role in laminate failure in case of biaxial loading. Earlier, as seen in uniaxial tensile loading study, specimen failure was occurring in a notch-sensitive way. This establishes the length of crack as a crucial governing parameter in uniaxial tensile loading. Biaxial failure also should follow the similar trend as it only depends on material, laminate and crack characteristics rather than how and in which direction load is applied.

In this connection, specimen shape is important factor as it greatly affects the critical fracture toughness properties. Based on different specimen shape, different geometric factor is introduced in critical stress intensity factor expressions. Accordingly, the geometric factor relevant to cruciform specimen also needs to be determined to correctly measure the critical mode I and mode II stress intensity factors. Finite element (FE) simulation is used to determine the geometric factor associated with cruciform shape. Various load ratios are introduced in both axial and transverse direction to numerically calculate both mode I and mode II stress intensity factors. FE is performed in ABAQUS using 2D planar deformable shell structure.

Blunt cracks of 45 mm length and 1 mm width are simulated with 0° and 45° orientation with respect to transverse direction of specimen. Boundary condition is applied to stationary ends of primary and secondary axis as like optimized cruciform geometry based FE simulation. Shell edge loadings of various ratios are applied to the top and right gripping zone of primary and secondary axis, respectively. These include axial to transverse shell edge load ratio of 1:0, 1:0.25, 1:0.5, 1:1, 1:2, 1:4, 0:1. Meshing is done as like previous optimized cruciform geometry based FE simulation.

Corresponding mode I and mode II fracture toughness for cruciform geometry can be expressed as:

$$\begin{aligned} K_I &= Y_1 \sigma_{yff} \sqrt{\pi a} \\ K_{II} &= Y_2 \sigma_{yff} \sqrt{\pi a} \end{aligned} \quad (5.8)$$

where, Y_1, Y_2 indicates non-dimensional geometric correction factor associated with K_I and K_{II} , respectively and σ_{yff} denotes uniform stress as reached away from crack tip. Values of K_I, K_{II} are evaluated from simulation results by linear elastic fracture mechanics (LEFM) principle based on Eq. (5.2).

Fig. 5.29: Y_1 for 0° crack from FE simulation

Maximum tangential stress crack initiation criterion is used in the simulation. Average data of 3rd, 4th and 5th contour integral are used for K_I , K_{II} calculation. First contour integral is calculated based on all elements inside the crack front and one layer of elements outside the crack front. Every next contour integral value is computed by adding a single layer of elements to the region defined by previous region. Based on different values of mode I and mode II fracture toughness, corresponding far-field stresses and crack lengths, Y_1 vs. transverse to axial stress ratio plot is achieved, as shown in Fig. 5.29. It appears from Fig. 5.29 that the value of Y_1 linearly decreases with increase in transverse to axial stress ratio; i.e., higher transverse vs. axial stress ratio carries a lower Y_1 value for cruciform geometry with a 0° crack. Similarly, Y_1 vs. transverse to axial stress ratio and Y_2 vs. transverse to axial stress ratio plots are achieved for 45° crack; as highlighted in Fig. 5.30.

Fig. 5.30: Y_1 , Y_2 for 45° crack from FE simulation

Next, based on experimental transverse to axial stress ratios, values of respective Y_1 and Y_2 are taken from the figures corresponding to FE simulation for appropriate crack orientation (Figs. 5.29-5.30). Then mode I and mode II fracture toughness are calculated according to Eq. (5.8). Corresponding mode I stress intensity factor is highlighted for 0° crack with respect to transverse to axial stress ratio in Fig. 5.31. It appears that fracture toughness for both 40 mm and 45 mm 0° crack are ~16 MPa.m^{0.5} which appears to be comparable with uniaxial testing mode I stress intensity factor values.

Fig. 5.31: Mode I fracture toughness versus transverse - axial stress ratio for 0° cracks of varying dimension

Fig. 5.32: Mode I versus mode II fracture toughness of 45° cracks of varying dimension

Next, corresponding calculations are done for 45° crack for determining both mode I and mode II fracture toughness. Final mode I and mode II fracture toughness are shown in Fig. 5.32. With increasing transverse to axial load ratio, K_I increases from 6.07 (uniaxial testing) to 20.16 MPa.m^{0.5} (1:1) whereas K_{II} decreases from 10.92 MPa.m^{0.5} to -

2.35 MPa.m^{0.5}. This matches well with relevant stress transformation equations and corresponding K_I , K_{II} calculation.

5.4.5.5 Digital image correlation (DIC) results

As mentioned, digital image correlation works on the basis of the images grabbed during the experiment and accordingly measures the deformation between distinct points of the surface.

Fig. 5.33: Comparison between DIC and experimental result

Vic-2D DIC software is used to analyze the deformation distribution by comparing all collated images taken at regular interval with reference to the pre-experiment image. The axial displacement data is taken from machine readings and it is compared with DIC data taken at the gage boundary for all specimens, as illustrated in Fig. 5.33. Both of these axial displacement results appear to be comparable for various load cases.

(a)

(b)

Fig. 5.34: Axial strain distribution over whole gage section - (a) specimen loaded with axial to transverse stress ratio 1:0.667; (b) specimen loaded with axial to transverse stress ratio 1:0.333

Figure 5.34 shows axial strain data in specimen failure based on DIC analysis. From figure it is vivid that high strain concentration occurs around the crack tip before failure; this represents an accurate crack initiated failure. In addition, there are other regions of the specimen subjected to high levels of strain as denoted by the yellow and green patches. Similar strain distribution is observed for all other specimens, both for 0° and 45° crack. This matches well with experimental failure modes where crack initiation dictates the failure mechanism in each individual specimen once stress intensity factor reaches the critical values.

5.5 Summary

This chapter presents fracture characterization of composite material under experimental biaxial loading, as been undertaken in line with the development of a biaxial test frame in NTU. The chapter first introduces an optimal cruciform geometry based on finite element (FE) simulation. Final geometry is achieved with a corner rounding of 25 mm at the intersection of two perpendicular arms with a constant thickness profile throughout the body. Next, few uniaxial testings on uniaxial and cruciform specimens are performed to determine equivalent loaded area in both axial and transverse directions of specimen. Afterwards, notch sensitivity behavior of $[0/90]_s$ cross-ply GFRP material is determined under uniaxial tensile failure. This establishes the length of crack as a crucial governing parameter for failure response of the GFRP laminate sequence. Critical fracture toughness properties are determined consequently following respective test standards. Mode I properties are specially important as this is usually the lowest one in comparing to others and, hence, the most critical.

In actual experimental biaxial testing, two types of crack orientations (0° , 45°) are studied, both with two different crack lengths (40 mm, 45 mm). Gradient lines between corresponding axial and transverse loads are always found positive; this means for 0° crack and for a symmetric cross-ply specimen, biaxial tensile-tensile failure will always occur at higher stresses than uniaxial failure; i.e., material will attain higher load bearing capability under tensile-tensile loading scenario. In another scenario, ultimate axial and transverse failure stresses corresponding to angle cracks are always found higher than 0° cracks for all load cases. This is an expected phenomenon in case of angle crack whereby it actually experiences a decreased stress value. Similar crack tip strain behavior is visible in all load cases, and both for 0° and 45° cracks, which matches well with experimental failure modes as crack initiation dictating the failure criterion in each individual specimen once stress intensity factor reaches the critical values.

Appropriate geometry factor relevant to cruciform shape is determined simultaneously. Once this geometric factor is considered, stress intensity factor values both from uniaxial and biaxial testing match with each other.

Chapter 6

Failure mode map for tubular columns under eccentric loading

6.1 Introduction

This chapter evaluates various compressive failure modes on a tubular column in case of eccentric loading. These failure modes include global buckling, local buckling, yield or onset of plasticity and axial crushing. Global buckling, local buckling and yield failure are known to occur in eccentric condition. The critical failure loads for these modes are available in literature. However, axial crushing is still to be explored in case of eccentric loading. Hence, this study first analyzes whether axial crushing failure is possible in case of eccentric loading or not. Later, it focuses on developing a failure map based on major compressive failure modes in case of eccentric loading and consequently it proposes mass minimal equations with respect to load and shape factor.

In real world, many structures experience non-uniformly distributed loading more frequently than uniformly distributed one. Wind turbine tower is a typical example of this scenario whereby nacelle weight maintains an eccentricity to the tower neutral axis.

This eccentric compressive loading shifts the failure boundary regions and subsequent optimal design issues in comparison to uniformly distributed loading condition.

6.2 Axial crushing under non-uniformly distributed loading

6.2.1 Application of non-uniformly distributed loading

Applying non-uniform loading on a tubular structure based on a usual universal testing machine is not straightforward. Uniaxial mechanical characterization testing frames only provide uniform axial loading for one axis. Hence, some adjustments were made to a three-point bending jig to apply non-uniform loading by a uniaxial machine. For this, a three-point bending jig of 100 kN load bearing capacity is used with one bottom roller removed during experimentation. The jig is then fixed to the top fixture with its another bottom roller lying upon a small solid metal piece. The stainless steel metal piece ensures high stiffness properties and carries a negligible weight, so as to behave like a rigid body. This metal piece remains on top of the specimen to transfer the loading from the test frame to the specimen. A schematic diagram of the experimental setup is shown in Fig. 6.1.

Finite element (FE) simulations are conducted simultaneously in ABAQUS to numerically verify whether the experimental fixture can generate a non-uniformly distributed loading. A thick solid plate is used on top of aluminum alloy based solid cylindrical model (with elastic modulus of 70 GPa, Poisson's ratio 0.35). The solid plate is made of steel alloy (with elastic modulus of 200 GPa, Poisson's ratio 0.3) and it works as a rigid body on top of the cylindrical model which is a deformable one (elastic-plastic solid). A surface interaction is created between rigid and deformable body to ensure in-between sliding where the master surface remains in the rigid body and the slave surface remains in the deformable body.

Fig. 6.1: Schematic diagram of experimental set-up

A pressure loading of 10 MPa is applied at various eccentricities. Figure 6.2 shows the pressure loading at eccentricity ratio ($\bar{e} = e/r$) 0.25, where e and r indicate eccentricity of loading axis from neutral axis and radius of the cylinder, respectively. Boundary condition is applied in the bottom of aluminum cylinder by fixing all translational movements in vertical (y) and out-of-plane (z) direction. Meshing is done with 3D hexagonal stress element (C3D8R); it is an 8 node linear brick, reduced integration

element. Total 4456 elements are used in this simulation. The FE model geometry is highlighted in Fig. 6.2.

Fig. 6.2: Finite element simulation model of non-uniformly distributed loading

Fig. 6.3: Non-uniformly distributed stress from FE simulation with eccentricity ratio $\bar{e} = 0.33$; $\bar{e} = 0.66$

Various eccentricity ratios are studied accordingly and simulations results of $\bar{e} = 0.33, 0.66$ are presented in Fig. 6.3. The negative stresses in these figures indicate

compressive stress values which vary from one end to another diametric end of specimen top section. Based on simulation runs, it is visualized that the setup is capable of applying non-uniformly distributed loading.

6.2.2 Specimen preparation

Henceforward, a set of aluminum shell structures are prepared from long aluminum tube of alloy grade 6063-T5. At the outset, it was a 6 m long tubular structure with 34 mm outer diameter and 1.05 mm thickness. Small pieces of 124 mm length are prepared out of it by shear cutting machine. This resulted in almost desired length but with rough edges in the end section with respect to specimen axis. Though steel is the major component of wind turbine tower till today; yet, an aluminum tubular structure is studied herein on the basis that aluminum with lower stiffness and yield strength in comparison to steel will fail earlier to the latter; thus the experiment can be carried on for different eccentricity ratio with a 50 kN bending jig; as was available in the lab.

To get further accuracy in end face finishing, tubes are further processed in a Harrison M300 manual lathe machine. Here, two ends of 124 mm long pieces are shortened further to almost 120 mm by reducing 2 mm from each side for needed precision. This is done in agreement with ASTM compressive testing standard of metals (ASTM E9-09). This ensures perpendicular end face to specimen axis which is important in axial crushing study. All specimens eventually possess almost equal dimension of 120 mm length, 34 mm outer diameter and 1.05 mm thickness.

The stainless steel metal piece is simultaneously prepared by subtractive machining process using LAGUN FU-1250 manual milling machine. Steel metal piece acts as a rigid body and converts the line load of bending jig into a distributed load. The bottom

roller of bending jig can be translated up to few centimeters. By translating the roller, necessary offset (i.e. distance between centre of the specimen and loading point) is maintained to apply both uniformly distributed and non-uniformly distributed loading. Thus the bending jig converts the Instron test-frame's uniform loading into an eccentric, concentrated line loading and, afterwards, the rigid body in contact to the bending jig converts the same into a non-uniformly distributed loading.

6.2.3 Experimental procedure and data analysis

After machining stage, the axial crushing testing is initially performed for a uniform load case on an Instron 5569 uniaxial test machine. Here, the objective is to study the axial crushing failure pattern under uniform loading. The specimen was loaded under displacement control at a rate of 2 mm/min. The test machine has an accuracy within 3% of indicated load. Important test data are recorded using Instron test-frame's software. The observed failure mode is shown in Fig. 6.4.

(a)

(b)

**Fig. 6.4: Axial crushing of Al 6063-T5 specimen under uniformly distributed loading
– (a) top view; (b) front or side view**

The highlighted failure mode represents a true axial crushing failure (non-axisymmetric or diamond mode) for aluminum alloy specimen; as observed in literature [111]. Next, non-uniformly distributed loading is applied by moving the loading roller off-axis at an eccentricity (e) of 2 mm, 4 mm, 6 mm and 8 mm.

(a)

(b)

(c)

(d)

Fig. 6.5: Top view of failed samples – with eccentricity of 2 mm (a, b); (b) with eccentricity of 8 mm (c, d)

(a)

(b)

(c)

(d)

Fig. 6.6: Front view of failed samples - with eccentricity of 2 mm (a, b); (b) with eccentricity of 8 mm (c, d)

The failure modes are highlighted in Figs. 6.5-6.6 which resemble partial end folding in only highly loaded top sections of tubular specimen. Total 8 specimens are tested with 2 mm/min displacement loading rate and all of the specimens showed same mode of

failure in top section. Associated load versus displacement curves with 2 mm and 8 mm eccentricity are presented in Fig. 6.7. These represent an eccentricity ratio ($\bar{e} = e/r$) of 11.76% for 2 mm and 47.06% for 10 mm. The plot is initially linear up to a peak load at point A, then load decreases with increasing deformation until point B, then due to interaction effect between deformed zones, load again increases until point C [see Fig. 6.7].

Fig. 6.7: Load-displacement plot from non-uniformly distributed compressive testing of shell structures

It is evident from failure modes that these do not represent any axysymmteric (ring) or non-axysymmteric (diamond) mode failure. To analyze the failure profiles further according to existing knowledge base, the yield strength of the material is compared

with experimental failure stress associated with peak load of Fig. 6.7. The critical axial crushing failure load for a thin shell structure can be represented empirically as [111]:

$$P_{crush} = 4\pi r t \sigma_{ys} * \psi^{0.7} \quad (6.1)$$

where σ_{ys} indicates the yield strength of the material, t indicates thickness of specimen and ψ indicates solidity ratio $\left[with \psi = \frac{2t}{r}\right]$.

On the other hand, based on combined bending and compressive failure of tubular specimen, the critical failure load can be represented as:

$$P_{combined} = \frac{2\pi\sigma_{ys}r^2t}{r + 2e} \quad (6.2)$$

Fig. 6.8: Yield strength measurement of shell specimen – (a) Setup with averaging extensometer of 50 mm gage length; (b) Initial and failed condition of Al 6063-T5 sample

The yield strength of the material is accordingly measured by testing aluminum specimens of same grade material. Few aluminum test specimens are prepared following the specifications of ASTM E8/E8M standard [112]. Curved tensile test specimens are machined using electrical discharge machining (EDM) from 300 mm long tubular pieces. Final tensile specimens are 50 with mm gage length (extensometer gage length), 12.8 mm width and 1.05 mm thickness. An averaging type extensometer is used for strain measurement as it is recommended for tube sections. The specimen is subjected to load control at a rate of 1.15 MPa/sec in Instron 5569 machine according to the above-mentioned standard. The test machine has an accuracy within 3% of indicated load. The failed samples are shown in Fig. 6.8. The associated stress versus strain responses are plotted in Fig. 6.9 which show typical elastic-plastic behavior.

Fig. 6.9: Stress-strain plot from tensile test experiment for Al 6063-T5 specimen

The stress-strain diagrams indicate a yield strength value of at least 190 MPa. This affirms that combined bending and compressive failure is occurring rather than axial crushing failure; as expressed in Eqs. (6.1) and (6.2) which means no axial crushing failure occurs in case of non-uniformly distributed loading.

6.3 Failure mode map under non-uniform compressive loading

This study, henceforward, evaluates a failure plot for thin shell structures based on all competitive load cases as prevalent in case of eccentric, non-uniform compressive loading. Many structures, in their operation life, frequently encounter various modes of eccentric compressive loading. A competitive failure diagram based on structural parameters will help designing the structures with optimal requirements. Efficient shaped section is a good design principle in this connection. An efficient shape ensures a design with the least material for a given loading condition. The shape design, however, varies based on material's maximum attainable shape efficiency and manufacturing constraints.

Manufacturing of many different shapes is possible in case of metals; but for other structural material families such as for composite materials, the manufacturing constraints play a wider role due to complexity in processing and manufacturing routes. Similar issue holds applicable for wood and other natural composite materials. Manufacturing constraints also play a bigger role for thinner and narrower sections. For example, often thickness needs to be maintained larger than a minimum value which may not be possible with all material families. However, an overall failure diagram representing every mode of failure with their associated structural limits serves as a basis for optimal design.

As no axial crushing failure is available for eccentric compressive loading cases, only three prominent compressive failure modes are considered for failure map generation; these are global buckling, yield failure and local buckling. A companion flow-chart is presented in Fig. 6.10 for reference.

Fig. 6.10: Flow chart of failure mode map and mass minimal design under eccentric loading

A thin shell structure in axial compression can fail either by column buckling (buckling in a wavelength related to overall length) mode or by local or shell buckling (buckling in a wavelength related to section) mode or by onset of plastic failure (influenced by the yield strength of the material). Out of this, column buckling mode is greatly influenced by end condition which can be expressed in terms of effective length. Column

straightness also plays a key role in buckling. Contrarily, local buckling is mostly affected by shape imperfection issues.

The failure load equations relevant to various compressive failure modes are presented in next section. Due to non-uniformity in loading, additional eccentricity factors are considered in load equations.

6.3.1 Global buckling

Let us consider a simply-supported column of length L under an eccentric axial load P . The load P is applied at an eccentricity e from the central column neutral axis. Normal stress in the column results from both axial load P and bending moment M due to eccentricity (e) and the maximum stress develops at the extreme distance from column neutral axis. Accordingly, the maximum stress can be represented as [113]

$$\sigma_{gb} = \frac{P}{A} + \frac{M_{max}C}{I} = \frac{P}{A} + \frac{P(w_{max} + e)C}{I} \quad (6.3)$$

where w_{max} represents the maximum transverse displacement, A is the cross-section area, I is the moment of inertia of the cross section and C is the distance from the neutral axis to the concave side extreme fiber of the column. The maximum transverse displacement of the column occurs at mid-section of the column length. Substituting this expression of $w_{max} = e \left[\sec \left(\frac{\pi}{2} \sqrt{\frac{P}{P_{cr}}} \right) - 1 \right]$ in Eq. (6.3), buckling stress can be given by:

$$\sigma_{gb} = \frac{P}{A} \left[1 + \frac{A}{I} ec \left\{ \sec \left(\sqrt{\frac{P}{EI}} \frac{L_{eff}}{2} \right) \right\} \right] \quad (6.4)$$

$$\sigma_{gb} = \frac{P}{A} \left\{ 1 + \frac{ec}{r_g^2} \sec \left(\frac{L_{eff}}{2r_g} \sqrt{\frac{P}{EA}} \right) \right\} \quad (6.5)$$

where r_g ($r_g = \sqrt{I/A}$) represents the radius of gyration.

The secant formula indicates that in addition to axial load P and cross-section area A , the maximum column buckling stress also depends on the eccentricity ratio ec/r_g^2 .

Global buckling equation finally stands as

$$\sigma_{gb} = \frac{P}{A} \left\{ 1 + \frac{2e}{r} \sec \left(\frac{L_{eff} \sqrt{P}}{2\sqrt{EI}} \right) \right\} \quad (6.6)$$

According to secant expansion formula:

$$\sec x = 1 + \frac{x^2}{2} + \frac{5x^4}{24} + \dots \quad (6.7)$$

Accordingly, by taking only first two terms of secant function, σ_{gb} can be approximated as:

$$\begin{aligned} \sigma_{gb} &= \frac{P}{A} \left\{ 1 + \frac{2e}{r} \left(1 + \frac{1}{8} \frac{L_{eff}^2 P}{EI} \right) \right\} = \frac{P}{A} \left\{ 1 + \frac{2e}{r} + \frac{2e}{8r} \frac{L_{eff}^2}{EI} \frac{\pi^2 EI}{L_{eff}^2} \right\} = \frac{P}{A} \left(1 + 4.47 \frac{e}{r} \right) \\ \sigma_{gb} &= \frac{P}{A} (1 + 4.47 \bar{e}) = \sqrt{\frac{PE\pi\phi}{4L_{eff}^2}} (1 + 4.47 \bar{e}) \end{aligned} \quad (6.8)$$

where $\phi = r/t$ and $\bar{e} = e/r$

6.3.2 Yielding

Next, the onset of plasticity can be expressed by material's yield strength property as:

$$\sigma_y = \sigma_{ys} \quad (6.9)$$

where σ_{ys} represents material's yield strength.

6.3.3 Local buckling

Local or shell buckling mode can be expressed as [114]:

$$\sigma_{lb} = \frac{\alpha}{\gamma} \frac{E}{\sqrt{3(1-\nu^2)}} \frac{1}{\phi} \quad (6.10)$$

where E represents material's Young's modulus, ν represents Poisson ratio, α indicates shell strength reduction factor and γ represents an additional safety factor. Shell strength reduction factor (α) covers the effect of buckling load reduction due to impending imperfection of support structure upon the foundation. Due to post-buckling behavior, the cylindrical shells are also subjected to meridional compressive stresses; this is covered by an additional safety factor of $\gamma = 1.33$. In presence of bending and buckling load, the shell strength reduction factor (α) is calculated as:

$$\alpha = \frac{\alpha_c \sigma_c + \alpha_b \sigma_b}{\sigma_c + \sigma_b} \quad (6.11)$$

where σ_c and σ_b indicate uniform compressive stress due to design axial load and maximum compressive stress due to design bending moment, respectively. And α_c and α_b indicate compressive and bending reduction factors:

$$\alpha_c = \frac{0.83}{\sqrt{1 + 0.01\phi}} \quad (6.12)$$

$$\alpha_b = 0.1887 + 0.8113\alpha_c \quad (6.13)$$

Here ϕ indicates a ratio between radius and thickness of shell structure; eventually, it matches with elastic bending shape factor (ϕ_B^e) of the structure which signifies the stiffness ratio between a shaped section and a solid circular section of same material, length and cross-sectional area. According to published literature, typical ϕ ratio for a wind turbine tower stands in the range of 15-40 [72]. Based on this, compressive

reduction factor α_c varies from 0.63 to 0.73. A α_c value of 0.68 is assumed for next calculation.

$$\begin{aligned}
 \alpha &= \frac{0.68\sigma_c + 0.74\sigma_b}{\sigma_c + \sigma_b} \\
 \sigma_c &= \frac{P}{A} = \frac{P}{\pi Dt} \\
 \sigma_b &= \frac{MC}{I} = \frac{4Pe}{\pi D^2 t} \\
 \alpha &= \frac{0.68PD + 2.96Pe}{PD + 4Pe} = 0.68 \frac{1 + 2.18 \frac{e}{r}}{1 + 2 \frac{e}{r}} = 0.68 \frac{1 + 2.18\bar{e}}{1 + 2\bar{e}} \quad (6.14)
 \end{aligned}$$

Eventually, local/shell buckling stress (σ_{lb}) can be expressed as:

$$\sigma_{lb} = 0.295 \frac{1 + 2.18\bar{e}}{1 + 2\bar{e}} \frac{E}{\sqrt{1 - \nu^2}} \frac{1}{\phi} \quad (6.15)$$

6.3.4 Determination of boundary line for different eccentricity

To represent failure mode chart in a uniform scale, it is necessary to define the boundary zones of each failure mode in respect to structural and loading parameters. Hence, the boundary zones of three aforementioned failure modes are determined accordingly by equating all the above-mentioned failure stresses of Eqs. (6.8)-(6.9) and (6.15). All boundary line equations are finally expressed in terms of two non-dimensional factors: structural loading coefficient or load factor ($P/\sigma_{ys}L^2$), and shape factor (ϕ).

Accordingly, the boundary line equation between global buckling and yield can be expressed as:

$$\frac{P}{\sigma_{ys}L_{eff}^2} = \frac{4}{\pi} \frac{\sigma_{ys}}{E} \frac{1}{\phi} \left(\frac{1}{1 + 4.47\bar{e}} \right)^2 \quad (6.16)$$

The boundary line equation between global buckling and local buckling can be expressed as:

$$\frac{P}{\sigma_{ys} L_{eff}^2} = \frac{4}{\pi} \frac{E}{\sigma_{ys}} \frac{1}{1-v^2} \frac{1}{\phi^3} \left\{ \frac{0.295(1+2.18\bar{e})}{(1+4.47\bar{e})(1+2\bar{e})} \right\}^2 \quad (6.17)$$

The boundary line equation between yield and local buckling can be expressed as:

$$\phi = \frac{E}{\sigma_{ys} \sqrt{1-v^2}} \frac{0.295}{1+2\bar{e}} \frac{1+2.18\bar{e}}{1+2\bar{e}} \quad (6.18)$$

Equations (6.16)-(6.18) are applicable for any kind of column irrespective of its end support condition. However, wind turbine tower, as being a fixed-free column according to definition, its effective length (L_{eff}) is about twice of its actual length (L). For a fixed-free column condition, the boundary line equations stand as follows. The boundary line equation between global buckling and yield condition for a fixed-free column stands as:

$$\frac{P}{\sigma_{ys} L^2} = \frac{16}{\pi} \frac{\sigma_{ys}}{E} \frac{1}{\phi} \left(\frac{1}{1+4.47\bar{e}} \right)^2 = \frac{16}{\pi} \frac{\sigma_{ys}}{E} \frac{1}{\phi} X_1 \quad (6.19)$$

The boundary line equation between global and local buckling for a fixed-free column stands as:

$$\frac{P}{\sigma_{ys} L^2} = \frac{16}{\pi} \frac{E}{\sigma_{ys}} \frac{1}{1-v^2} \frac{1}{\phi^3} \left\{ \frac{0.295(1+2.18\bar{e})}{(1+4.47\bar{e})(1+2\bar{e})} \right\}^2 = \frac{16}{\pi} \frac{E}{\sigma_{ys}} \frac{1}{1-v^2} \frac{1}{\phi^3} X_2 \quad (6.20)$$

The boundary line equation between yield failure and local buckling for a fixed-free column stands as:

$$\phi = \frac{E}{\sigma_{ys} \sqrt{1-v^2}} \frac{1}{1+2\bar{e}} \frac{0.295(1+2.18\bar{e})}{1+2\bar{e}} = \frac{E}{\sigma_{ys} \sqrt{1-v^2}} X_3 \quad (6.21)$$

where X_1 , X_2 , X_3 are functions of eccentricity, given by:

$$X_1 = \left(\frac{1}{1 + 4.47\bar{e}} \right)^2, X_2 = \left\{ \frac{0.295(1 + 2.18\bar{e})}{(1 + 4.47\bar{e})(1 + 2\bar{e})} \right\}^2 \text{ and } X_3 = \frac{0.295(1 + 2.18\bar{e})}{1 + 2\bar{e}} \quad (6.22)$$

Eventually, to plot all failure modes with respect to various shape factor ratio, sequence of the failure modes is established with a limiting value of shape factor ϕ approaching to 0. This value is applied in stress ratios expressions and corresponding dominance is determined based on boundary line equations that already defined all boundary segments by only two failure modes in each different time. Thus, $\phi \rightarrow 0$ criterion defines which failure modes will dominate to the left of each boundary segment, as follows:

$$\frac{\sigma_{gb}}{\sigma_y} = \sqrt{\frac{PE\pi\phi}{16L^2} \frac{(1+4.47\bar{e})}{\sigma_{ys}}} \quad (6.23)$$

So if $\phi \rightarrow 0$; $\sigma_{gb} < \sigma_y$

$$\frac{\sigma_{gb}}{\sigma_{lb}} = \sqrt{\frac{PE\pi\phi}{16L^2} \frac{(1+4.47\bar{e})(1+2\bar{e})}{1+2.18\bar{e}}} \frac{\sqrt{1-v^2}}{E} \frac{\phi}{0.295} \quad (6.24)$$

So if $\phi \rightarrow 0$; $\sigma_{gb} < \sigma_{lb}$

$$\frac{\sigma_y}{\sigma_{lb}} = \sigma_{ys} \frac{(1+2\bar{e})}{1+2.18\bar{e}} \frac{\phi\sqrt{1-v^2}}{0.295E} \quad (6.25)$$

So if $\phi \rightarrow 0$; $\sigma_y < \sigma_{lb}$

Accordingly, an overall failure mode sequence can be established from Eqs. (6.23)-(6.25), as such:

$$\sigma_{gb} < \sigma_y < \sigma_{lb} \quad (6.26)$$

6.3.5 Failure mode map for different eccentricity for a cast iron alloy

A failure mode plot can be represented based on this sequence of dominance. A high strength cast iron alloy material of BS 900/2 grade (cast and normalized) is selected arbitrarily to highlight a material-specific failure-mode plot. Properties of this cast iron alloy grade are collected from commercial material selection software [93]. Two logarithmic scales are used in x, y axes with the axis notation $x = \log_{10} \phi$ and $y = \log_{10} \frac{P}{\sigma_{ys} L^2}$.

Using different values of eccentricity ratio ($\bar{e}$), the eccentricity factors X_1, X_2, X_3 are calculated from Eq. (6.22). Simultaneously, boundary segment equations for cast iron, BS 900/2, cast and normalized material are determined by generic boundary line equations, as represented in Eqs. (6.19)-(6.21). Finally, cast iron alloy specific boundary line equations can be expressed as:

$$\left\{ \begin{array}{l} y = -1.81 - x \\ y = 2.2 - 3x \\ x = 2.001 \end{array} \right| \text{for } \bar{e} = 0 \quad (6.27)$$

$$\left\{ \begin{array}{l} y = -2.46 - x \\ y = 1.57 - 3x \\ x = 2.014 \end{array} \right| \text{for } \bar{e} = 0.25 \quad (6.28)$$

$$\left\{ \begin{array}{l} y = -2.83 - x \\ y = 1.21 - 3x \\ x = 2.02 \end{array} \right| \text{for } \bar{e} = 0.50 \quad (6.29)$$

$$\left\{ \begin{array}{l} y = -3.08 - x \\ y = 0.96 - 3x \\ x = 2.024 \end{array} \right| \text{for } \bar{e} = 0.75 \quad (6.30)$$

Fig. 6.11: Dominant failure modes with respective boundary zones in varying eccentricity ratio ($\bar{e}$)

Accordingly, based on Eqs. (6.27)-(6.30), failure modes are plotted for different eccentricity ratio. Each dominant failure region is highlighted in respect to both load and shape factor, as represented in Fig. 6.11. It appears that with increasing eccentricity, boundary line between global buckling and yield failure shifts towards lower load factor values. This indicates the global buckling failure dominating zone decreases and yield failure zone increases with eccentric loading. The higher the eccentricity develops,

the narrower the global buckling failure zone becomes. In another perspective, yield failure occurs at lower load factor with increasing eccentricity.

Yield and local buckling failure boundary line simultaneously shifts rightwards. This indicates yield failure can occur at higher shape factor with increase in eccentricity i.e. more thinner tubes can now fail by yield for which local buckling was the earlier usual failure mode. Due to shift in global and local buckling boundary line, global buckling failure zone also decreases and local buckling failure zone consequently increases with higher eccentricity.

6.3.6 Optimal column design

Mass of a tubular column can be expressed in terms of its length, cross-sectional area and material density as:

$$m = A\rho L \quad (6.31)$$

Considering A as a free variable, for a given length L and load P , the mass equations can be defined based on individual stress equations:

$$m = A\rho L = \frac{P}{\sigma_i} \rho L \quad (6.32)$$

where, σ_i indicates either global buckling, yield or local buckling stress. σ_i is replaced with σ_{gb} , σ_y and σ_{lb} of Eqs. (6.8), (6.9) and (6.15), respectively, to get each individual mass equation, as follows:

$$m_{gb} = \frac{P}{\sigma_{gb}} \rho L = \frac{4\rho\sqrt{P}L^2}{\sqrt{E\pi\phi}(1 + 4.47\bar{e})} \quad (6.33)$$

$$m_y = \frac{P}{\sigma_y} \rho L = \frac{\rho PL}{\sigma_{ys}} \quad (6.34)$$

$$m_{lb} = \frac{P}{\sigma_{lb}} \rho L = \frac{\rho \phi PL \sqrt{1-v^2}}{0.295 E} \frac{1+2\bar{e}}{1+2.18\bar{e}} \quad (6.35)$$

where m_{gb} , m_y , m_{lb} indicate mass related to global buckling, yield and local buckling failure, in Eqs. (6.33)-(6.35), respectively. It is realized from Eq. (6.33) that, in case of global buckling, mass is inversely proportional to the square-root of shape factor; whereas for yield failure, it is independent of shape factor, as vivid from Eq. (6.34), and, for local buckling, mass increases proportionately with the increase in shape factor, as represented in Eq. (6.35). Based on boundary line equations of Eqs. (6.19)-(6.21), corresponding load factor equations can be expressed in terms of material property, eccentricity factor and limiting shape factor as follows:

$$\frac{P}{\sigma_{ys} L^2} = \begin{cases} \frac{16}{\pi} \frac{\sigma_{ys}}{E} \frac{1}{\phi} X_1; & \text{if } 1 < \phi \leq \frac{E}{\sigma_{ys} \sqrt{1-v^2}} X_3 \end{cases} \quad (6.36a)$$

$$\frac{P}{\sigma_{ys} L^2} = \begin{cases} \frac{16}{\pi} \frac{E}{\sigma_{ys}} \frac{1}{1-v^2} \frac{1}{\phi^3} X_2; & \text{if } \frac{E}{\sigma_{ys} \sqrt{1-v^2}} X_3 < \phi \end{cases} \quad (6.36b)$$

Equation (6.36a) indicates boundary line between global buckling and yield failure based on Eq. (6.19); this boundary line equation remains valid until a specific value of shape factor, as expressed in Eq. (6.21). For higher shape factor, the load factor equation can be expressed by Eq. (6.36b) which is the boundary line between global buckling and local buckling. Figure 6.12 is plotted based on Eq. (6.36) which highlights different failure zones based on different boundary lines. Based on associated load factor expressions of Eq. (6.36), the optimum shape factor and minimal mass equations can be defined. For this perspective, limiting load factors are determined for different limiting shape factors of Eq. (6.36). Accordingly, if $\phi = 1$, which is a limiting shape factor limit in Eq. (6.36), limiting load factor can be expressed as:

$$\frac{P}{\sigma_{ys} L^2} = \frac{16}{\pi} \frac{\sigma_{ys}}{E} X_1 \quad (6.37)$$

If $\phi = \frac{E}{\sigma_{ys}} \frac{1}{\sqrt{1-v^2}} X_3$, which is another limiting shape factor limit in Eq. (6.36), corresponding limiting load factor can be expressed as:

$$\frac{P}{\sigma_{ys} L^2} = \frac{16}{\pi} \left(\frac{\sigma_{ys}}{E} \right)^2 \frac{X_1}{X_3} \sqrt{1-v^2} \quad (6.38)$$

Fig. 6.12: Dominant failure modes with respect to non-dimensional load and shape factor

Limiting load factor expressions are represented in Eqs. (6.37)-(6.38). Optimal shape factor based design is the one which for a given material and load factor uses the least amount of material. The optimum shape factor equations are expressed in the following by inversing the load factor expression and limiting load factors of Eqs. (6.36)-(6.38):

$$\phi = \phi_{opt} = \begin{cases} 1; & \text{if } \frac{16 \sigma_{ys}}{\pi E} X_1 \leq \frac{P}{\sigma_{ys} L^2} \\ \frac{16 \sigma_{ys}^2 L^2}{\pi E P} X_1; & \text{if } \frac{16}{\pi} \left(\frac{\sigma_{ys}}{E} \right)^2 \frac{X_1}{X_3} \sqrt{1-v^2} \leq \frac{P}{\sigma_{ys} L^2} \leq \frac{16 \sigma_{ys}}{\pi E} X_1 \\ \sqrt[3]{\frac{16 E L^2}{\pi (1-v^2) P}} X_2; & \text{if } \frac{P}{\sigma_{ys} L^2} \leq \frac{16}{\pi} \left(\frac{\sigma_{ys}}{E} \right)^2 \frac{X_1}{X_3} \sqrt{1-v^2} \end{cases} \quad (6.39)$$

Based on different eccentric factor parameters (X_1, X_2, X_3), optimal shape factor will vary for fixed-free column condition. By replacing the respective optimal shape factors (ϕ_{opt}) of Eq. (6.39) into mass equations of Eqs. (6.33)-(6.35), mass minimal equations can be represented as:

$$m_{min} = \begin{cases} \frac{\rho PL}{\sigma_{ys}}; & \text{if } \frac{16}{\pi} \left(\frac{\sigma_{ys}}{E} \right)^2 \frac{X_1}{X_3} \sqrt{1-v^2} \leq \frac{P}{\sigma_{ys} L^2} \\ \frac{1.72 P^{2/3} L^{5/3} \rho (1-v^2)^{1/6}}{E^{2/3} X_2^{1/6} (1 + 4.47 \bar{e})}; & \text{if } \frac{P}{\sigma_{ys} L^2} \leq \frac{16}{\pi} \left(\frac{\sigma_{ys}}{E} \right)^2 \frac{X_1}{X_3} \sqrt{1-v^2} \end{cases} \quad (6.40)$$

By changing the load factor limits, Eq. (6.40) can be represented as:

$$m_{min} = \begin{cases} \frac{\rho PL}{\sigma_{ys}}; & \text{if } \left(\frac{\sigma_{ys}}{E} \right)^2 \sqrt{1-v^2} \leq \frac{P}{\sigma_{ys} L^2} \frac{\pi X_3}{16 X_1} \\ \frac{1.72 P^{2/3} L^{5/3} \rho (1-v^2)^{1/6}}{E^{2/3} X_2^{1/6} (1 + 4.47 \bar{e})}; & \text{if } \frac{P}{\sigma_{ys} L^2} \frac{\pi X_3}{16 X_1} \leq \left(\frac{\sigma_{ys}}{E} \right)^2 \sqrt{1-v^2} \end{cases} \quad (6.41)$$

Equation (6.41) expresses mass minimal design equations for fixed-free column condition based on global buckling, yield and local buckling failure criteria. By introducing different eccentricity values and material properties, individual material-specific failure plots can be demonstrated henceforward. A generic failure plot diagram is accordingly shown in Fig. 6.12 based on non-dimensional load and shape factor.

As an example, let us consider a thin shell structure made of AISI 9255 normalized steel material for mass minimal case study. The properties of this material are taken from CES database [93], which are as follows: $\rho = 7850 \text{ kg/m}^3$, $\sigma_{ys} = 577 \text{ MPa}$, $E = 211 \text{ GPa}$, $\nu = 0.29$. For $\bar{e} = 0.25$, following mass minimum equation can be deduced from Eq. (6.41):

$$m_{min} = \begin{cases} \frac{\rho PL}{\sigma_{ys}}; & \text{if } \left(\frac{\sigma_{ys}}{E}\right)^2 \sqrt{1-\nu^2} \leq \frac{0.27P}{\sigma_{ys}L^2} \\ \frac{1.55P^{2/3}L^{5/3}\rho(1-\nu^2)^{1/6}}{E^{2/3}}; & \text{if } \frac{0.27P}{\sigma_{ys}L^2} \leq \left(\frac{\sigma_{ys}}{E}\right)^2 \sqrt{1-\nu^2} \end{cases} \quad (6.42)$$

For a given $P = 100 \text{ kN}$ and $L = 30 \text{ m}$,

$$\left(\frac{\sigma_{ys}}{E}\right)^2 \sqrt{1-\nu^2} > \frac{0.27P}{\sigma_{ys}L^2}$$

So from Eq. (6.42),

$$m_{min} = \frac{1.55P^{2/3}L^{5/3}\rho(1-\nu^2)^{1/6}}{E^{2/3}} = 203.47 \text{ kg}$$

From Eq. (6.39),

$$\phi_{opt} = \sqrt[3]{\frac{16}{\pi} \frac{E}{1-\nu^2} \frac{L^2}{P}} X_2 = 604$$

According to CES, the maximum possible shape factor for AISI 9255 steel is 43 [93]. Hence, the shape factor needs to be reduced from to $\phi = 604$ to $\phi = 43$ which moves the failure of the thin shell to a little left in global buckling zone (see Fig. 6.12). Hence, corresponding optimal mass can be achieved from global buckling zone mass expression, as presented in Eq. (6.33):

$$m = \frac{P}{\sigma_{gb}} \rho L = \frac{4\rho\sqrt{P}L^2}{\sqrt{E\pi\phi}(1+4.47\bar{e})} = 789.56 \text{ kg}$$

Above calculations illustrate that shape factor should always be the minimum one between ϕ_{opt} and maximum possible shape factor of that particular material. Mass minimal design can be achieved likewise for each individual material. Based on above-mentioned analysis, if a column is designed as a general-purpose component, it is better to design in such that the yield failure strength remains the lowest of all three failure stresses, it will ensure the final failure to be a slowly progressive one rather than a sudden catastrophic mode. In other cases, specially, if the load factor is pre-assigned, the optimal shape factor can be readily achieved based on aforementioned mathematical representations and corresponding competitive failure zone plot.

6.4 Summary

The chapter first evaluates axial crushing failure and then highlights competitive failure boundary of thin cylinder structure; both are under non-uniformly distributed loading. Various eccentric loading conditions are experimentally tested with aluminum specimens by varying the eccentricity whereby no axial crushing failure is seen to occur. The study, henceforward, evaluates a failure diagram for thin shell structures based on all prevalent load cases in case of eccentric, non-uniform compressive loading.

It appears that with increasing eccentricity, boundary line between global buckling and yield failure shifts towards lower load factor values. This indicates that global buckling failure dominating zone decreases and yield failure zone increases with eccentric loading. The higher the eccentricity develops, the narrower the global buckling failure zone becomes. In another perspective, yield failure occurs at lower load factor with increasing eccentricity. Yield and local buckling failure boundary line simultaneously shifts rightwards. This indicates yield failure can occur at higher shape factor with increase in eccentricity i.e. more thinner tubes can now fail by yield for which local

buckling was the earlier usual failure mode. Due to shift in global and local buckling boundary line, global buckling failure zone also decreases and local buckling failure zone consequently increases with higher eccentricity.

The eccentricity parameter based load equations ultimately aim for achieving a mass minimal design criterion for thin shell structures for various cases of non-uniformly distributed loading.

Chapter 7

Conclusions and future work direction

7.1 Conclusions

This chapter summarizes the major conclusions from various works that have been undertaken in this PhD dissertation along with the scope for future work. The novelties of various investigations are also highlighted.

7.1.1 Life cycle assessment of wind farms and corresponding impact reduction

The LCA chapter has addressed few novel issues that are not covered in any previous study. Firstly, to the best of the author's knowledge, the wind turbines are never compared in any article at sub-assembly level which is significantly important to determine which parts of the wind turbine are actually contributing more impact. Next, there is no referred work published in the domain of life cycle analysis of vertical axis wind turbine technology; as prominent citation databases affirm. The present dissertation includes these issues as potential areas of LCA study.

With the progression of life cycle analysis thinking, life cycle impact assessment methods also have improved and, accordingly, employing one of the robust methodologies for LCA impact calculation deserves significant attention. Eco-indicator 99 is one of the most robust LCA methodologies that is utilized in this study for LCA calculation of wind farms; previous studies did not focus on this LCA approach. Moreover, there is no published LCA research work which compared various wind farms; this is done holistically in this study. Next, it is of utmost importance to identify how different wind farm materials and their life cycle processes contribute to overall wind turbine life cycle; this can eventually pave the way to substitute the existing materials with environment friendly ones. Though LCA researches are being conducted for the last two decades, LCA guidelines are just standardized in the beginning of new millennium and, in this stage, these studies need to be streamlined with advanced impact-savvy material selection process which has not been pursued so far. LCA chapter opens up avenues of low impact material selection process and future LCA studies are expected to follow the same course to accommodate and optimize the respective inventories based on reduced life cycle eco-imprint principles and sustainable design guidelines. Overall, the key contributions of this study can be documented as follows.

The list of leading life cycle impact categories in wind farms, sequentially, include carcinogenic emission, respiratory inorganics, ecotoxicity emission, and increase in extraction energy of minerals and fossil fuels. Out of damage categories, 'human health' with 2.09×10^6 (for HAWT onshore), 1.85×10^6 (for HAWT offshore) and 1.07×10^6 (for VAWT) eco-point appears to be the most significant in all three wind farms. Offshore horizontal wind farm exhibits higher

impact values than onshore horizontal one. However, as its capacity factor is almost 50% more than onshore one, per unit electricity profile of HAWT offshore farm is more lucrative and eco-friendly than HAWT onshore. Out of various onshore and offshore horizontal sub-assemblies, nacelle carries the highest eco-point with 1.87×10^6 Pt and, thus, appears as the most crucial one; largely surpassing other sub-assemblies by huge difference; e.g., onshore tower which is the next highest impact carrier, carries only 37% of nacelle eco-point. A large impact carrying foundation, however, accompanies with a low impact carrying tower and vice versa in onshore and offshore turbines.

VAWT onshore farm performs the best within all with lowest eco-point (Pt) values. Eco-profile of each unit electricity generation stands at 7.19×10^{-4} eco-point (Pt) in VAWT farm while for HAWT offshore and onshore, it equals 0.001105 and 0.001225 Pt, respectively. It is worth pointing out, at this stage, how an offshore VAWT farm will carry its life cycle eco-values. Within the LCI list, copper carries the most significant impact that attends to be 1.82 times of combined respective impact values of all materials that are consumed more than 1% by weight in three wind farms. The study subsequently replaces copper with aluminum and eventually reduces overall LCA impacts by 30%, 25%, 7% for modified HAWT onshore, HAWT offshore and vertical axis farm, respectively. Additional impact reduction can be ensured with introduction of low LCA impact based monolithic and sandwich composite blade designs. A concrete tower design, comparing to a presently used steel tower, also greatly decreases the life cycle eco-point in the expense of structural superiority.

Finally, a comparison of all major renewable and non-renewable power generation sources reveal the wind farms as competing remarkably with all the

technologies except the hydro-kinetic ones. And within the two most leading renewable technologies, solar farm exhibits 5 times more impact of onshore wind farm which rises to almost 8.7 times in case of onshore VAWT farm for per kWh electricity generation.

7.1.2 Material selection for horizontal axis turbine blade and tower

In wind turbine material selection chapter, the author has explored horizontal axis wind turbine blade and tower material based on multiple constraints and conflicting objective based strategies. Earlier, there is no research work in published domain which covered wind turbine material selection based on compound objectives. The design objectives, chosen herewith, comply with material efficiency pathways in design and other environmental concerns of the hour. Out of the objectives, mass minimization is assigned a higher weightage (50%) while, per unit carbon footprint, cost and embodied energy consumption minimization were assigned 20, 20 and 10% weightage. Concerning this weightage, PEEK/IM carbon fiber, UD composite and epoxy/HS carbon fiber, UD composite have appeared as the best blade materials. As for Vestas 90 m 3.0 MW turbine, PEEK/IM carbon composite exhibits 74% mass, 17% carbon footprint, 30% embodied energy reduction while epoxy/HS carbon ensures 70% mass, 34% carbon footprint, 23% embodied energy with concurrent 37% cost reduction.

Hybrid natural materials like oak (*quercus* spp.), *lignumvitae* (l) also enter into small scale onshore blade design space for low wind regions. In turn, for offshore blade applications, considering the moisture absorption permeability, neither PEEK/IM carbon nor hybrid natural materials appear suitable; thus leading to an epoxy/HS carbon blade with the highest potentiality. Oak (*quercus* spp.),

lignumvitae (l) also do not appear as optimal materials as small scale turbines are undesirable in offshore locations because of increased cost of bottom support structure.

For tower material, some of the cast iron based alloys appear as the most competitive with superior yield, compressive and fatigue strength. As demonstrated in Chapter 4, a BS 900/2 grade tower ensures 6% mass, 81% carbon, 80% energy content and 90% cost reduction in a Repower 5M turbine tower in comparison to presently used material. Conversely, epoxy/HS carbon composite provides far more weight savings (almost 78%) with simultaneous 26% carbon and energy content reduction. Limitations of the study lay in the variation of material properties from suppliers' perspective who often intend to publish only the favorable results and also the material cost will vary slightly once the inflationary effects are counted.

7.1.3 Fracture characterization of glass fiber composite material under biaxial loading

Chapter 5 focuses on fracture characterization of glass fiber reinforced composite laminate under experimental biaxial loading in the presence of various centre cracks with varying length and orientation which, to the best of author's knowledge, is the first study of its sort in corresponding research domain. Key contributions of this chapter can be documented as follows.

In a tension-tension biaxial loading condition, axial failure stresses increase by around 20% from 1:0 to 1:1 load ratio, both for 40 mm and 45 mm 0° cracks; whereas the transverse failure stresses increase from 0 MPa (in 1:0 load ratio) to 138 MPa and 126 MPa (in 1:1 load ratio) for 40, 45 mm crack, respectively. This

suggests that the specimen is able to withstand a higher axial stress when coupled with a transverse tensile stress. In another perspective, ultimate axial and transverse failure stresses corresponding to angle cracks are always found higher than 0° cracks for all load cases. This is an expected phenomenon in case of an angle crack whereby it actually experiences a mixed-mode fracture due to the presence of both crack opening and sliding modes.

For every geometric shape, there is a particular geometric factor that needs to be considered in mode I and mode II stress intensity factor calculation. Particular geometric factor of cruciform shape is numerically evaluated by FE simulation. Based on that geometric factor, mode I and mode II fracture toughness properties are calculated. It appears that the fracture toughness for both 40 mm and 45 mm cracks are about $16 \text{ MPa}\cdot\text{m}^{0.5}$ which appears to be comparable with uniaxial testing mode I fracture toughness values.

Cracks with higher length fail at lower stresses in comparison to shorter cracks and it is found that an increase in crack length by 12.5% reduces both axial and transverse failure stresses by 7%; which is a significant decrease in material strength. Consequently, DIC results show the occurrence of high strain concentration around the crack tip before failure; this represents the failure is initiated by crack. Similar strain distribution is observed for all specimens, both for 0° and 45° crack. This matches well with experimental failure modes where crack initiation dictates the failure mechanism once stress intensity factor reaches the critical values.

7.1.4 Failure mode map for tubular columns under eccentric loading

Chapter 6 highlights the failure mode map of tubular column structure under eccentric loading perspective. Eccentric compressive loading shifts competitive failure boundaries which ultimately pave the way to optimal mass and shape efficiency based design, and, to the best of author's knowledge, no published literature focused on any failure mode map with eccentric loading. Corresponding chapter initially evaluates axial crushing based compressive failure under eccentric compressive loading condition. Several experiential failure points prove that plastic flow of material closely match with yield failure rather than any distinct axial crushing mode. Accordingly, it is visualized that no ring or diamond mode failure occurs for non-uniformly distributed loading.

The study, henceforward, presents a compressive failure plot based on major compressive failure modes (global buckling, local buckling and yield failure) under varying eccentricity parameters. With increasing eccentricity, global buckling failure zone decreases and, simultaneously, yield failure zone increases in comparison to non-eccentric loading condition. This means some structures that are supposed to fail in a global buckling mode in non-eccentric condition will eventually fail in yield mode with eccentric compressive condition. Yield vs. local buckling failure zone also shifts to a higher shape factor; this indicates yield failure can now occur with comparatively higher radius or lesser thickness of tube for which local buckling was the earlier usual failure mode. Due to the shift in global and local buckling boundary line, global buckling failure zone also decreases and local buckling failure zone consequently increases with higher eccentricity.

A mass minimal tubular column design guideline is proposed consequently based on above-mentioned failure plot. A mass minimization case-study of a 30 m long tube made of AISI 9255 normalized steel is shown in Chapter 6 which only consumes ~790 kg material under 100 kN compressive loading with a shape factor of 43 which is the limiting shape factor of the particular material. Decreasing shape factor, however, increases the material consumption.

7.2 Future work direction

A list of future work directions are mentioned in the following in relevance to previous chapters.

In line with material optimization for future green wind turbines, process route optimization is very important. Though metal manufacturing processes are quite optimized, composite and sandwich material processes need to be streamlined further in agreement with net shape manufacturing strategies. This should be addressed in combination with material optimization study for designing various engineering structures.

With the progression of LCA researches, more comprehensive impact assessment methodologies are developing now-a-days that cover many newer impact categories. ReCiPe is one of such more advanced impact assessment methodology with much more impact categories (total 18) which the author used for life cycle assessment of the world's first commercialized marine current turbine during PhD. The author, however, finds this work out of scope of this thesis. In addition to using advanced impact assessment methodology, spatio-

temporal consideration in LCA is also important that needs to be studied holistically.

Biaxial tests should be extended in tensile-compressive and compressive-tensile regime for various composite laminates. In tensile-compressive test case, the material is expected to fail at comparatively lower stress than uniaxial case. Apart from cross-ply laminate, many other different layouts of GFRP composites need to be studied both for tensile-tensile and tensile-compressive space. Additionally, other composite materials like carbon fibre reinforced plastic or various natural composites also need to be studied for biaxial fracture characterization.

Next, sandwich column designs can be explored for optimal shape factor based mass minimal study under different eccentric loading condition. Sandwich column will be more failure-resistant to local buckling and, accordingly, the competitive failure boundaries will shift. Corresponding mass minimal and optimal shape factor based equations also will change simultaneously.

List of Publications

1. **A. Rashedi**, I. Sridhar, K.J. Tseng. "Multi-objective material selection for wind turbine blade and tower: Ashby's approach." *Materials and Design*, 37: 521-532, 2012.
2. **A. Rashedi**, I. Sridhar, K.J. Tseng. "Life cycle assessment of wind farms and strategies for impact reduction." *Renewable & Sustainable Energy Reviews*, 21: 89-101, 2013.
3. **A. Rashedi**, I. Sridhar, K.J. Tseng. Life cycle assessment of a marine current turbine, 3rd International Conference on Engineering and ICT (ICEI 2012), Malaysia, 4 - 6 April 2012.
4. **A. Rashedi**, I. Sridhar, K.J. Tseng. Numerical simulation of composite crack propagation under biaxial loading, 5th MRS-Singapore Conference on Advanced Materials, 20 – 22 March, 2012.
5. **A. Rashedi**, I. Sridhar, K.J. Tseng. Cruciform specimen design for biaxial analysis of composite, 15th European Conference on Composite Materials, Venice, 24 - 28 June, 2012.
6. **A. Rashedi**, I. Sridhar, K.J. Tseng, N. Srikanth. "Fracture characterization of glass fiber reinforced polymer matrix composite laminates under biaxial loading." [In preparation for 'Composites Science and Technology' (Elsevier)].
7. **A. Rashedi**, I. Sridhar, K.J. Tseng, N. Srikanth. "Minimum mass design of thin tubular structures under eccentric compressive loading". Submitted to *Thin-Walled Structures*.