NTNU – Trondheim
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Aeroelastic analysis of an offshore wind turbine

Design and Fatigue Performance of Large
Utility-Scale Wind Turbine Blades

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Master of Science in Product Design and Manufacturing
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MASTER THESIS

for

Peter Kalsaas Fossum

Spring 2012

Aeroelastic analysis of an offshore wind turbine
Aeroelastisk analyse av en offshore vind turbin

Background

The development of wind turbines has been dramatic the last decade. The largest turbine on the market is 7 MW and 10 MW seems to be coming soon. Publications in various media shows that 10 MW, 15 MW and even 20 MW turbines are under development. The traditional horizontal axis wind turbine design is the basis for most of them, and it seems like the turbine blades become larger and larger. It is a question whether this development can continue. The forces on the blades will increase linearly with the swept area of the turbine blades and the material strength will be challenged.

Objective

Aeroelastic analysis of different designs for a 10 MW offshore wind turbine

The following tasks are to be considered:

1. Literature survey
    a. The student must complete a literature survey on state of the art within wind turbine blade design.
    b. Familiar him selves with the standard IEC 61400

2. Parametric study of the 10 MW wind turbine design
    a. Get familiar with the BEM programmed in Matlab
    b. Change the design of the turbine blades

3. Carry out aeroelastic analysis for at least three different turbine blade designs

Within 14 days of receiving the written text on the master thesis, the candidate shall submit a research plan for his project to the department.

When the thesis is evaluated, emphasis is put on processing of the results, and that they are presented in tabular and/or graphic form in a clear manner, and that they are analyzed carefully.

The thesis should be formulated as a research report with summary both in English and Norwegian, conclusion, literature references, table of contents etc. During the preparation of the text, the candidate should make an effort to produce a well-structured and easily readable report. In order to ease the evaluation of the thesis, it is important that the cross-references are correct. In the making of the report, strong emphasis should be placed on both a thorough discussion of the results and an orderly presentation.

The candidate is requested to initiate and keep close contact with his/her academic supervisor(s) throughout the working period. The candidate must follow the rules and regulations of NTNU as well as passive directions given by the Department of Energy and Process Engineering.

Risk assessment of the candidate's work shall be carried out according to the department's procedures. The risk assessment must be documented and included as part of the final report. Events related to the candidate's work adversely affecting the health, safety or security, must be documented and included as part of the final report.

Pursuant to "Regulations concerning the supplementary provisions to the technology study program/Master of Science" at NTNU §20, the Department reserves the permission to utilize all the results and data for teaching and research purposes as well as in future publications.

The final report is to be submitted digitally in DAIM. An executive summary of the thesis including title, student's name, supervisor's name, year, department name, and NTNU's logo and name, shall be submitted to the department as a separate pdf file. Based on an agreement with the supervisor, the final report and other material and documents may be given to the supervisor in digital format.

Department of Energy and Process Engineering, 10. January 2012

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PREFACE

I would like to thank the wind turbine team at the Waterpower Laboratory at NTNU for interesting discussions and great teamwork. Thank to Ole G. Dahlhaug for introducing me to the field of aeroelasticity, acquiring financial support for seminars and excellent supervision. Thank to co-supervisor Lars Frøyd for educating me and for effectively collaboration with this project. Thanks to fellow colleagues at the laboratory for everyday motivation and support.

The thesis is formulated as a scientific paper.

Peter Kalsaas Fossum

Peter Kalsaas Fossum

Trondheim, June 8, 2012

ABSTRACT

Aeroelastic design and fatigue analysis of large utility-scale wind turbine blades are performed. The applied fatigue model is based on established methods and is incorporated in an iterative numerical design tool for realistic wind turbine blades. All aerodynamic and structural design properties are available in literature. The software tool FAST is used for advanced aero-servo-elastic load calculations and stress-histories are calculated with elementary beam theory.

According to wind energy design standards, a turbulent wind load case is implemented. Fatigue loads are estimated based on 100% availability and a site-specific annual wind distribution. Rainflow cycle counting and Miner’s sum for cumulative damage prediction is used together with constant life diagrams tailored to actual material S-N data. Material properties are based on 95% survival probability, 95% confidence level, and additional material safety factors to maintain conservative results.

Fatigue performance is first evaluated for the baseline blade design of the 10MW NOWITECH reference wind turbine. Results show that blade damage is dominated by tensile stresses due to poorer tensile fatigue characteristics of the shell glass fiber material. The interaction between turbulent wind and gravitational fluctuations is demonstrated to greatly influence the damage. The need for relevant S-N data to closely predict such blade stress cycle events is investigated to avoid non-conservative conclusions.

State-of-art wind turbine blade trends are discussed and different designs of the NOWITECH baseline blade are analyzed in a parametric study focusing on fatigue performance and material costs.

SAMMENDRAG

Aeroelastisk design og utmattingsanalyse av store vindturbin blader er utført. Utmattingsmodellen som er anvendt er basert på etablerte metoder og den er integrert i et iterativt numerisk design verktøy for realistiske vindturbin blader. Aerodynamiske og strukturelle design egenskaper er tilgjengelige i litteraturen. Programmet FAST er brukt for avanserte aeroelastiske last beregninger og spennings-historier er regnet ut ved bruk av elementær bjelketeori.

Et turbulent vindlast case er implementert i følge design standarder. Utmattingslaster er estimert basert på 100% tilgjengelighet og arealbetinget årlig vindfordeling. Regndråpe syklus telling (rainflow cycle counting) og Miner's sum er brukt for kumulativ estimering av skade, i tillegg til at konstante levetids diagrammer (constant life diagrams) er skreddersydd til virkelig S-N data. Material egenskaper er basert på 95% overlevelsessannsynlighet, 95% konfidens nivå, og materielle sikkerhetsfaktorer er inkludert for å oppnå konservative resultater.

Utmattingsanalyse er først gjennomført for grunndesignet på 10MW referanseturbinen til NOWITECH. Resultatene viser at skade er dominert av strekk-spenninger på grunn av svakere strekk-egenskaper for glass fiber materialet. Interaksjonene mellom svingninger fra turbulent vind og gravitasjonskrefter er påvist å ha stor effekt på skadeestimeringen. Viktigheten av å bruke relevant S-N data for å gjøre realistiske prediksjoner for slike kompliserte lasttilfeller er videre undersøkt.

Toppmoderne bladtrender er diskutert og forskjellige design av 10MW NOWITECH referansebladet er analysert i et parameterstudie med fokus på utmattelsesyteevne og materialkostnader.

CONTENTS

Design and Fatigue Performance of Large Utility-Scale Wind Turbine Blades

Peter K. Fossum, Lars Frøyd and Ole G. Dahlhaug

***Abstract*—Aeroelastic design and fatigue analysis of large utility-scale wind turbine blades are performed. The applied fatigue model is based on established methods and is incorporated in an iterative numerical design tool for realistic wind turbine blades. All aerodynamic and structural design properties are available in literature. The software tool FAST is used for advanced aero-servo-elastic load calculations and stress-histories are calculated with elementary beam theory.**

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State-of-art wind turbine blade trends are discussed and different designs of the NOWITECH baseline blade are analyzed in a parametric study focusing on fatigue performance and material costs.

***Index Terms*—Wind Turbine, Blade Design, Aero-servo-elasticity, Fatigue, Composites.**

I. Introduction

IN the 21th century the world is faced with challenges of increasing energy demand, global warming and joblessness. The European Wind Energy Association (EWEA) believes wind energy will be important to meet Europe's 2020 renewables target, tackle climate change, strengthen energy security and create green jobs [1]. EWEA estimates the energy potential of Europe's coast to be seven times consuming energy, which puts offshore wind industry in a key position to meet 2020 and later clean energy targets.

According to EWEA's latest key trends and statistics, 235 offshore wind turbines were fully grid connected in 2011. Average size of the turbines was 3.6MW up 20% from 2010 [2]. TABLE 1 lists a selection of large commercial wind turbines which are currently available, or will be under construction in near future. Although the list only includes European and American wind turbines, the interest in offshore wind energy has spread globally, with many turbine announcements being made by companies in China, Japan, South Korea and Israel [2]. Turbines in the 10-15MW range are also studied extensively and it demonstrates the move towards larger turbines. Increased turbine size and rougher environmental conditions introduce more engineering difficulties, especially regarding rotor design. The blades are exposed to greater ultimate and fatigue loads which call for stiffer blade structures to withstand 20 years in operation. At the same time lightweight structures are crucial to reduce material costs and limit loads on other turbine components.

An important criterion for success is extensive research to ensure technology developments. Norwegian Research Centre of Offshore Wind Technology (NOWITECH) is co-funded by the Research Council of Norway and industrial partners to lay a foundation for industrial value creation and cost-effective offshore wind farms. They are in the process of designing a 10MW reference wind turbine for research purposes. As part of the project, this study has focused on the fatigue aspects of an integrated numerical design tool for large offshore wind turbine blades with the purpose of generating realistic blades.

The basic approach of the tool is to run aero-servo-elastic simulations of a preliminary aerodynamic and structural design, then modify and run over if the design does not match predefined failure criteria. Focus has been on including loads from fluctuating wind to the fatigue failure criteria, which until now only have included gravitational loads. A general fatigue life prediction model is implemented which can be applied to different blade designs. As commercial wind turbine blades use fiber reinforced plastics (FRPs) as main load-bearing material, it is essential to introduce constant life diagrams (CLDs) to model the composites' different behavior in various loading.

The fatigue life model is applied to different blade designs of the current 10MW NOWITECH reference turbine and the results are evaluated in a parametric study focusing on fatigue performance and blade material costs.

II. Fatigue Life Model

A. Integrated Design

The integrated numerical design tool is described in detail in [3]. In brief, the blade element momentum (BEM) method is used to calculate aerodynamic loads and the blade structure is modeled with elementary beam theory. First step in the design

TABLE 1
LARGE OFFSHORE WIND TURBINES (COMMERCIAL PRODUCTS AND *PROTOTYPES*)[1]

Manufacturer	Turbine name	*P* (MW)	ω_{max} (RPM)	*D* (m)	*L* (m)	*W* (tons)	Material[2]	IEC turbine class
Enercon	*E-126*	*7.5*	*11.7*	*127*	-	-	*GRE*	*IA*
Vestas	*V164-7.0MW*	*7*	*12.1*	*164*	*80*	*35*	*CFRP*	*S*
REpower (LM[3])	6M	6.15	12.1	126	61.5	19.1	GRP	IB / S
Alstom (LM)	*Haliade 150-6MW*	*6*	*12.1*	*150*	*73.5*	*26*	*GRP*	*IB*
Siemens	*SWT-6.0MW*	*6*	*11*	*154*	*75*	-	*GRE*	*IA*
REpower (LM)	5M	5.075	12.1	126	61.5	19.1	GRP	IB / S
Bard	Bard 5.0	5	12.1	112	60	28.5	GRE	IC
Arewa Wind	M5000-116	5	14.8	116	56	16.5	CFRP	IA
GE Energy	GE 4.1-113	4.1	19	113	54	-	CFRP	IB
Siemens	SWT-3.6-120	3.6	13	120	58.5	17.4	GRE	IA
Siemens	SWT-3.6-107	3.6	13	107	52	15.5	GRE	IA
Mervento	*3.6-118*	*3.6*	*12.6*	*118*	*57*	*16*	*GRE*	*IIA*
Vestas	V112-3.0 MW Offshore	3	13.8	112	54.65	11.9	CFRP	IB

[1]Wind turbine technical data are collected from http://www.4coffshore.com/ (5th May 2012) and from product brochures at manufacturer's websites. The data may deviate from actual specifications, but the information is only meant to illustrate offshore wind energy marked trends.
[2]The materials listed are the blade main load-bearing materials.
[3]LM Wind Power (LM) manufactures the blades for REpower and Alstom.

process is to set up an initial design that can be evaluated in a number of relevant load cases with a more advanced aeroelastic software tool. Design properties can be exported to various time-domain aeroelastic software tools, but for the purpose of this study, FAST [4] from NREL is used. Deflections, stresses and structural frequencies are calculated and checked against chosen failure criteria [3]. If a structural design fails, material is added at strategic locations, beam properties are recalculated and the aeroelastic simulations are repeated.

In this study only fatigue aspects are further investigated, but parallel studies are ongoing for structural and aeroelastic instabilities like buckling and flutter [5, 6].

B. Load Cases

Until now the preliminary fatigue evaluation has only included gravitational loads. However, certification of offshore wind turbines requires a minimum number of design load cases (DLCs) to be considered which also include loading due to wind and waves. The load cases are listed and discussed in IEC 61400-3 (Design requirements for offshore wind turbines, 2009) [7]. Compared to wind conditions, marine conditions are assumed to have little influence on the rotor and are neglected in this preliminary blade design study. TABLE 2 summarizes the turbulent wind load case which is incorporated in the numerical design tool.

TABLE 2
FATIGUE DESIGN LOAD CASE[1]

Design situation	DLC	Wind condition
Power production	1.2	NTM $V_{in}<V_{hub}<V_{out}$

[1]From IEC 61400-1(onshore) / IEC 61400-3(offshore).

How to implement the normal turbulence model (NTM) and generate wind data at mean hub wind speeds (V_{hub}) between cut-in (V_{in}) and cut-out wind speed (V_{out}) are explained in detail in the design standard.

Turbine faults, start-ups, shut-downs, parked conditions and transport are disregarded in the fatigue evaluation by assuming 100% turbine availability. With this assumption complicated control system models are avoided, and the simplistic approach is maintained. For a final design study, though, such load cases should be considered according to design standards.

The IEC standard categorizes the environmental conditions according to turbine design classes. Most offshore turbines are designed according to the turbine classes IA, IB or IC, where "I" represents a strong wind regime (V_{ref}=50m/s) and the letters denote reference turbulence intensity (I_{ref}) from high (A) to low (C). The class S indicates that design values are specified by manufacturer.

In order to get statistically reliable load estimates, calculations must be carried out for at least six 10-min stochastic realizations or for a continuous 60 min period [8]. For the fatigue analysis wind data are generated for every wind speed in the range $V_{in}<V_{hub}<V_{out}$ (step size of 1m/s). To represent the wind data on annual basis, the probability of each wind speed is considered according to a site-specific Weibull distribution [9]. Figure 1 shows the annual wind speed distribution at a specific site in the southern North Sea.

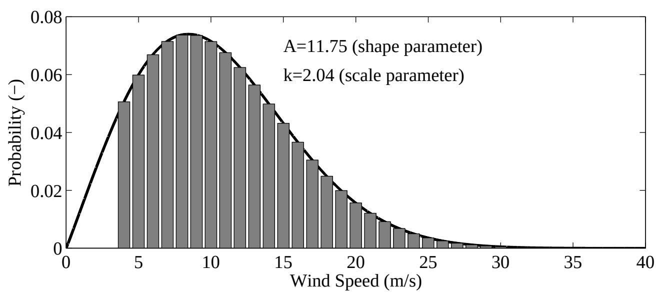

Figure 1. Weibull wind distribution of the K13 Deep Water Site [10].

For the fatigue life evaluation wind speeds below V_{in} and above V_{out} are assumed to give no damage contribution due to small loads and low Weibull probability, respectively.

C. Aero-servo-elastic Load Calculations

FAST is an aero-servo-elastic software tool that can predict ultimate and fatigue loads of two- and three-bladed horizontal-axed wind turbines. Entire turbines are modeled which can be exposed to both wind and wave external conditions. Flexible structures like tower and blades are modeled with linear modal representations [4]. Aerodynamic loads are calculated with AeroDyn [11], using either quasi-static BEM method or a generalized dynamic wake (GDW) model. IEC requires dynamics to be included and the GDW model is also recommended with respect to computation time. However, the BEM version should still be used at low speeds where the wake progresses slowly downstream and the GDW model gives unreliable results [11].

Advanced turbine control systems can be implemented in block diagram form by using a MATLAB developed interface between FAST and Simulink. Among the relevant control strategies for fatigue analysis, only normal power production is so far implemented in the integrated design tool. The control strategy allows the turbine to operate at variable speed at an optimal tip speed ratio below the maximum rotor speed. For high winds the rotor speed and rated power are kept constant by pitching the blades.

Based on blade aerodynamic and structural properties FAST considers the aero-servo-elastic interactions and calculates deflections and loads. For the fatigue life evaluation only time-histories of blade bending moments and axial forces are of interest. This is based on several model simplifications which are further described below.

D. Strain/Stress Calculation

The blade structural design method is based on an isotropic beam assumption with deformations only in longitudinal direction [3]. Another common assumption for wind turbine blade designs is to neglect torsional deflections due to very high torsional stiffness [9]. With these simplifications transverse and shear stresses can be disregarded, respectively, and the combined stresses can be calculated from normal stresses alone. Studies do still demonstrate a great influence of transverse and shear stresses in off-axis loading of glass fiber composites [12, 13], but due to lack of torsional degree of freedom in FAST, and to keep the simplicity of the model, these contributions are neglected.

Superposing of orthogonal stress contributions is recommended to preserve phase and magnitude of simultaneous loads [8]. From simple beam theory the normal strain at a point (y, x) at a cross-sectional area is given by

$$\varepsilon(y,x) = -\frac{M_1}{EI_1}x + \frac{M_2}{EI_2}y + \frac{N}{EA} \qquad 1$$

where M_1, M_2 and EI_1, EI_2 are bending moments and stiffness about local principal axes which have origin in the cross-sectional elastic center, as illustrated in Figure 2.

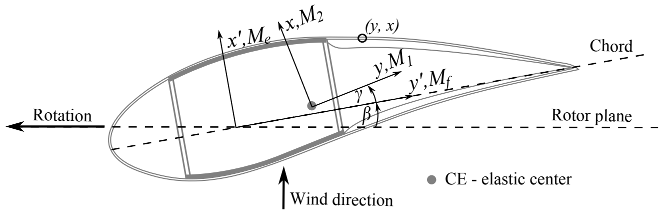

Figure 2. Blade cross-section seen from the root.

Except for the root section, M_1 will mainly be flapwise (f) bending (in wind direction) due to fluctuating wind, and M_2 will be edgewise (e) bending (in rotor plane) due to gravitational forces. N is the axial force positive into the sheet and EA is the axial stiffness. M_1, M_2 and N are outputs from the aero-servo-elastic simulations, while the geometrical properties are calculated in the design process.

The signs in Eq. 1 are defined such that there will be positive strain in tensile regions and negative strain in compressive regions. For an airfoil cross-section with the first principal axis aligned with the chord (γ=0), a positive M_2 causes tension at trailing edge and compression at leading edge, while a positive M_1 causes tension at high-pressure side and compression at low-pressure side. By multiplying Eq. 1 with Young's modulus, E, (Hooke's law) of the selected material, combined stresses can be calculated.

E. Rainflow Cycle Counting

The stress distribution from a 60 min load calculation is shown in Figure 3, illustrating highly variable stress amplitudes.

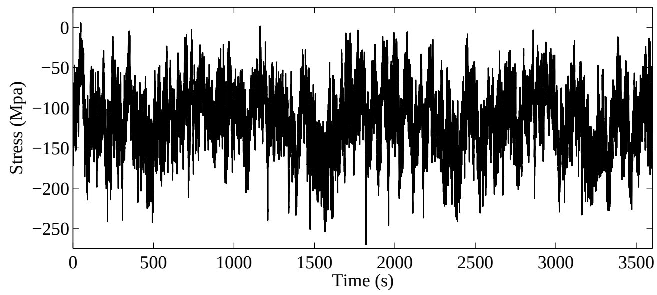

Figure 3. Stress distribution at low pressure side of an airfoil section at R=65m and V_{hub}=15m/s.

There is a general agreement in the wind energy industry that cycle damage from such random loading can be estimated by use of rainflow cycle counting and a cumulative damage model [8]. The rainflow algorithm used in this fatigue evaluation is prepared according to ASTM standard (Standard practices for cycle counting in fatigue analysis) [14]. The code is implemented in MATLAB and is available at MATLAB Central File Exchange [15].

Mean stress (σ_m), stress amplitudes (σ_a), and number of cycles (N) must be sampled for every point (y, x) of interest for the entire simulation time-history. A simple way to do this is by mean and amplitude binning. I. e. to predefine a matrix M_{ij} where every location (i, j) in the matrix has a unique mean and amplitude stress in the intervals $\sigma_{m,min} < \sigma_{m,i} < \sigma_{m,max}$ and 0

$< \sigma_{a,j} < \sigma_{a,max}$. Number of cycles from the stress history of a single simulation can be stored in $M_{ij}(V_{hub})$ and then multiplied by a factor equivalent to annual number of simulation periods for that wind speed. For a 60 min simulation this factor is $(24 \cdot 365 \cdot P_k(V_{hub}))$ where $P_k(V_{hub})$ is the Weibull probability. Eq. 2 summarizes the cycle sampling process which analogously can be performed on strain basis

$$N_{ij} = \sum_k M_{ij}(V_{in} < V_{hub} < V_{out}) \cdot (24 \cdot 365 \cdot P_k(V_{in} < V_{hub} < V_{out})) \qquad 2$$

F. Cumulative Damage Prediction

The most widely used method of predicting fatigue damage is to assume that it accumulates linearly and independently for each cycle [8]. In such case the total damage, D, can be calculated according to Miner's sum [16]

$$D = \sum_{ij} \frac{N_{ij}}{N_{f,ij}} \qquad 3$$

$N_{f,ij}$ is a matrix with number of cycles which lead to failure for the unique mean and amplitude stresses belonging to M_{ij}. This information comes from specific material S-N data and is discussed later. A wind turbine blade must be designed with a minimum lifetime of 20 years [8]. The unit of Eq. 3 is (*1/year*) and hence the fatigue failure criterion becomes $D < 1/20$. D is a non-psychical damage parameter and does not consider load sequences for instance. Limitations are further discussed in [17, 18]. In [19] the lifetime is over predicted by a factor up to 100, while [20] demonstrates a better agreement between Miner's sum and a limited set of experimental data. For this fatigue performance study the Miner's sum is assumed to give close enough predictions when constant life diagrams tailored to fatigue experimental data are implemented.

G. Constant Life Diagram

Composite materials behave differently depending on the stresses they are exposed to. There are different ultimate static strengths and fatigue failure mechanisms whether the materials are in compression or tension [17]. To be able to capture these effects it is essential to implement sufficient amount of S-N material data in so-called constant life diagrams (CLDs).

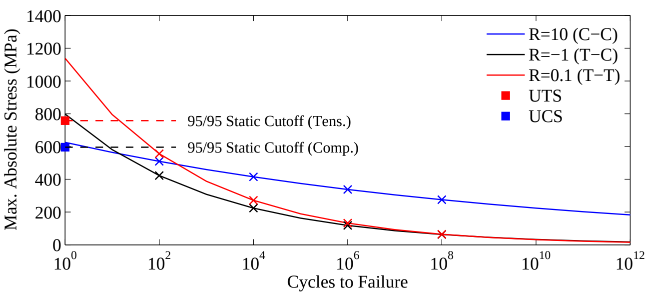

Figure 4. 95/95 S-N data for material QQ1.

Figure 4 shows S-N data for the E-glass based laminate material QQ1 from the DOE/MSU database [21]. This database contains test data from over 190 glass and carbon fiber materials and is widely used for wind energy research purposes. The basic idea of a CLD is to construct constant life lines in the (σ_m, σ_a)-plane by using static and fatigue strength data from material tests at different R-values

$$R = \frac{\sigma_{min}}{\sigma_{max}} = \frac{\sigma_m - \sigma_a}{\sigma_m + \sigma_a} \qquad 4$$

Typical R-values are R=0.1, which represents pure tension (T-T), R=10, which represents pure compression (C-C), and R=-1, which represents a combination of both tensile and compressive loading (T-C). It has been demonstrated that CLDs with multiple R-values together with the Miner's sum give predictions close to experimental data [17]. However, fatigue testing of composite materials is time-consuming and often performed by manufacturers who keep the results confidential. Currently the DOE/MSU database is the best available resource to look for blade materials, although the number of R-values tested for some materials are limited. To get accurate life predictions it is recommended to use R=10, R=-1 and R=0.1 as a minimum to construct CLDs [22].

The S-N curves in Figure 4 are fitted to experimental data with a power law model

$$\sigma_a = \frac{\sigma_0 K N_f^{-1/m}}{\gamma_m \gamma_n} \qquad 5$$

where σ_0 is single-cycle static strength, K and m are curve fit parameters and γ_m and γ_n are partial material safety factor and consequence of failure factor. The fitted curves match the data very well, except below 10^2 cycles where the (T-T)-curve exceeds the ultimate tensile strength (UTS) and the (T-C) and (C-C)-curve exceed the ultimate compressive strength (UCS).

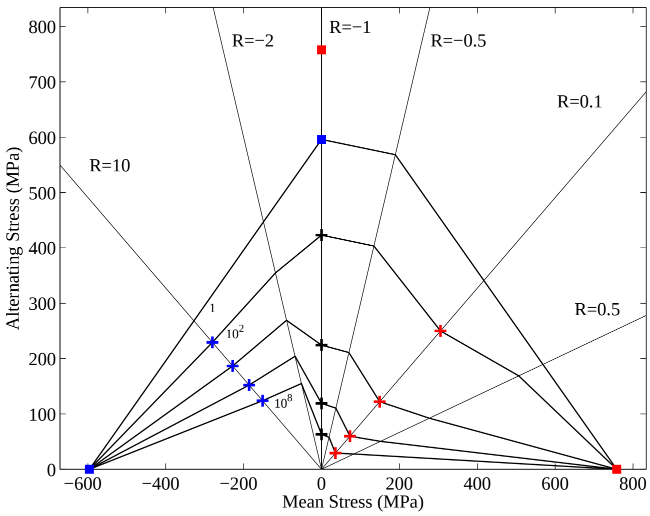

Figure 5. CLD of the QQ1 material. The lines are constructed with the method in section II.G and verified with figures in [23].

To fit the data here the curves should flatten out towards the ultimate strengths, as illustrated with the dotted lines in Figure 4. This is fairly easily implemented in a CLD with static strength cutoffs, as recommended by [23].

For R=-1 (T-C), which is a vertical line at σ_m=0, the material is assumed to fail at UCS. The single-cycle strengths at other R-values can be calculated with the below relations [24]

$$\sigma_0^{\sigma_m<0} = UCS + \sigma_m = \frac{UCS}{1-r} \quad 6$$

$$\sigma_0^{\sigma_m>0} = UTS - \sigma_m = \frac{UTS}{1+r} \quad 7$$

$$r = \frac{\sigma_m}{\sigma_a} = \left(\frac{1+R}{1-R}\right) \quad 8$$

$$\sigma_m = \sigma_a r \quad 9$$

It is now possible to calculate (σ_m, σ_a) for all R-values for chosen values of N_f. These points with addition of the ultimate strengths form the constant life lines, more exactly by linearly connecting the points between the ultimate strengths for each N_f separately. The only corrections that have to be made are the static strengths cutoffs. This can be incorporated by checking that the sum of the mean and amplitude stress calculated is less or equal to the restrictive ultimate strength for that region. If not, the single-cycle strength has to be gradually reduced until the sum is within the static strength limitation.

Figure 5 shows a CLD for the QQ1 material constructed with 6 R-values. The CLD is equivalent to Fig. 13 in [23], except that this CLD is not normalized to the mean static tensile strength. The plus signs in Figure 5 correspond to the crosses in Figure 4 and illustrate the connection between SN-data and CLDs. In a CLD the maximum absolute stress is found as the sum of the absolute mean and amplitude stress.

Ultimate strengths and curve fit parameters used to construct the CLD are statistically treated to represent a 95% survival probability with a confidence level of 95% (95/95 model). The process is carried out and further described in [23] following Echtermeyer [25]. The 95/95 model is recommended practice for fatigue analyses [8]. Beyond this, partial material safety factor and consequence of failure factor have to be included (not included in Figure 5) according to design standards.

For fatigue analyses the consequence of failure factor is 1.15, while the partial material safety factor is given as a product of several factors. A detailed description is found in DNV-DS-J102 (Design and Manufacture of Wind Turbine Blades, Offshore and Onshore Wind Turbines) [26]. A typical material safety factor for a FRP material lies between 1.3-1.7.

When all this have been considered, $N_{f,ij}$ can be calculated by interpolating between the constant life lines in the constructed CLD. This is not a straightforward process, but a step by step procedure can be found in [26]. This also completes the fatigue life model, as the cumulative damage can be predicted by Eq. 3.

III. Case Study – Fatigue of the NOWITECH Blade

Fatigue analysis is performed for different blade designs of the NOWITECH reference wind turbine. This section discusses the results and emphasizes design challenges with respect to fatigue. An objective of NOWITECH is to develop realistic design tools for novel offshore wind energy concepts. As part of the project this study has worked on developing a trustworthy fatigue model based on established methods, and the model is applied to different designs of the NOWITECH baseline blade. Focus has been on composite material properties, which highly affect the fatigue behavior and have significantly importance for the blades' total mass.

A. The 10MW Baseline Design

The reference wind turbine is designed for a fictitious offshore wind farm in the Dutch North Sea [10] where the climate is characterized by strong average winds and medium wind shear and turbulence (turbine class IEC IB). The annual wind distribution is shown in Figure 1. The bars illustrate the wind probability for each wind speed between the cut-in (V_{in}=4 m/s) and cut-out wind speed (V_{out}=30 m/s) of the reference turbine. Although the wave conditions are assumed to be rough and dimensioning for the substructure, wave forces are disregarded in the blade design. The diameter of 141m is determined by the design wind velocity (V_{design}=13 m/s), which for this turbine is chosen rather arbitrarily to get high specific rating (power per area) and relatively short blades (R=68 m) for a given rated power of 10MW. The iterative aeroelastic design process is described in detail in [3]. Briefly the airfoil family is similar to the one used for the NREL 5MW turbine and the blade structure is designed with shell, spar caps and shear webs to carry the main pitching and

TABLE 3
Spar-cap materials used in the NOWITECH blade designs[3]

Material	UTS (MPa)	UCS (MPa)	E (GPa)	ρ (kg/m^3)	v_f (%)	UD-fibers (%)	R=10 K, m	R=-2 K, m	R=-1 K, m	R=-0.5 K, m	R=0.1 K, m	R=0.5 K, m
GG2[2]	468.9	269.2	29	1880	0.50	70	1.10, 15	-	1.06, 13.5	-	1.30, 7.4	-
GH4	600.6	400.4	74.3	1621	0.50	70	1.03, 28	-	1.02, 17	-	1.01, 48	-
QQ1[1]	453.9	356.9	34	1890	0.52	70	1.05, 22.5	1.05, 16.7	1.34, 7.3	1.36, 7.1	1.50, 6.4	1.63, 7.6
WS HiPer-tex™	422.2	-	38.3	2080	0.63	70	-	-	-	-	1.12, 10.15	-

[1]QQ1 and and WindStrand™ HiPer-tex™ are modified to represent 70% UD fibers, but fatigue properties are assumed to be unchanged.
[2]A glass fiber material similar to GG2 with 50% UD fibers is used as shell material for all designs. Fatigue properties are assumed to be similar to GG2.
[3]All actual material properties are available in the literature.

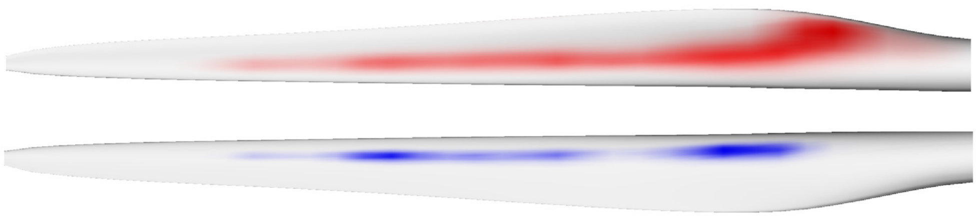

Figure 6. Fatigue damage of the 10MW NOWITECH baseline blade design. The upper blade shows tensile damage (red) at high-pressure side for the GG2 material. The lower blade shows compressive damage (blue) at low-pressure side for the GH4 material. The damage is up-scaled to visualize damage locations.

bending moments. The intention of NOWITECH is to have four-legged lattice substructure as a combined tower and foundation, but for the purpose of this study, a standard tubular tower is used. The control system allows the turbine to operate at variable speed at tip speed ratio 7.8 below maximum rotor speed (ω_{max}=12.1 rpm). For high winds the rotor speed and rated power are kept constant by pitching the blades.

The baseline blade materials are FRPs with $0^o/\pm45^o$ (triaxial) fiber lay-up. Such lay-up is practical from a manufacturing point of view and the 0^o and 45^o plies yield good resistance for bending and torsion, respectively. The blade is designed with a hybrid spar flange where carbon fiber reinforced plastic (CFRP) is the main load-bearing material. Both glass- and carbon fibers are 70% unidirectional (UD) in the spar, while the shell material is pure glass with slightly less UD fibers (50%).

The materials for the baseline blade are from a blade design study performed for Sandia National Laboratories where material properties are determined by MSU based on a combination of test data and laminate theory [27]. Fatigue properties are given for ten different spar cap materials with S-N data for 3 *R*-values. The chosen hybrid and glass fiber material, denoted GG2 (Griffin Glass #2) and GH4 (Griffin Hybrid #4), are listed in TABLE 3. GG2 has woven UD fibers with stitched triaxial fibers, while GH4 has stitched UD carbon fibers and glass biaxial ($\pm45^o$) fibers.

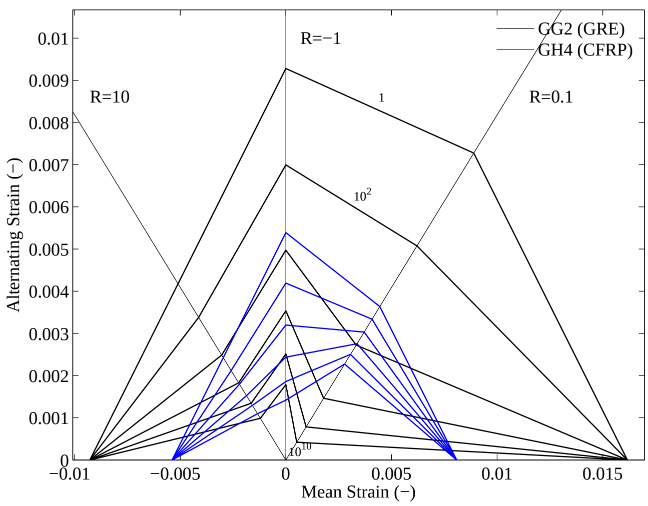

Figure 7. GG2 and GH4 CLDs constructed with 95/95 strain S-N model.

Based on the material properties CLDs are constructed on strain basis and the diagrams are shown in Figure 7. On strain basis CLDs of different materials are comparable and hence more useful in evaluation of material fatigue properties. The CLDs are constructed with 95/95 S-N model and safety factors are included.

According to DNV-DS-J102 the consequence of failure factor is 1.15 for fatigue modeling, base material factor is 1.2, material strength reduction factor for epoxy is 1.1, manufacturing strength reduction factor for UD fibers with ply drops is 1.1, and combined they multiply up to a total safety factor of $\gamma_m\gamma_n$=1.67 (for all materials).

The glass fiber material shows greater ultimate strains, but steeper S-N curves result in poorer tensile fatigue strength for high cycles (10^4-10^{10} cycles). In compressive regions carbon is generally poorer except at very high cycles (10^{10} cycles). The GH4 material shows much higher elastic modulus (74.3 GPa) compared to GG2 (29 GPa).

The glass fiber shell material is assumed to have similar fatigue characteristics as GG2.

B. Fatigue Analysis of the 10MW Baseline Design

FAST aeroelastic analysis is performed and loads about local principal axes are calculated at 36 span-wise locations. The analysis includes a 60 min continuous simulation for every wind speed between V_{in} and V_{out}. By assuming 100% availability and using the K13 Deep Water Site Weibull distribution, annual strain distributions are calculated at 72 cross-sectional points at every span-wise location. Strain calculations and all further steps in the life prediction model are implemented in MATLAB. The unique mean and amplitude strain values that are basis for the N_{ij} and $N_{f,ij}$ matrices are chosen to be within the intervals UCS < $\varepsilon_{m,i}$ < UTS and 0 < $\varepsilon_{a,j}$ < UCS to include all regions in the CLDs. To reduce computational time the intervals are limited to hundred values, but resolution tests are performed to check for inaccuracies in the sampling process.

Figure 6, Figure 8-Figure 9 and TABLE 4 summarizes the fatigue damage results of the 68 m NOWITECH baseline blade. Fatigue analysis with GH4 S-N data is performed to predict the lifetime of the spar flange carbon fibers and GG2 S-N data are used to investigate the shell glass fibers.

The bottom blade in Figure 6 shows compressive carbon fiber damage at low-pressure side of the blade. Since CFRP perform better in tension than compression this result is as expected. A lifetime in the order of 10^5 years, however, witnesses the superior carbon fiber fatigue characteristics. The color intensity is up-scaled to indicate the carbon damage locations as it would not have been visible if scaled with glass fiber tensile damage.

Figure 9 shows that the maximum CFRP damage is slightly alternated towards the leading edge. This can be explained by the principal axes which are rotated counterclockwise from the zero-principal angle coordinate system aligned with the chord. The rotation is due to a geometrical transition towards the root; from relatively thin airfoil-sections to thick cylinders. With this change the edgewise stiffness levels out and the resultant bending moment gets a greater edgewise contribution. In Figure 8 it is clearly seen that EI_2 flattens out, the distance between EI_2 and EI_1 reduces, and principal angle increases.

The GG2 glass fiber material is predicted to fail in tension at high-pressure side of the blade after 318 years. Maximum damage is located just below the trailing edge as can be seen in Figure 9. The much lower lifetime compared to carbon fiber is as expected because of the much poorer tensile characteristics of GG2 for high cycles.

Fatigue characteristics are highly dependent on moment amplitude values, however, the mean values give some indication of blade behavior. At the locations of both compressive and tensile damage M_1 retains its value even though principal angle increases. Since M_2 is small the resultant moment acts almost perpendicular to the M_1-axis, as indicated by the dotted arrows in Figure 9. The resultant moments are slightly alternated clockwise because of the negative values of M_2 closer to the root.

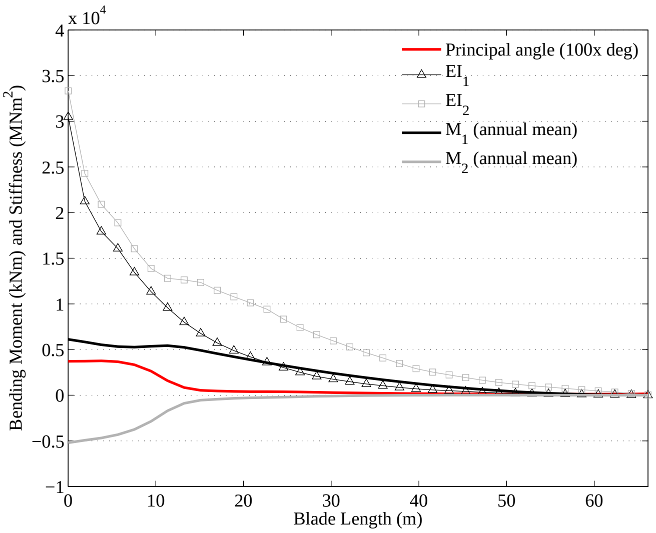

Figure 8. Span-wise distributions of blade stiffness and annual mean bending moments.

TABLE 4
FATIGUE DAMAGE OF THE NOWITECH BASELINE BLADE

Parameter	GG2 (GRE)	GH4 (CFRP)
20 year damage	0.0628	$4.58 \cdot 10^{-5}$
Lifetime	318 years	$4.27 \cdot 10^{5}$ years
Span-wise location	R=12.86 m	R=20.42 m
Cross-sectional location	Flap./edge. tension	Flap. compression

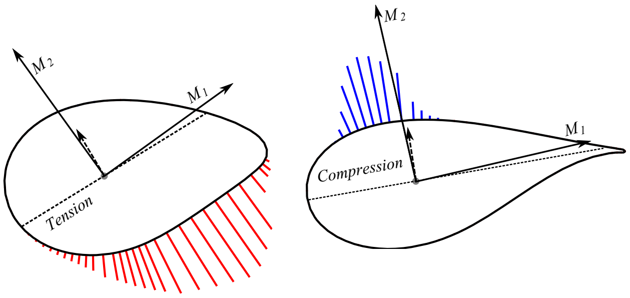

Figure 9. Cross-sectional location of tensile damage (red) for GG2 and compressive damage (blue) for GH4. The bar lengths are up-scaled to visualize damage.

The loads calculated in FAST are results from complex aero-servo-elastic interactions. Turbulent wind gives fluctuating aerodynamic loads, gravitational loads are highly alternating, local accelerations and centrifugal forces are present, and it all interacts with an elastic blade.

Figure 3 shows a typical stress distribution dominated by loads from turbulent wind. The cycle events are spread to a wide range of mean and amplitude stresses and damage is normally caused by the more extreme events.

Gravitational loading alone results in sinusoidal variations with constant amplitude. An annual average rotor speed of 10 rpm gives 10^8 cycles after 20 years in operation, which for a heavy blade could cause significant fatigue damage.

Leading and trailing edge of a wind turbine blade are typically equally exposed to tensile and compressive stresses from the gravity loading. On the other hand, the aerodynamic loading causes non-zero mean stresses which to a greater extend give regions with pure tension and pure compression, at the high-pressure side and low-pressure side, respectively.

The flapwise damage (both tensile and compressive) outwards on the blade is dominated by oscillations due to turbulent wind. When the blade geometry and stiffness changes towards the root, though, the stresses get more affected by gravitational loading. At the point where maximum damage occurs the two sources of loads are greatly interacting. The combined effect of large flapwise loads and high-cycle edgewise gravity oscillations lead to cycle events in the order of 10^8 which are moved into pure tensile region. The concentration of tensile damage cycles are located where the "10^8"-line intersects with the "R=0.1"-line in the GG2 CLD in Figure 7.

The maximum CFRP damage (compressive damage) is dominated by extreme cycle events of the turbulent wind, located at the crossing between the "10^{10}"-line and the "R=10"-line in the GH4 CLD in Figure 7.

C. The Importance of Tailored CLDs

Fatigue lifetime and damage locations of the NOWITECH baseline blade are greatly dependent on the CLD used. The GG2 CLD is constructed with 3 *R*-values to include effects of reversed loading (*R*=-1), pure compression (*R*=10) and pure tension (*R*=0.1). CLDs established from actual FRP material S-N data are recommended practice according to IEC.

Simplified methods are still frequently used in the literature. In the recent Upwind 20MW Wind Turbine Pre-Design project [28] a Germanischer Lloyd (GL) 2003 CLD was used for the fatigue analyses. In the 2010 edition of the standard (Guideline for the Certification of Wind Turbines) [29] it states that a shifted linear CLD can be used if no S-N data are available for actual material.

Figure 11 compares the GG2 CLD with a GL 2010 simplified CLD which is based on GG2 material properties. The material safety factors in the GL standard do slightly differ from the DNV standard, but the resulting simplified CLD will anyhow differ a lot from the CLD tailored to actual S-N data. The poorer tensile characteristics of GG2 is not represented and because of the much higher UTS compared to UCS, the diagram is shifted far into tensile region, making the CLD non-conservative for tensile loads and conservative for compressive and alternating loads.

The lifetime is predicted to 15027 years at low-pressure side (max. compressive damage at *R*=20.42m) and 34136 years at trailing edge (max. tensile damage at *R*=14.75m). The dominating compressive damage at low-pressure side is not expected for GG2 because of the much poorer tensile characteristics. There is small damage at high-pressure side, but evidently not as much as it should be. The tensile damage result is clearly non-conservative compared to the fatigue analysis where actual material S-N data are used and the lifetime is predicted to 318 years. For reversed and compressive loading, though, damage is over-predicted compared to previous results.

By use of GL 2010 CLD the lifetime is 4625% greater compared to the lifetime predicted by use of tailored CLD. This is even based on a comparison between tensile damage (GG2 CLD) and conservative compressive damage (GL 2010 CLD). The predicted damage in pure tension regions differs with a factor up to 1000 and it demonstrates how non-conservative the GL 2010 CLD is in this region.

Fatigue damage predicted by use of the simplified GL2010 CLD is up-scaled and plotted at the blade surface in Figure 10.

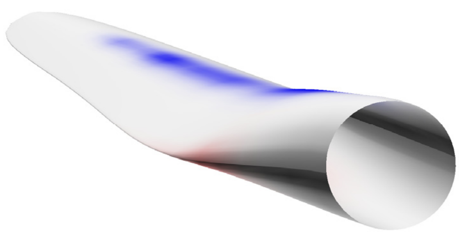

Figure 10. Fatigue damage of the NOWITECH baseline blade by use of GL 2010 shifted linear CLD.

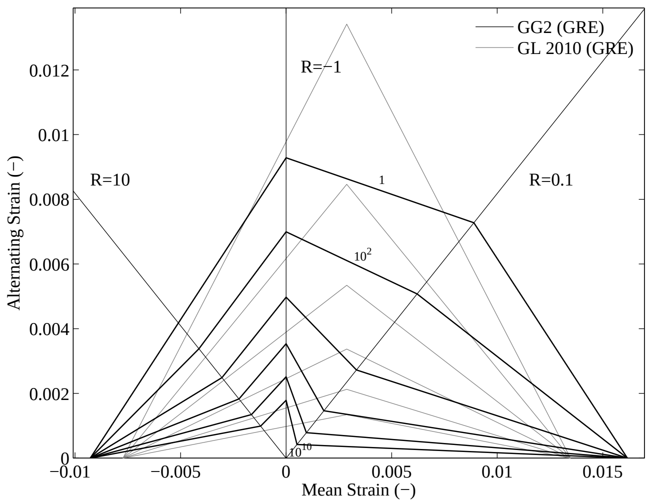

Figure 11. Comparison of the GG2 CLD and a shifted linear CLD from the GL 2010 standard.

Damage predictions can also be affected by the resolution of the rainflow cycle binning. Analysis with much higher resolution is performed and the damage is seen to increase only few percent (up to 4%). The conservative trend can be explained by the fact that number of cycle events generally decreases with increasing stress level, which means that more cycles are shifted to worse areas in the CLD. Analysis with relatively high resolution is still suggested since the predictions can be more sensitive for other CLDs.

The most important part of fatigue analysis proves to be the use of CLDs that are tailored to S-N data of the actual material. The simplified shifted linear CLD (GL2010) over-predicted the lifetime of the baseline blade with 4625% compared to the CLD constructed with S-N data for *R*=10, *R*=-1 and *R*=0.1. This supports the recommendation by [22] to use these 3 *R*-values as minimum to construct a CLD. Similar results were discussed in the OPTIMAT blade fatigue project and extended research on the most relevant *R*-values for wind turbine blades in practice was suggested [30].

For the baseline blade materials only those 3 *R*-values are available. In order to study the life prediction sensitivity of more *R*-values, fatigue analysis is performed for a modified baseline blade where the hybrid CFRP material is exchanged with the much tested QQ1 E-glass material. The material properties including S-N data for 6 *R*-values are listed in TABLE 3. In Figure 12 CLDs constructed with both 3 and 6 *R*-values are compared.

The QQ1 fatigue properties are discussed in [23] and the poor tensile fatigue characteristics are pointed out. Compared to GG2, QQ1 has poorer tensile characteristics, but better compressive characteristics.

Fatigue analyses are performed for the QQ1 blade with both 3 and 6 *R*-values being used. The *R*=0.5 S-N data do not change the CLD at high cycles, but the changes at *R*=-2 and *R*=-0.5 have significant importance for the stress cycles represented. The constant life lines are lifted upwards for both compressive-dominated (*R*=-2) and tensile-dominated (*R*=-0.5) reversed loading. The lifting indicates that QQ1 is stronger in reversed loading than pure compression and tension. A similar tendency is seen in the GG2 CLD.

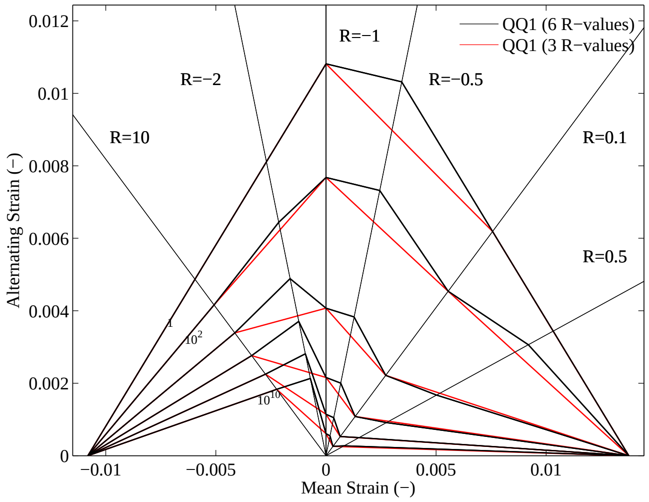

Figure 12. QQ1 CLDs constructed with S-N data for 6 *R*-values (black) and 3 *R*-values (red).

The lifetime predicted with 3 *R*-values is under-predicted 20-25% compared to lifetime by use of all 6 *R*-values. At locations which are more exposed to large reversed loading, such as the trailing edge at the inner airfoil-sections, damage differences up to 150% are seen between the two fatigue tests.

With these findings and by recognizing the trends in the GG2 CLD, the lack of *R*-values in the reversed loading region is believed to give conservative life prediction. Although a CLD based on 6 *R*-values gives more realistic life prediction, 3 *R*-values do still fairly represent the different loading effects and it should be sufficient for a preliminary blade design study.

As a concluding remark of the baseline blade fatigue study, 318 years predicted by a conservative fatigue model are clearly within fatigue design requirements.

D. Blade Material Trends

As can be seen in TABLE 1 large offshore wind turbine blades are manufactured with FRPs. Glass fiber reinforced epoxy (GRE) is the dominating load-bearing material due to its availability, cost and strong mechanical properties. LM Wind Power is a leading blade manufacturer and their latest flagship; the 73.5 m glass fiber reinforced polyester (GRP) blade is stretching the capabilities of low-cost polyester. Polyester resin is one-third the price of epoxy and easier to process, but with poorer mechanical properties it usually results in heavier blades. By focusing on optimizing the cheaper technology, LM Wind Power still believe they provide the best possible balance between price and performance [31], [32].

The blue diamonds in Figure 13 are the latest polyester-blades of LM Wind Power and they follow the weight trendline of CFRP blades (red markers). The 73.5 m blade is even located below the trendline of commercial CFRP blades (red diamonds).

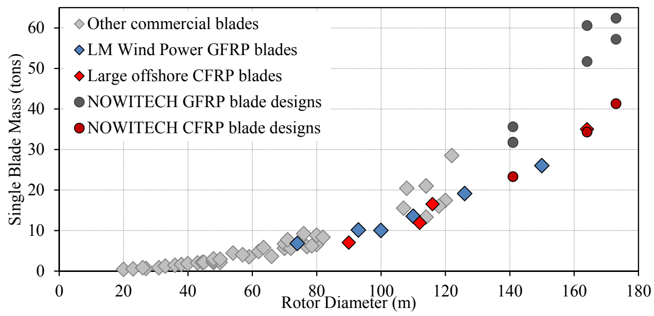

Figure 13. Weight trends of commercial wind turbine blades together with NOWITECH blade designs.

The grey diamonds represent other commercial blades, most of them designed with GRE. The great weight-differences indicate the lightweight focus of the newer generation offshore turbines. A heavy blade leads to increased material costs for the blade itself, but more loads are also generated which must be compensated for in the design of the other turbine components. For offshore turbines in rough climates and at deep water, the foundation designs are for instance highly affected by the rotor loads.

To reduce loads and make large blades cost-efficient it is therefore essential to have slim, stiff and lightweight designs. As demonstrated with the NOWITECH baseline blade fatigue analysis, CFRPs show superior fatigue characteristics and are increasingly used in blade spar cap designs. By considering mechanical properties alone CFRP is an obvious first choice, but the current carbon cost reduces the advantage. Future large-scale production of carbon fiber-rich airplanes by Boeing and Airbus give carbon suppliers a huge backlog of orders, and other potential users (e.g. wind turbine blade manufacturers) are in the upcoming decade expected to be challenged by a carbon fiber shortage [33]. The increasing use of CFRP for various engineering applications may still involve successfully developments of less costly fibers which not necessarily are designed with aerospace-quality, but more tailored to wind energy applications [33].

Because of the uncertainties around the carbon fiber marked it is reasonable in a blade design study to consider 100% glass fiber blade designs. Besides the remarkably lightweight blade designs of LM Wind Power, this strategy is supported by the fact that the most extensively used offshore turbine nowadays is the GRE-based SWT 3.6-120 machine by Siemens [2].

In the following section different designs of the baseline blade are discussed with focus on fatigue performance and material costs.

IV. Parametric Study of Different Blade Designs

A. Cost Estimates

In the following parametric study different high performance FRP materials are evaluated as the main load-bearing spar-cap material of the 10MW NOWITECH baseline blade design. 15MW versions of the baseline blade and additional blade designs based on the Vestas V164-7.0MW [34] turbine are included in the study to evaluate trends for various rotor diameters. The diameters of the 10MW and 15MW designs are 141 m and 173 m based on a 13 m/s design wind speed. The Vestas-like designs are based on Vestas design basis (P=7MW, V_{design}=11 m/s) resulting in 164 m rotors. The shell material for all designs assumed to have the fatigue characteristics of GG2.

In the Sandia blade material trade-off study [27] the cost of carbon stitched UD fabric was estimated to be $K=5.1$ times the cost of E-glass stiched triax fabric, where the cost-ratio K is the price of carbon divided by the price of glass fiber. With the current uncertainties in the composite marked, these numbers are of course changing, but at least they point out a considerable cost gap. The trade-off study estimates the cost of next-generation carbon stitched UD fibers and carbon pre-impregnated UD fibers to be K=3.4 and K=2.3 times the cost of triaxial E-glass, respectively. However, the study is performed before the aerospace carbon-rush. As no clear carbon price is available the parametric study is focused on what the carbon price must be for the hybrid designs to be cost competitive against glass designs.

B. *High-Performance Reinforcements*

As shown with the fatigue results of the 10MW hybrid baseline blade, GH4 has superior fatigue characteristics and stiffness-to-weight ratio. GH4 is used as spar-cap material for all the hybrid designs.

For the pure glass fiber designs GG2 is used as a less-expensive, average-performance reference design. A slightly stiffer 10MW blade was designed with the QQ1 material, but rejected due to the poor tensile fatigue characteristics (6 years lifetime).

Owen Corning designs high performance glass fiber reinforcements especially intended for wind turbine technology [35]. The HiPer-tex™ WindStrand™ (WS) series has been tested by Risø National Laboratory and show superior tensile fatigue characteristics. An elastic modulus of 38.3 GPa for the triaxial fabric is also a great leap compared to 29 GPa of GG2. Since only R=0.1 95/95 S-N data are available for the WS material, some assumptions had to be made in order to adopt the material for blade design and carry out fatigue analyses.

Fatigue results of the NOWITECH baseline blade show conservative results by use of 3 R-values; R=10, R=-1 and R=0.1. In addition results clearly show that pure tensile damage is dominating compared to damage from pure compressive (R=10) and reversed loading (R=-1). By using the WS R=0.1 S-N data for reversed loading and the QQ1 R=10 S-N data for pure compression, the CLD should be greatly conservative. The resulting modified WS material, denoted WS*, is shown together with the GG2 CLD in Figure 14.

In the region where the WS* CLD is tailored to actual material S-N data (R=0.1) and where maximum damage is expected to occur, WS* shows better fatigue characteristics. The strictly conservative WS* data at R=-1 lead to poorer characteristics for alternating loads, compared to GG2. The actual WS material will most likely perform better also in this region. The much smaller UTS for WS* has negligible impact on fatigue performance.

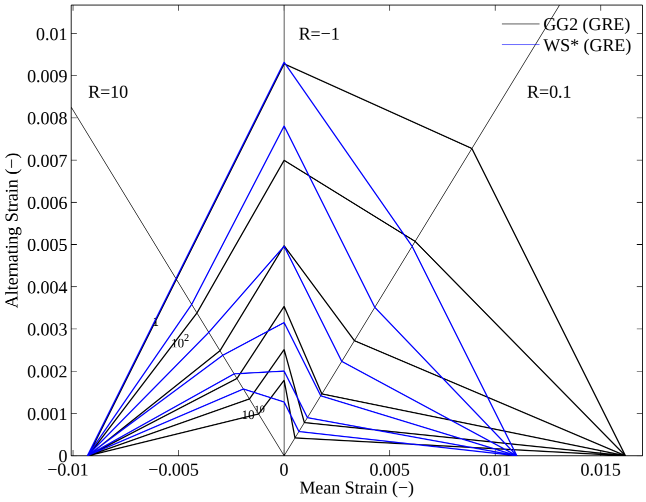

Figure 14. GG2 CLD and WS* CLD (constructed with WindStrand HiPer-tex™ R=0.1 S-N data for T-T and T-C regions and QQ1 R=10 S-N data for C-C).

C. *Fatigue Performance versus Material Costs*

TABLE 5 lists the different designs with corresponding total weight, fiber weights, cost-ratio and fatigue lifetime. A complete list of material properties is found in TABLE 3.

TABLE 5
MATERIAL WEIGHTS, COST ESTIMATE AND FATIGUE PERFORMANCE

Design	Spar FRP	W (tons)	W_{H-P} (tons)	W_{GG2} (tons)	K (-)	L (years)
H-10MW	GH4	23.3	2.31	8.45	3.8	318
G-10MW	GG2	35.6	-	17.22	-	146
G-10MW	QQ1	31.8	-	-	-	6
G-10MW	WS	31.7	5.76	10.71	1.1	121
H-V164	GH4	34.3	2.61	11.39	5.7	1166
G-V164	GG2	60.6	-	26.33	-	520
G-V164	WS	51.7	6.64	13.61	1.9	455
H-15MW	GH4	41.3	4.06	15.20	3.7	208
G-15MW	GG2	62.4	-	30.20	-	60
G-15MW	WS	57.2	10.21	18.47	1.1	78

The GG2-10MW blade has a stiffness-to-weight ratio of 0.81 (GPa/tons) which compared to 3.2 of the GH4-10MW blade leads to shorter lifetime. The lifetime of the GG2-15MW blade is reduced analogous compared to the GH4-15MW blade.

If the carbon price is below K=3.8 and K=3.7 for the 10MW design and 15MW design, respectively, the hybrid designs are cost competitive with respect to blade fiber costs alone. These numbers are based on the hybrid blades' total weight of high-performance spar-cap fibers (W_{H-P}) and shell glass fibers (W_{GG2}) compared to the weight of the overall glass fibers for the pure GG2-designs. Although the cost of the next-generation carbon materials were expected to fall below these K-values in the 2002 material trade-off study, such low carbon prices are less expected at present. On the other hand, stiffer carbon materials lead to reduced rotor loads which positively affects the material usage of other turbine components. For the purpose of this preliminary blade design study, however, the *K*-values give some indication of to what extent high-performance reinforcements should be used.

This comes out clearer by introducing the V164-designs. For the hybrid version to be cost competitive the carbon price must be less than K=5.7 times the price of GG2. This result together with the fact that the actual Vestas V164-7.0MW turbines use CFRP in the blade spar-cap design, indicate that hybrid blade design is more favorable for such a turbine design.

The *K*-variation may be explained by different dimensioning failure criteria in the blades' structural design. The 10MW- and 15MW-blades are based on relatively high design wind speed (13m/s) which makes ultimate loads more dimensioning for the blades' total mass. In Figure 7 the much greater ultimate strains of GG2 compared to GH4 are clearly seen and it involves a reduced advantage for CFRP. The smaller performance gap is recognized by the less hybrid-to-glass weight increase for the 10MW- and 15MW designs (51-52%), compared to weight increase for the V164-design (77%).

The lower design wind speed of the 7MW turbines reduces the ultimate loads and the masses will to a greater extent be dimensioned by requirements for blade structural frequencies and tip deflection. When such failure criteria are dimensioning the material stiffness is the main determining factor, and hence the hybrid design is more favorable. As expected, the results of the WS-designs show similar trends.

Largest fatigue damage is found in the shell glass fiber material for all designs. A stiffer blade leads to reduced deflections and it is clearly recognized by lower mean strain values. In the GG2 CLD it is seen as the concentration of edgewise gravity oscillations is shifted towards zero-mean where the material performs remarkably better.

The WS-designs do not have noticeably better fatigue performance than the GG2 designs. A small increase in strain amplitude values is seen for the WS-designs which counteract the effect of less deflection. However, all designs gain great weight reduction compared to GG2-designs. The cost-ratios do still indicate that the price of the high-performance fibers cannot exceed the price of average-performance fibers considerably. Again, this is based on blade fiber weights alone and is only adequate for a preliminary blade design study. For instance, costs of epoxy resin are not considered which are favoring for the glass designs in this parametric study.

A final decision cannot be made before the entire turbine is designed, analyzed and evaluated based on actual material costs.

V. Conclusions

Aeroelastic design and fatigue analysis of large utility-scale wind turbine blades are performed. The fatigue model is based on established methods and is incorporated in an iterative numerical design tool for realistic wind turbine blades. All aerodynamic and structural design properties are available in literature. The software tool FAST is used for advanced aero-servo-elastic load calculations and stress-histories are calculated with elementary beam theory. Much focus is put on implementation of actual material S-N data.

The fatigue life model is applied to different blade designs of the NOWITECH 10MW reference wind turbine. Several conclusions can be drawn from the results:

1. Blade damage is tensile dominated due to the poorer tensile fatigue characteristics of the shell glass fiber material.
2. For all designs analyzed maximum damage occurs at high-pressure side, upstream trailing edge, in the region where the blade geometry turns cylindrical. Here, more edgewise deflections are introduced because the edgewise stiffness levels out.
3. To conservatively and fairly predict blade damage mainly caused by interactions between turbulent wind fluctuations and gravitational oscillations, S-N data for R=10 (pure compression), R=-1 (reversed loading) and R=0.1 (pure tension) should be used as minimum in the construction of constant life diagrams (CLDs). The lifetime is over-predicted with 4625% (compared to the 3 *R*-value CLD) by use of a shifted linear CLD (GL2010).
4. The importance of tailored CLDs (more *R*-values) in regions close zero-mean strain/stress is further demonstrated by comparing fatigue analysis where both 3 and 6 *R*-values are used. Damage is over-predicted up to 150% at blade locations where damage from reversed loading typically occurs (e.g. trailing edge) when only 3 *R*-values are used. Consequently, a 3 *R*-value CLD is likely to be conservative and may be used in a preliminary blade design study.
5. State-of-art wind turbine blade material trends are discussed and a parametric study is performed focusing on fatigue performance and material costs. For large blades designed for lower wind speeds, a higher performance material (e.g. carbon) is more favorable in the spar-cap design.

Acknowledgment

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References

[1] E. W. E. Association. (2010). *Wind Energy Factsheets*. Available: http://www.ewea.org/fileadmin/ewea_documents/documents/publications/factsheets/Factsheets.pdf
[2] E. W. E. Association, "The European offshore wind industry key 2011 trends and statistics," 2012.
[3] L. Frøyd and O. G. Dahlhaug, "A Conceptual Design Method for Parametric Study of Blades for Offshore Wind Turbines," 2011.
[4] J. M. Jonkman and M. L. B. Jr., "FAST User's Guide," National Renewable Energy Laboratory2005.
[5] K. Cox, "Structural design and analysis of a 10 MW wind turbine blade," in *Deep Sea Offshore Wind R&D Seminar*, Trondheim, Norway, 2012.
[6] P. K. Fossum, "Aeroelastic analysis of an offshore wind turbine," Project, EPT, NTNU, Trondheim, 2011.
[7] IEC, "IEC 61400-3: Wind Turbines - Part 3: Design requirements for offshore wind turbines," ed, 2009.
[8] IEC, "IEC 61400-1: Wind Turbines - Part 1: Design requirements," ed, 2005.

[9] M. O. L. Hansen, *Aerodynamics of wind turbines*: Earthscan/James & James, 2008.

[10] T. Fischer, W. de Vries, and B. Schmidt, "Upwind design basis," *Upwind deliverable (WP4: Offshore foundations and support structures), Endowed Chair of Wind Energy (SWE) at the Institute of Aircraft Design Universit at Stuttgart,* 2010.

[11] D. J. Laino and A. C. Hansen, "USER'S GUIDE to the Wind Turbine Aerodynamics Computer Software AeroDyn," National Renewable Energy Laboratory under Subcontract No. TCX-9-29209-012005.

[12] T. P. Philippidis and A. P. Vassilopoulos, "Complex stress state effect on fatigue life of GRP laminates.: part I, experimental," *International journal of fatigue,* vol. 24, pp. 813-823, 2002.

[13] T. P. Philippidis and A. P. Vassilopoulos, "Complex stress state effect on fatigue life of GRP laminates. Part II, Theoretical formulation," *International journal of fatigue,* vol. 24, pp. 825-830, 2002.

[14] ASTM, "E1049-85(2011)e1 Standard Practices for Cycle Counting in Fatigue Analysis," ed, 2011.

[15] *Rainflow Counting Algorithm - File Exchange - MATLAB Central*. Available: http://www.mathworks.com/matlabcentral/fileexchange/3026-rainflow-counting-algorithm

[16] M. A. Miner, "Cumulative damage in fatigue," *Journal of applied mechanics,* vol. 12, pp. 159-164, 1945.

[17] R. P. L. Nijssen, "Fatigue life prediction and strength degradation of wind turbine rotor blade composites," *Contractor Report SAND2006-7810P, Sandia National Laboratories, Albuquerque, NM,* 2006.

[18] T. Burton, D. Sharpe, N. Jenkins, and E. Bossanyi, *Wind energy: handbook*: Wiley Online Library, 2001.

[19] D. Van Delft, G. De Winkel, and P. Joosse, "AIAA-97-0951 Fatigue behaviour of fibreglass wind turbine blade material under variable amplitude loading," in *AIAA/ASME Wind Energy Symposium*, Reno, Nevada, 1997.

[20] J. Schön and T. Nyman, "Spectrum fatigue of composite bolted joints," *International journal of fatigue,* vol. 24, pp. 273-279, 2002.

[21] J. F. Mandell and D. D. Samborsky, "DOE / MSU Composite Material Fatigue Database," Montana State University2009.

[22] A. P. Vassilopoulos, B. D. Manshadi, and T. Keller, "Influence of the constant life diagram formulation on the fatigue life prediction of composite materials," *International journal of fatigue,* vol. 32, pp. 659-669, 2010.

[23] D. D. Samborsky, T. J. Wilson, and J. F. Mandell, "Comparison of tensile fatigue resistance and constant life diagrams for several potential wind turbine blade laminates," *Journal of solar energy engineering,* vol. 131, p. 011006, 2009.

[24] K. O. Merz, "Conceptual Design of a Stall-Regulated Rotor for a Deepwater Offshore Wind Turbine," PhD, Department of Civil and Transport Engineering, NTNU, Trondheim, 2011.

[25] A. T. Echtermeyer, "Fatigue of Glass Reinforced Composites Described by One Standard Fatigue Lifetime Curve," presented at the EWEA Conference, 1994.

[26] DNV, "DNV-DS-J102 Design and Manufacture of Wind Turbine Blades, Offshore and Onshore Wind Turbines," in *App. H*, ed, 2010.

[27] D. A. Griffin, "Blade system design studies Volume I: Composite technologies for large wind turbine blades," *SAND2002-1879, Sandia National Laboratories, Albuquerque, NM,* 2002.

[28] J. Peeringa, R. Brood, O. Ceyhan, W. Engels, and G. d. Winkel, "Upwind 20MW Wind Turbine Pre-Design - Blade design and control," 2011.

[29] G. Lloyd, "IV Rules and Guidelines Industrial Services," in *1 Guideline for the Certification of Wind Turbines*, ed, 2012.

[30] J. Wedel-Heinen, J. K. Tadich, C. Brokopf, L. G. J. Janssen, A. M. v. Wingerde, D. R. V. Delft, C. W. Kensche, T. P. Philippidis, P. Brøndsted, A. G. Dutton, and R. P. L. Nijssen, "Reliable Optimal Use of Materials for Wind Turbine Rotor Blades," 2006.

[31] R. Stewart. (2012, January 24) Wind turbine blade production – new products keep pace as scale increases. *Reinforced Plastics* Available: http://www.reinforcedplastics.com/view/22657/wind-turbine-blade-production-new-products-keep-pace-as-scale-increases/

[32] J. Gotro. (2011, May 27, 2012). Polymer Challenges in the Wind Turbine Industry. *Polymer Innovation*. Available: http://polymerinnovationblog.com/polymer-challenges-in-the-wind-turbine-industry-3/

[33] J. Sloan. (2011, May 27, 2012). Carbon fiber market: Cautious optimism. Available: http://www.compositesworld.com/articles/carbon-fiber-market-cautious-optimism

[34] V. W. S. A/S, "Vestas V164-7.0MW Offshore Product Brochure " 2011.

[35] L. Peters, G. Adolphs, J. I. Bech, and P. Brøndsted, "HiPer-tecTM WindStrandTM: A New Generation of High Performance Reinforcement," in *27th Risø International Symposium on Materials Science: Polymer Composite Materials for Wind Power Turbines*, Roskilde, Denmark, 2006.