Fatigue Behavior and Safe Life Analysis of Composite Wind Turbine Blades under Constant and Variable Spectrum Loading

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Abstract

Damage analysis of wind turbine blades made of composite materials (fiber glass) requires a detailed description of the fatigue load spectrum and fatigue behavior. This can be achieved by introducing an experimental investigation on rotating windmill model. The experimental data collected from the change of air speed, rotor speed and amplitude moments together with the ultimate moment obtained from a classical bending test were used to build a theoretical M-N relationship (applied amplitude moment- number of cycles to failure). A theoretical fatigue slope of (12) was found to give the best fit to the experimental constant amplitude fatigue results.

The proposed theoretical M-N relationship and Palmgren–Miner's rule were used to predict the safe life for the blades of the windmill model. Visual observation has shown that the windmill failure was due to de-lamination at adhesive joints. In such cases, amplification of the amplitude moment by five times has given reasonable and safe life predictions for the windmill.

Key Words: fatigue failure, windmill blades, composite materials

Nomenclature

Symbol	Definition	Units
B	Number of blades	-
c	Chord length of blade	m
C_D	Drag coefficient	-
C_L	Life Coefficient	-
d	Diameter of rotor	m
D	Damage parameter	-
F	Bending force	N
i	First level of variable stressing	-
j	Last level of variable stressing	-
L	Blade length	m
M_a	Amplitude moment	N.m
Map	Maximum applied moment	N.m
M_m	Mean moment in a loading cycle	N.m
M_u	Ultimate moment	N.m
m	Fatigue slope	-
n	Number of test cycles	cycles
N	Number of cycles to failure	cycles
r	Radial distance from rotor axis to root of the blade	m
R	Load ratio	-
R_a	Arching radius	m
R_m	Radius of the model	m
R_r	Radius of rotor	m
V_r	Rated wind speed	m/s
α	Angle of attack (i.e. angle of incidence)	deg
α_{opt}	Optimum angle of attack	deg
β	Blade setting, twist angle	deg
λ_d	Design tip speed ratio of rotor	-
Φ	Relative flow angle	deg

Subscript

t	Total
max	*Maximum*
min	*Minimum*

Introduction

The engineer's perception of the phenomenon of fatigue is closely linked with the behavior of homogeneous isotropic metallic materials, that there has often been a tendency to treat modern fiber composites as though they were metals. Unlike metals, composite materials are inhomogeneous and anisotropic. They accumulate damage in a general rather than a localized fashion and failure does not always occur by the propagation of a single macroscopic crack. The micro-structural mechanisms of damage accumulation, including fiber breakage and matrix cracking, de-bonding, transverse-ply cracking and de-lamination, occur sometimes independently and sometimes interactively, and the predominance of one or the other may be strongly affected by both materials variables and testing conditions [1].

At low level of stress in monotonic loading or early in life during cyclic loading, most types of composites sustain damage. This damage is distributed throughout the stressed region and although it does not always immediately reduce the strength of the composite, it's often reduces stiffness. Such strength reduction as might occur is sometimes off-set in early stage of life by slight increase in strength. These increases may be a result of the slightly improved fiber alignment which follows small, stress-induced, visco-elastic or creep deformations in the matrix. Later in life the amount of damage accumulated in some region of the composite may be so great that the residual load-bearing capacity of the composite in that region falls to the level of maximum stress in fatigue cycle and failure ensues. This process may occur gradually when it is simply referred to as degradation, or catastrophically, when it is termed 'sudden death' [2].

Engineers require knowledge of the effect of the main variables on the life of the product or component in order to ensure they are safe for the intended service life or the component can be replaced prior to failure. To satisfy this need,relationships between the stress and number of cycles to failure are needed to establish the total damage and the resistance of materials to repetitive loading [3, 4].

The objective of this research is to identify blade's failure and to estimate safe life of the windmill turbine under simplified load spectrum, in which the life of the blade and cast up in years can be assessed. An experimental technique has been introduced to achieve these objectives.

Theory

The Simplified Load Spectrum Method

The goal of the simplified load spectrum method is to derive an equivalent damage fatigue test that is only dependent on the design load spectrum. Using this method, the test loads are derived by assuming that the blade geometry and material fatigue properties are not accurately known. Then the test article is allowed to (and probably does) deviate from the intended design without affecting the test conditions.

This method remains strictly in the load domain. Therefore, there is no conversion to stresses so that instead of S-N curve there will be M-N curve (applied moment versus allowable cycles to failure). The curve will be defined by the following relationship [5]:

$$Ma = M_u N^{(1/m)} \tag{1}$$

The damage associated with variable amplitude loading can be calculated by applying Palmgren-Miner's rule as follows:

$$\text{Damage} = \sum_{i=1}^{j} \frac{n_i}{N_i} \tag{2}$$

The load ratio is defined as the ratio of minimum to maximum moment in one load cycle.

$$R = M_{min} / M_{max} \tag{3}$$

In terms of mean and amplitude moments, the load ratio takes the form:

$$R = \frac{Mm - Ma}{Mm + Ma} \tag{4}$$

To get the equivalent damage test-load, the test conditions must be specified and typically this includes test load ratio (R_t), number of test cycles (n_t) and total damage parameter (D_t). When these parameters are known, the test amplitude load and the mean load can be calculated [5, 6]. In this investigation, it is assumed that the test load ratio is (-1).

Rotor Blade Model

A horizontal axis wind turbine (HAWT) rotor model was designed and constructed for the purpose of the present investigation. The performance of (HAWT) rotor is greatly influenced by the magnitudes of the chord length and twist angle. The chord length and twist angles are governed by the relative flow angle at operating angle of attack. The values of local speed ratio and circulation reduction factor were selected according to the ideal approach design of (HAWT) rotors. The (HAWT) rotor used in the present investigation was selected according to the wind energy data given in [7]. The data of wind energy potential in Iraq as given in [8] were used to select the windmill that match the wind energy available. The suitable type of windmill rotor recommended by [9] is for common cut–in wind speed of (2.8 m/s) and cut–out (furling) wind speed of (12m/s).

The low speed blades selection methodology introduced by [10] has been used to determine the windmill rotor blade geometry. According to this method,the initial parameters required to obtain the blade geometry are those of design wind speed (V_r)(rated wind speed), rotor diameter (*d*), design tip speed ratio (λ_t) and number of blades (*B*). Since the wind speed is an important part in the calculation procedure of the wind turbine rotor, its value depends upon wind data information for a long period of time and this was taken from [8]. The selection of (V_r) basically determines the cut-in and cut-out speed, at this stage the turbine rotor diameter is chosen from information tabulated in [11] and [12], in which theyrepresented the effect of (C_D/C_L) ratio on the maximum coefficient of power for the rotor with an infinite number of blades. The results are good in selecting (λ_d) by taking the minimum (C_D/C_L) ratio as an input. The tip speed ratio is therefore chosen according to the minimum ratio of (C_D/C_L). This will be done with the help of the data collected by [13].The number of blades is chosen with the aid of data given by [14],with respect to the chosen design tip speed ratio (tip speed ratio with respect to number of blades and optimum angle of attack). In the design procedure, the blade is divided into a number of radial elements; each element represents a radial station. The blade geometry (i.e. blade chord (c) and twist angle (β)) are determined as described by [15]. Due to the potential lift differential along the blade resulting primarily from speed variation, blades at the present investigation are designed with a twist. Blade twist provides a higher pitch angle at the root, where speeds are low and lower pitch angles being nearer to the tip, where speed is higher. This design helps to distribute the lift more evenly along the blade. It increases both the induced air velocity and the blade loading near the inboard section of the blade. The twist angle of the blade for any radial station is determined as follows [10]:

$$\beta = \Phi\ \alpha_{opt} \quad (5)$$

where (α_{opt}) is the incidence angle for the profile section of the blade at which the *(C_D/C_L)* ratio is minimum. It was found that (8°) incidence angle is an optimum angle with *(C_D/C_L= 0.06)* as minimum for (10%) curved plate blade section. The number of blades was found to be (B=18) taking into account the effect of circulation reduction factor in the design process. The tip speed ratio used in the present prediction is (1.5), according to the number of blades.The blade chord Reynolds numbers are fixed for the design condition in the range of (10^4 to 10^5). The blade chord was calculated at (r=0.7 R_r) according to [9]. The arching radius of the blade (R_a) was calculated according to [13].

The aerodynamic forces and moments exerted on each blade were calculated according to the blade element theory. The input data to carry on calculations are the aerodynamic parameters such as maximum lift and drag coefficients according to blade configuration and Reynolds number.

Experimental Investigation

The Windmill Model

The advantage of laboratory testing is that the load amplitude may be increased to accelerate the level of damage per load cycle by as much as two orders of magnitude over the design condition in order to achieve the same total damage in a fraction of the time. As a matter of kinematic similarity, it has been recommended [15] that the model can be tested with wind speeds approximately equal to that of full size model environment and the tip speed ratio must be the same for the proto-type and model. If the similarity of the model and prototype are fulfilled, the expression of the torque, power and thrust coefficients are the same for both.

The experimental model was subjected to different air and rotor speeds according to the simplified load spectrum methodin orderto derivean equivalent damage fatiguetest that will be only dependent on the design-load spectrum. This work benefited from the generation of air in the wind extruder designed and constructed by [7].

Figure (1) shows the windmill model assembly. It consists of three main parts; a hub made of steel, a (225 mm) diameter ring made of a brass wire (5 mm) in diameter which is fixed on the blades by an adhesive material to increase rigidity, and the blades. The blades of the windmill model were made of composite material (fiberglass). The chemical composition of glass fibers and mechanical properties of the commercial glass mat-polyester resin used to manufacture the blades are shown in table (1). A duct of galvanized steel in the form of an arc was used to direct the generated air to the wind turbine. The windmill rotating speeds and acceleration with respect to different wind speeds were measured using a tachometer and vibration analyzer type SBS 1700. The wind speeds have been measured by using standard Pitot - static tube located just in front of the windmill model. Finally the windmill model was balanced at different low air speeds in the wind extruder, with the aid of a laser vibration analyzer, type SBS 1700.

Tensile and Bending Tests

Tensile specimens were manufactured from the same material of the blades and have the geometry and dimensions shown in figure (2). Tensile test sample's have different thicknesses (2.0, 2.5, 2.75, 3.0 and 3.5 mm). Mechanical testing indicated that the 3.0 mm thickness samples have the best tensile strength and modulus of elasticity, (97.8 Mpa) and (4.1 Gpa) respectively. Therefore, this thickness was selected to manufacture the blades. All the tensile tests were carried out at the Specialized Institute for Engineering Industries located in Baghdad.

In order to obtain ultimate moment of the blade, a 3-points bending test was performed. Two specimens were selected for this test, which have the same dimensions of the windmill blade model (230 mm long, 20 mm wide and 3 mm thick). The specimens were simply supported from their ends and the applied force was exerted at their center using Schenck tensile equipment. The tensile device gives the relation between the specimen deflection and the applied bending load as shown in figure (3). The maximum bending force obtained from this test was 28.5kg. Thus, for simply supported beam the ultimate moment was calculated according to the well known relation [16],

$$M_u = F_{max} L/4 \quad (6)$$

(Mu) was found to be (16.3875) N.m.

Fatigue Tests under Constant and Variable Amplitude Loading

A fatigue test specimen of standard geometry is shown in figure (4). Fourteen specimens were manufactured by using punch and die production technology. The samples are of the same material as the wind turbine blades. To estimate the number of cycles to failure experimentally, a fatigue test machine type (Avery 7305), speed 1400 rpm has been used. The type of test associated with this machine is the completely reversed bending. A cycle counter fixed on the machine registers the number of cycles to failure. Ten specimens were tested under constant amplitude loading to construct the M-N curve. Four specimens were tested under variable amplitude loading. Two tests were of the 5-step increasing block loading program (4, 4.5, 5, 5.5 and 6 N.m.) and two tests of the 5-step decreasing block loading program (6, 5.5, 5, 4.5 and 4 N.m.). The number of applied cycles at any loading step within the test program is constant and equals to 75000 cycles.

Results and Discussion

Fatigue Damage under Constant Amplitude Loading

The general form of the M-N relationship, as stated previously in equation (1), is

$$M_a = M_u N^{(-1/m)}$$

Substituting the value of M_u found from the bending test, equation (1) becomes:

$$M_a = 16.3875 N^{(-1/m)} \ldots\ldots (6)$$

Figure (5) shows the M-N relationship for the range of speeds used in this investigation (15-442) rpm at three different theoretical slopes of 8, 10 and 12. This figure shows a good understanding analysis of the windmill blades failure without the need for information about the blades specifications (material and geometry), and could forecast failure depending on the values of loads on the blade. The fatigue life varies inversely with the amplitude moment and for the same moment, fatigue life increases with increasing fatigue slopes. The designers assumed a generalized fatigue slope for windmill blade material of (m = 10), which is a widely used value for good performance on a basic unidirectional laminate [5].

To clarify the issue of fatigue slopes experimentally, a number of constant amplitude fatigue tests have been carried out. The experimental values are represented on the M-N curve shown in figure (6) and on log-log scale in figure (7). The experimental M-N relationship is as follows:

$$Ma = 20.516 N^{(0.0942)} \quad (7)$$

The experimental value of the fatigue slope (*m*) in this relation is m=1/0.0942=10.6157. It is really striking that the value of the theoretical constant (M_u) was found to be (16.3875).

To verify the general validity of the theoretical M-N model, figures (5) and (8) show comparisons between the experimental constant amplitude fatigue results and the theoretical model at three fatigue slopes (8, 10 and 12). All the experimental data lies very close to the theoretical results of the fatigue slope (m = 12). Hence the theoretical model takes the following final form:

$$Ma = 16.3875 N^{(1/12)} \quad (8)$$

Slopes less than 8 and approaching 6 were seen in poor performing fabrics and laminates as well as in laminates with fabrication flaws such as resin- and fiber-rich areas, ply drops and complex geometry. Slopes greater than 10 and approaching 12 are widely reported for good performance unidirectional laminates [17]. Fatigue failures of turbine blades occur in many locations by various modes e.g., skin laminate failure in tension or local buckling, adhesive failure in shear or peel, or bolted joint failure. Each failure mode could be characterized by a different M-N curve. A family of curves could be defined to bind the potential M-N combination for the turbine's blades [17].

In this investigation, a range of magnifications of moments from ($1.5 M_{ap}$) to ($5 M_{ap}$) was used. The (1.5) factor was selected for the lower bound curve, which matched 100 % extreme static test level and extreme design limit load, while the (5) factor was arbitrary selected for the higher bound to account for conservative design [5].

The family of M-N curves obtained for all the speeds investigated are shown in figure (9) at 3-levels of moment magnification (1.5, 3.0 and 5.0) and at 3-levels of fatigue slopes (m = 8, 10 and 12). These curves are very useful and will be used to predict the operational life of rotating windmill turbine blades according to the fatigue failure mode.

Cumulative Fatigue Damage under Spectrum Loading

Fatigue design and analysis of wind turbine systems require good and reliable knowledge of loads [17] that act on the system especially on the blades. The fatigue life of a horizontal axis wind turbine system was estimated by using the well-known S–N damage equation [18]. Fatigue under variable amplitude loading is generally caused by dynamic effects and wind variations [17]. This depends on the statistical nature of wind. To analyze the windmill turbine blades which are subjected to spectrum loading, the original linear damage accumulation rule according to Palmgren and Miner was used in addition to the proposed theoretical M-N relationship.

It has been originally suggested that the total damage at failure is one. It has to be realized, however, that values of (D) at failure may be either substantially smaller or larger than one depending on the actual combination of the relevant load spectrum, the material system, the component geometry, etc.

Figures (10) and (11) show details of damage analysis of the variable amplitude tests, and table (2) summarizes the total experimental damage for the four tests. It can be seen that for both types of spectrum loading in this investigation, increasing and decreasing, the actual damage value at failure is slightly more than one. Consequently, a design based on a damage value of one, i.e. Palmgren-Miner rule, is considered safe since it predicts failure to occur at lower lives than the actual lives that can be tolerated in service.

The direct application of Miner's rule and the theoretical M-N relationship to predict the lifetime of the wind turbine under spectrum loading has failed to make acceptable predictions. In actual fact, such predictions have shown that the windmill turbine can tolerate much higher life than the actual life. It should be mentioned however that the M-N relationship has been derived for the blade material and visual observations have shown that the windmill failure was due to shearing or de-lamination at adhesive joints used to fix a steel ring to the 18-blades turbine as shown in figure (12).

The adhesive material was of the same composite material used to manufacture the blades, and failure of few bonded joints led to excessive vibration of the wind turbine. The critical section that leads to failure is in fact in a stress concentration region that could be characterized by a different M-N curve. Magnification of the nominal load is needed to compensate for the stress concentration. After several trials, magnification of the amplitude moment by (5) times was selected to account for conservative design for the windmill turbine.

Figure (13) shows a comparison between the cumulative damage predictions according to the theoretical and experimental M-N equations after amplitude moment magnification. The theoretical predictions are very close to the experimental predictions. In both cases, the predicted damage at failure is higher than one, which indicates safe prediction.

Conclusions

Fatigue life varies inversely with air velocities and amplitude moments. A theoretical M-N relationship (amplitude moment-number of cycles to failure) has been proposed for the windmill blade composite material, and a theoretical fatigue slope of (12) was found to give the best fit to the experimental constant amplitude fatigue results.

The proposed theoretical M-N relationship and Palmgren-Miner's rule can be used to predict the fatigue life of the windmill rotor under spectrum loading.

Visual observation has shown that the windmill failure was due to de-lamination at adhesive joints. In such cases, amplification of the amplitude moment by five times has given reasonable and safe life predictions for the windmill.

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Table (1) Chemical composition (weight%) of glass fiber and typical properties of the commercial glass mat-polyester resin.

E-glass composition

Silicone oxide	54.3%
Aluminum oxide	15.2%
Calcium oxide	17.2%
Magnesium oxide	4.7%
Sodium oxide	0.6%
Boron oxide	6.0%

Properties of commercial glass mat-polyester resin

Glass content	(30 – 70)%
Specific gravity	1.4 – 2
Tensile strength	(70 -340) Mpa
Modulus of elasticity in tension	(5.5 – 30) Gpa
Barcol hardness	(40 – 65)Mpa

Table (2) Results of increasing and decreasing spectrum loading program.

Specimen No.	Type of loading	No. of program to failure	N_f cycles	Damage experimental
1-	Increasing	4.640	1740000	1.09705
2-	Increasing	4.181	1568000	1.09810
3-	Decreasing	3.547	1330000	1.01500
4-	Decreasing	4.725	1772000	1.13350

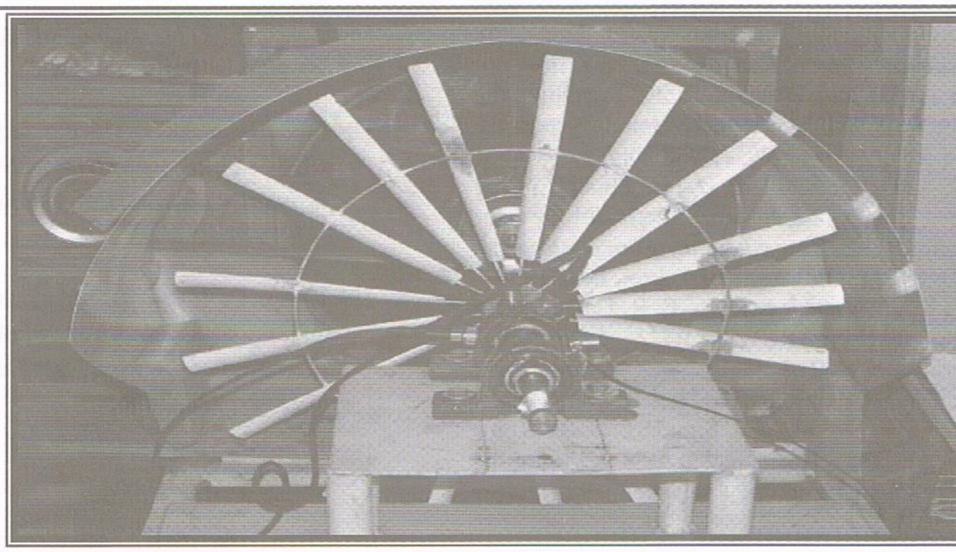

Figure (1) Wind turbine model test rig.

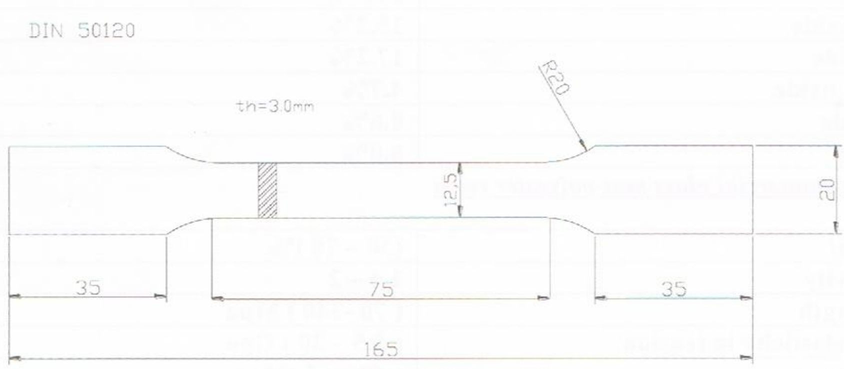

Figure (2) Standard specimen for tensile test (all dimensions in mm).

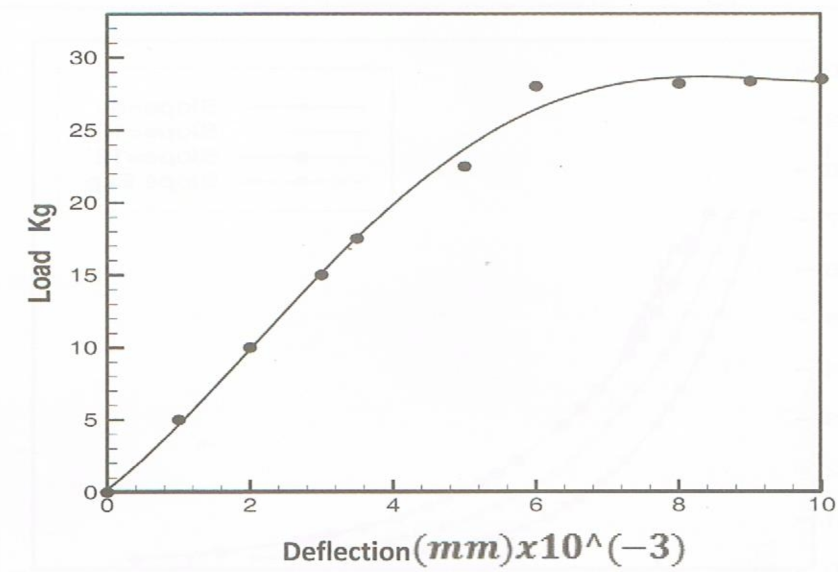

Figure (3) Applied load versus specimen's deflection for the bending test.

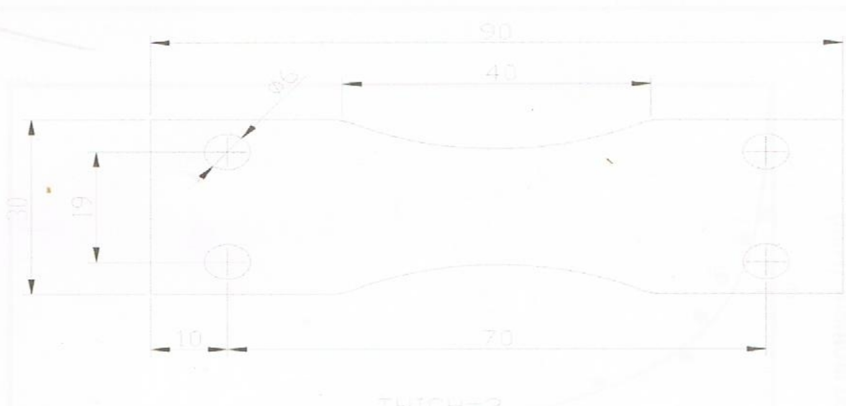

Figure (4) Fatigue specimen geometry (all dimensions in mm).

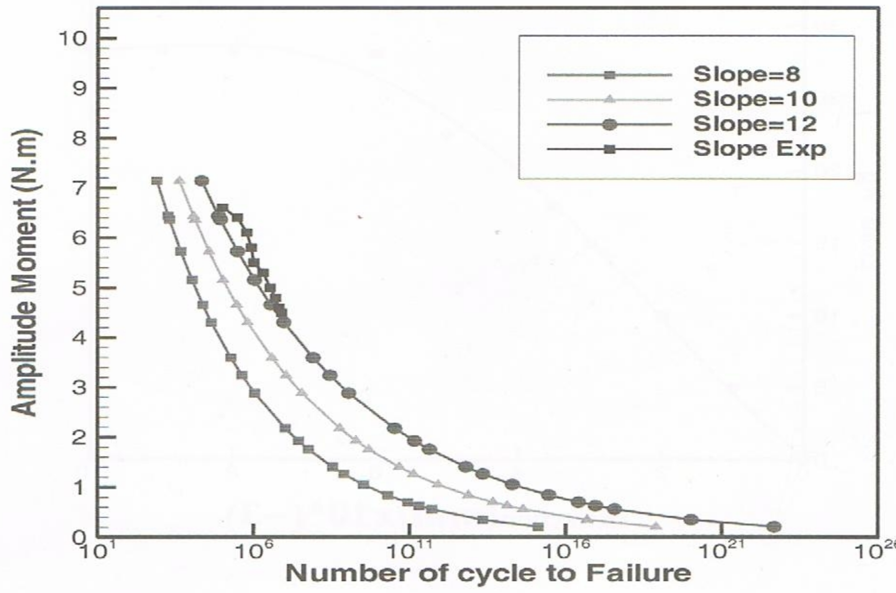

Figure (5) M-N Curves, speeds between (15) and (442) rpm.

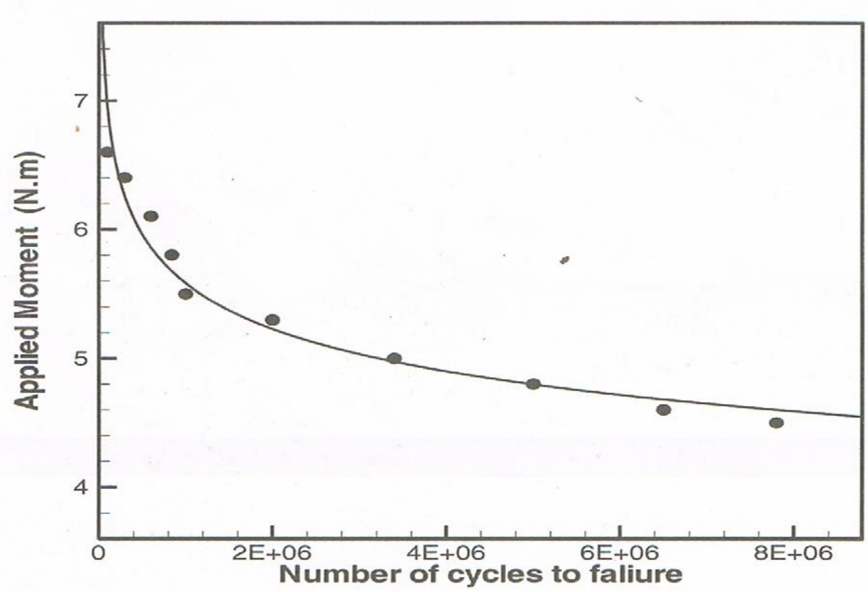

Figure (6) Applied moment versus number of cycles

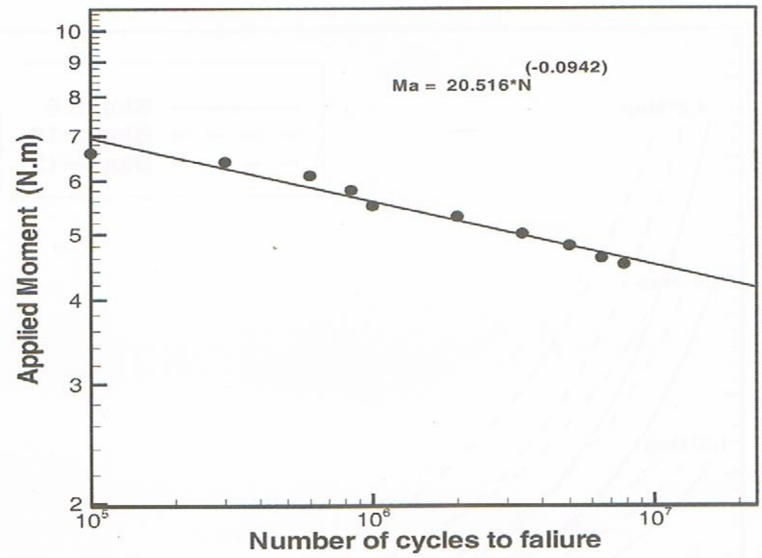

Figure (7) Applied moment versus number of cycles, log-log scale.

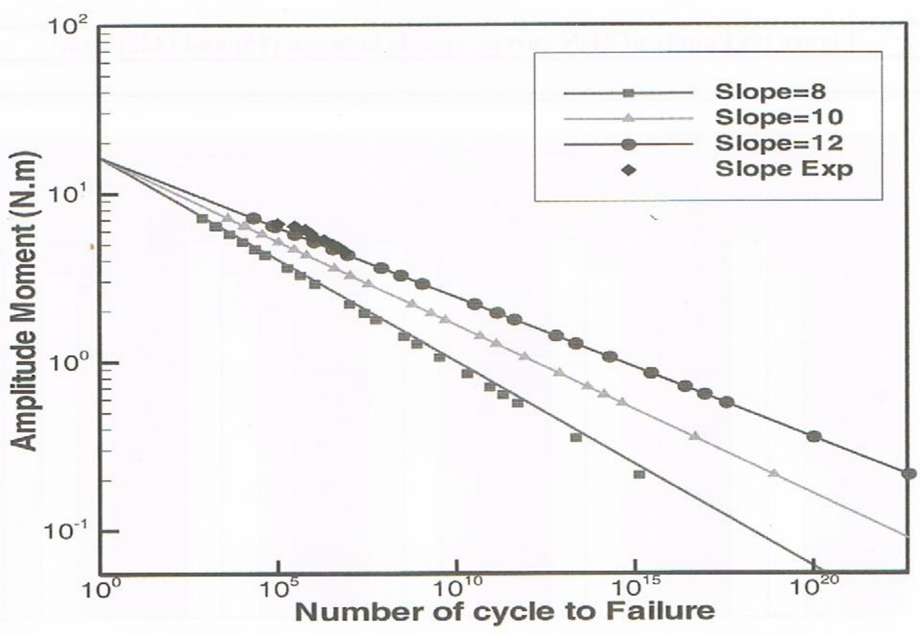

Figure (8) M-N Curves with 4 slopes, log-log scale.

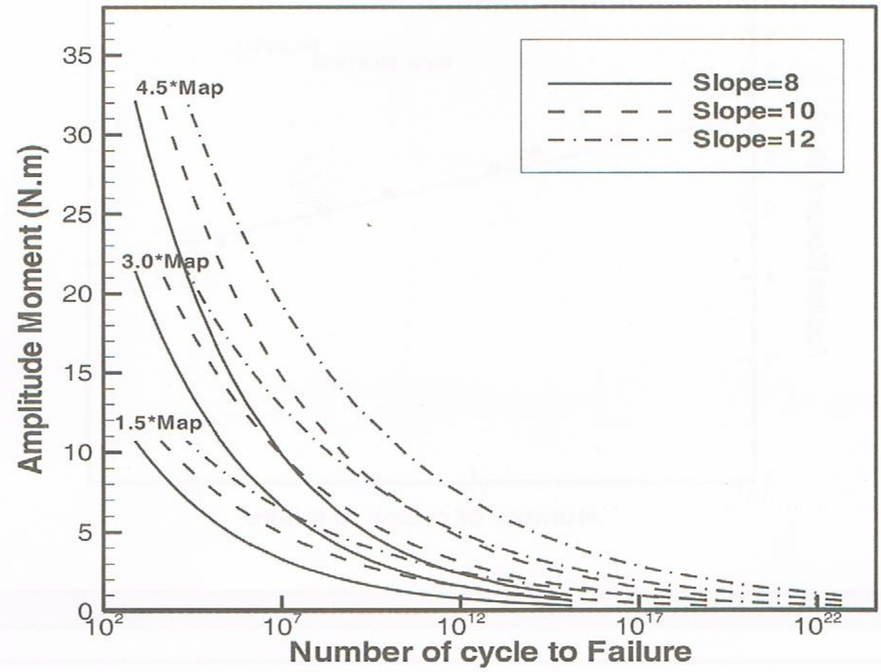

Figure (9) Family of M-N curves, speeds between (15) and (442) rpm.

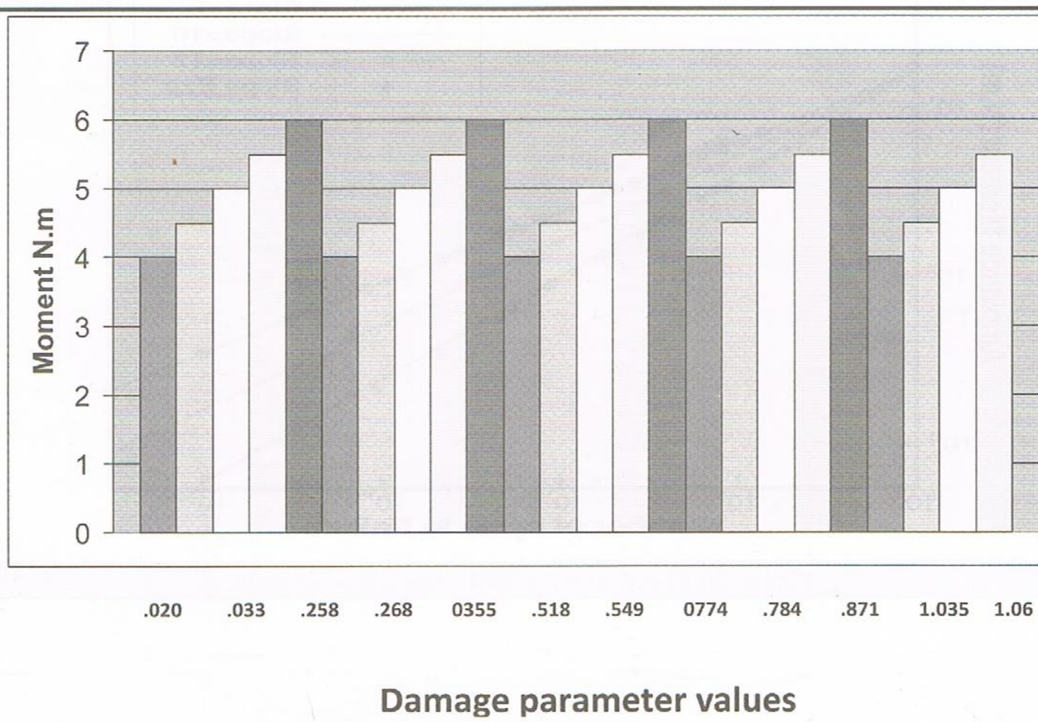

Figure (10) Damage parameter (increasing block loading in test program).

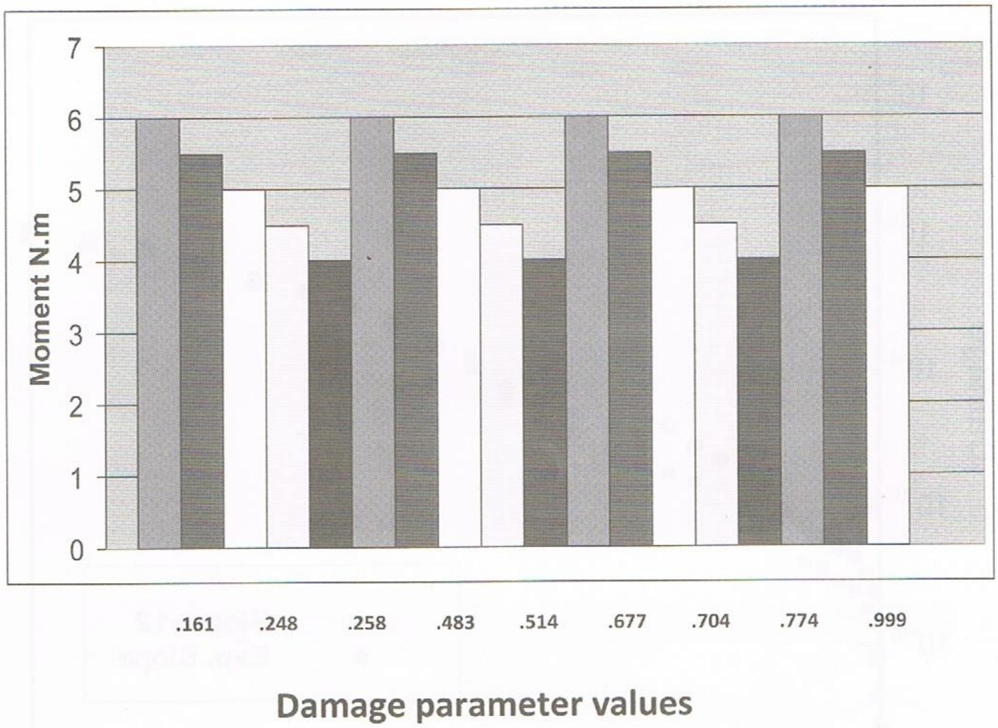

Figure (11) Damage parameter (decreasing block loading in test program).

Figure (12) Failure of wind turbine in bonded joint.

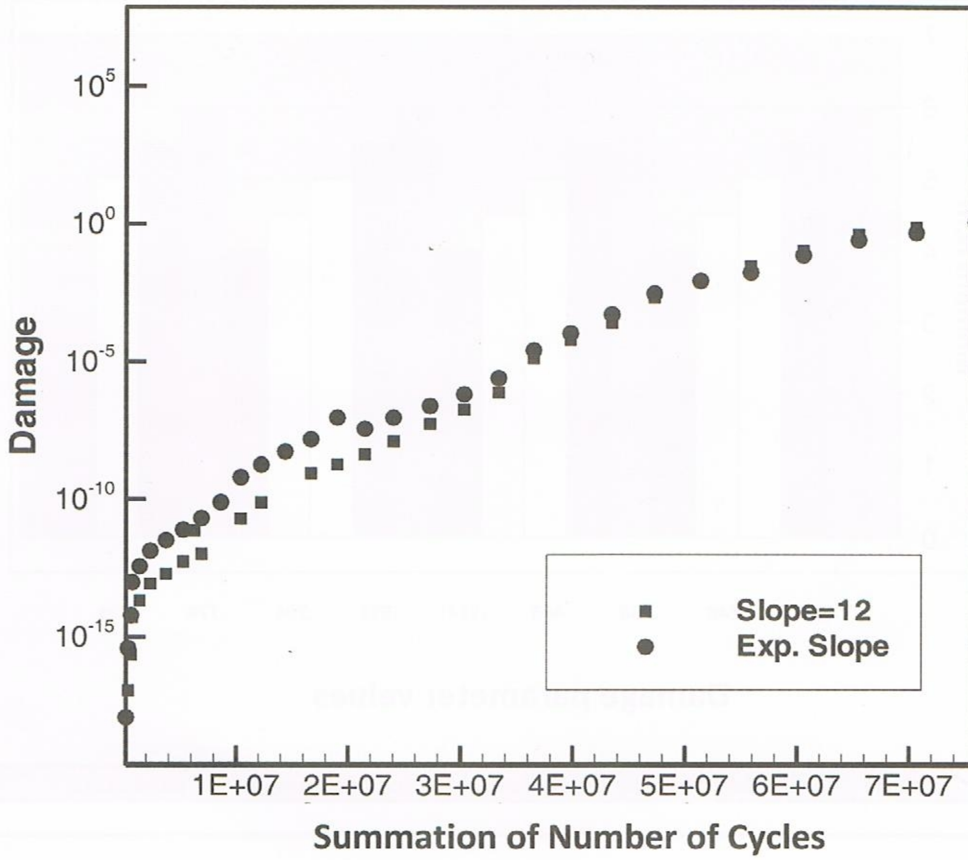

Figure (13) A Comparison between cumulative damage for the wind turbine according to the theoretical and exeperimental M-N equations.

سلوك الكلال وتحليل العمر الآمن لريش دوّار طاحونة الهواء المصنعة من مواد مركبة في ظروف التحميل الثابت والتحميل الطيفي المتغيّر

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الخلاصة

يتطلب تحليل الضرر لريش دوار طاحونة الهواء المصنعة من مـواد مركبة (الصوف الزجاجي) وصف تفصيلي لطيف حمل الكلال وسلوكه. يمكن تحقيق ذلك بأجراء تجارب فحص مختبرية على نموذج دوار طاحونة الهواء. إن المعطيات التجريبية المستحصلة من تغير سرعة الهواء وسرعة الدوار وسعة العزوم إضافة إلى العزم الأقصى المستحصل من اختبار الانحناء التقليدي استخدمت لبناء نموذج رياضي نظري لسعة العزم المسلط وعدد الدورات للفشل واستخدم هذا النموذج مع قاعدة بالمكرن-ماينر (Pelmgren-Miner) لتخمين العمر الآمن لريش دوار طاحونة الهواء.

أثبتت النتائج العملية لكلال ثابت السعة ان القيمة النظرية (12) لمنحنى الكلال أدت إلى أفضل ضبط للنتائج. بينت الملاحظة العينية ان حدوث الفشل لنموذج دوار طاحونة الهواء يكون بسبب فشل وتفكك المادة اللاصقة في منطقة الربط للريشة مع حلقة التثبيت وفي هذه الحالات يؤدي تضخيم العزم لخمسة أضعاف إلى تنبؤ مقبول للعمر التشغيلي الآمن لطاحونة الهواء.