# DESIGN OF LARGE COMPOSITE STRUCTURES UNDER DYNAMIC LOADS WITH FATIGUE ASSESSMENT

José Sancho Rodríguez

*MTorres Diseños Industriales, S.A. Carretera de Huesca Km. 9. 31119 Torres de Elorz (Navarra), Spain*

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## SUMMARY

Every day design of machinery parts and structures is becoming more restrictive in the simplifying hypothesis accepted for the calculations. Aeronautical applications have been pioneers in the field of detailed calculations (non lineal buckling of structures, compression strength after impact, etc.). Nowadays even in industrial applications performing quite detailed and accurate analysis both lineal and non lineal is starting to be mandatory. Parts made out with composite materials and with high responsibility are requiring non lineal dynamic calculations from which fatigue assessment is performed.

Another important aspect when designing a responsibility part with composite materials, is being capable of predicting with enough accuracy the interlaminar stresses appearing in free edges, holes, changes of number of plies, etc. Conventional calculations using classical theory of laminates are not capable of giving useful results, and even in the case of using higher order approximations, results are strongly dependent on the starting hypothesis of the theory used. A final keypoint in interlaminar assessment is the problem of using a failure criterion as conventional criteria are not always accurate enough to predict the onset of delamination.

The present paper presents the experience of designing a large composite structure (windmill blade) taking into account all the previous problems (orthotropic laminated material), performing also dynamic non linear and fatigue analysis.

### STRUCTURES ANALYSED

One of the common industrial applications of composite structures is in the field of energy generation, especially in wind energy [1]. In this kind of structures, glass fiber has always been the most important material for blade manufacturing. Nevertheless, the necessities of manufacturing larger machines capable to be economically profitable in low wind areas make necessary using carbon fiber blades. As cost is an aspect of vital importance in industrial application, the blades must be competitive when compared to conventional glass fiber ones. This point make necessary performing an accurate design to avoid the use of more material of what is necessary, because the balance between higher cost of carbon fiber and their higher characteristics that allow for lower weights can be otherwise broken.

In the present paper the design procedure followed for the main carbon fiber tube of a very long (36 meter radius rotor) blade is shown. Non linear finite element dynamic calculation is performed taking into account failure of individual plies, non lineal behavior of bolts and adhesives and contact between the blade and the extender. Interlaminar stresses are evaluated in highly loaded areas and adhesive joints are also checked to know the stress distribution and the influence of the shape of the flanges on this distribution. Dynamic behavior of the blade

and fatigue strength are of vital importance in the design to ensure the life of the design without loosing production.

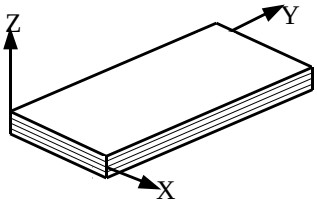
## CLASSICAL DESIGN OF COMPOSITE STRUCTURES

For performing design of structures and parts made out with composite materials, several conventional theories have been used in the past. The most traditional and well known method is the Classical Laminated Theory (CLT), based on the Kirchoff [2-4] hypothesis for shear behaviour that offers a good approach to the design of simple slender parts because some of the stresses and deformations are lost ($\varepsilon_{zz}$, $\varepsilon_{xz}$, y $\varepsilon_{yz}$). The equations of this method are the following ones:

$$u(x,y,z) = u_0(x,y,z) - z\frac{\partial w_0}{\partial x}(x,y)$$

$$v(x,y,z) = v_0(x,y,z) - z\frac{\partial w_0}{\partial y}(x,y)$$

$$w(x,y,z) = w_0(x,y)$$




Nevertheless the previous equations are only valid for simple calculations in which stresses are mainly constrained to appear in every plane (composite layer) of the structure. Unfortunately this is not a very common situation when working with real life structures as there are holes, changes in thickness, out of plane loading, free edges, etc, being every one of them a cause for appearing interlaminar stresses. The final translation of all these problems is the necessity of including non linear variation of the deflections with the thickness of the laminate. Next equation shows basically the way of introducing these non linear terms.

$$u(x,y,z) = u_0(x,y,z) - z\frac{\partial w_0}{\partial x}(x,y) + u_3 z^3$$

$$v(x,y,z) = v_0(x,y,z) - z\frac{\partial w_0}{\partial y}(x,y) + v_3 z^3$$

$$w(x,y,z) = w_0(x,y)$$

In any case, even with the inclusion of the higher order contribution to the equations, there are some discrepancies between theory and experimental results. These discrepancies can be seen in the next figure (fig 1) [5] for different higher order theories, being the 3D the full three dimensional approach or exact solution of the problem.

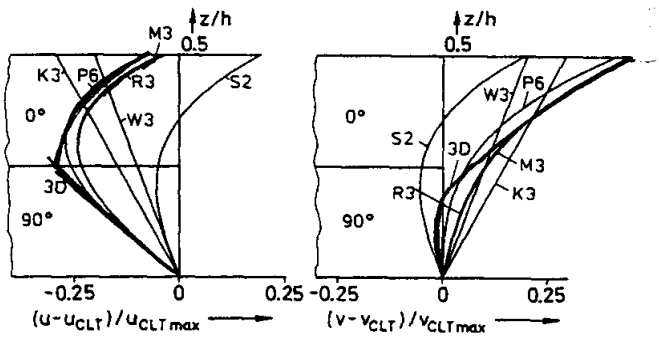



*Fig. 1. Comparison between in plane displacements in a 0/90º laminate with an slenderness ratio (length/thickness) of 5 [5]*

To avoid these problems, a full three dimensional calculation of the composite structure should be performed, the problem is that this kind of calculation is almost impossible to be carried out in most of the practical applications[6].

## ADEQUACY OF THE FINITE ELEMENT METHOD TO THE RESOLUTION OF THE INTERLAMINAR STRESSES SINGULARITIES IN THE FREE EDGES.

It is perfectly known that the Effective Modulus theories used for the mathematical treatment of the free edge problem in laminates, leads to stress singularities in the borders of the samples. Nevertheless, it is also recognized that these singularities are artificially caused by the mathematical equations (an infinite stress is something without sense from the physical point of view). The order of the free edge singularity has been studied and it is perfectly known that only a very small area close to the free edge is affected.

In a parallel way, the finite element method has demonstrated to be unable of predicting the stress singularity in the free edge when used with classical formulations for the elements. New mathematical developments where singularities of different order are included in the shape functions of the elements have been used for these studies. Nevertheless, commercial finite element codes do not include this sort of formulation doing impossible the analysis of real parts with these elements. Besides that, from the theory of the finite element method, it is clear that although the equations are formulated in terms of the nodal degrees of freedom, the only way of solving the integration required in the solution of the equations is through numerical methods. The method generally used is the Gauss one in which several interior points of the element are selected to perform the integration. These points are not the nodes so that, when checking in a finite element code nodal stresses, some sort of extrapolation of the values calculated in the integration points must be made. The extrapolation process may lead to inaccuracies in the results that in the case of this study can be important.

Next figure (fig. 2) shows the differences between nodal stresses (extrapolated) and integration point ones that can be seen in the interlaminar shear stresses distribution in the width direction. Mathematical model used includes a row of three dimensional brick elements per layer of material. Of course by the equilibrium equations, $\tau_{xz}$ and $\tau_{yz}$ should be zero at the free edge although in this case there are some differences both qualitative as quantitative.

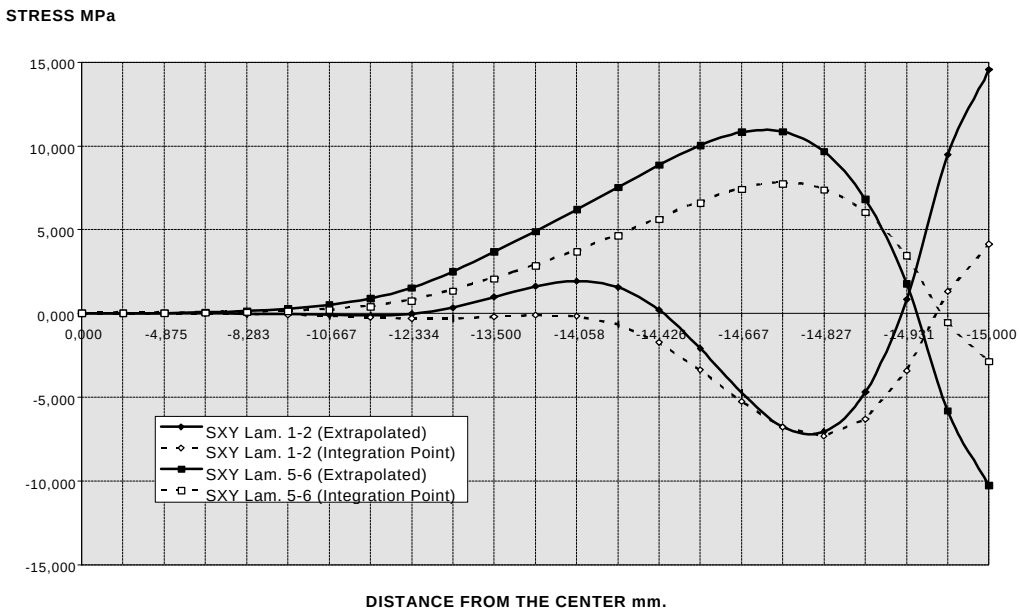



*Fig. 2. Shear Stress* $t_{xy}$ *evolution along width*

Different stacking sequences produce different stress fields and completely different mismatch between stresses in the free edge of the part.

Unfortunately, this approach is not valid for large structures, as the number of elements would be impressive if each layer needs at least one row of elements. The only feasible way to perform finite element analysis of real composite structures at an industrial level is using

laminated shell elements in which higher order theories are implemented and try to refine mesh in the areas of interest.

A possibility of using the latter idea is performing a simple model in the areas far away from the interest zones, using laminated shell elements in these areas. When coming closer to the area that wants to be studied, a finer three dimensional mesh made out with linear solid elements, with at least a row of elements for each layer can be made (figure 3). As at the interface between both regions tying equations are automatically generated by MARC [7], this is a quite simple approach to have a full three dimensional mesh in some areas without the penalty of a huge amount of elements in the entire model.

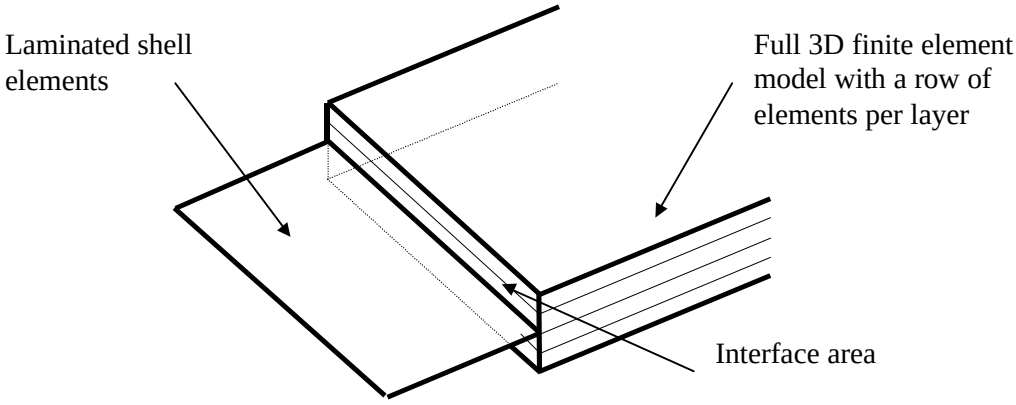
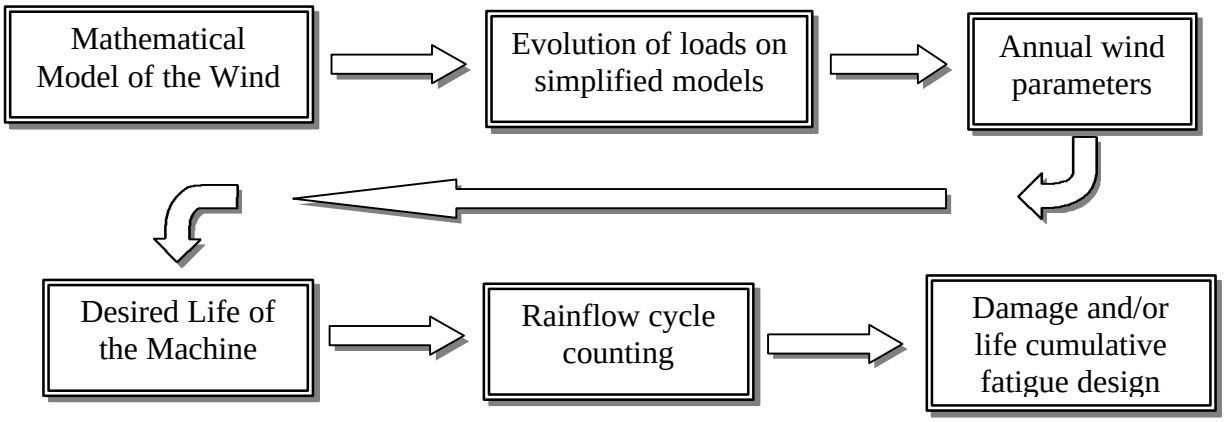



*Fig 3. Detail of the transition between shell and brick elements*

## CALCULATIONS PERFORMED ON THE BLADE OF A WINDMILL

The aforementioned technologies for simplifying the calculations of large composite structures are to be applied on the design of a windmill blade for a rotor of 72 meters diameter being its rated power of 1500 kW. This blade is something different to most of the conventional ones as it uses a carbon fiber main beam that carries most of the loading, while having a more flexible outer skin made out with glass fiber. Interconnection between the main beam and the outer skin is made through ribs. All the blade is designed to be manufactured from unidirectional prepreg materials both for the glass fiber as for the carbon fiber parts. Besides that, manufacturing process will be automatic tape lying on the five axis machine developed by MTorres Diseños Industriales, S.A.
As the machine has serious dynamic requirements (it is a rotating machine that receives fully random loading as the wind varies) and an strong requirement regarding life (20 years), dynamic calculations from which amplified loads are computed must be performed.
The procedure followed is the next one:



For performing the initial steps of the calculation, commercial software can be used. In this case, Bladed for Windows [8] has been applied to the design. Basic issues of interest are the way in which turbulence of the wind is computed. Von Karman models and Kaimal models are available. Basic equations for these models are the following ones:

Von Karmal $$\frac{nS_{uu}(n)}{\sigma_u^2} = \frac{4\tilde{n}_u}{(1+70.8\tilde{n}_u^2)^{5/6}}$$

Where $S_{uu}$ is the auto-spectrum of the wind speed variation, n is the frequency of the variation, $\sigma_u$ is the standard deviation of the wind speed variation, $\grave{o}_u$ is a non dimensional parameter calculated as

$\tilde{n}_u = \frac{n\,{}^xL_u}{U}$ And ${}^xL_u$ is the length scale of turbulence and U the mean wind speed.

The only difference with Kaimal model is the divisor term that becomes $(1+6.0\tilde{n}_u)^{5/3}$

Applying the wind speeds obtained with these methods to a simplified finite element model through aerolastic calculations taking into account the airfoils of the blade, a whole set of varying loads at different points of the wind turbine (not only the blades) are obtained. Of course, gravitational and rotation effects are taken into account. The following figure (figure 4) shows some of these loads (flap bending moments) at the blade root for the three blades.

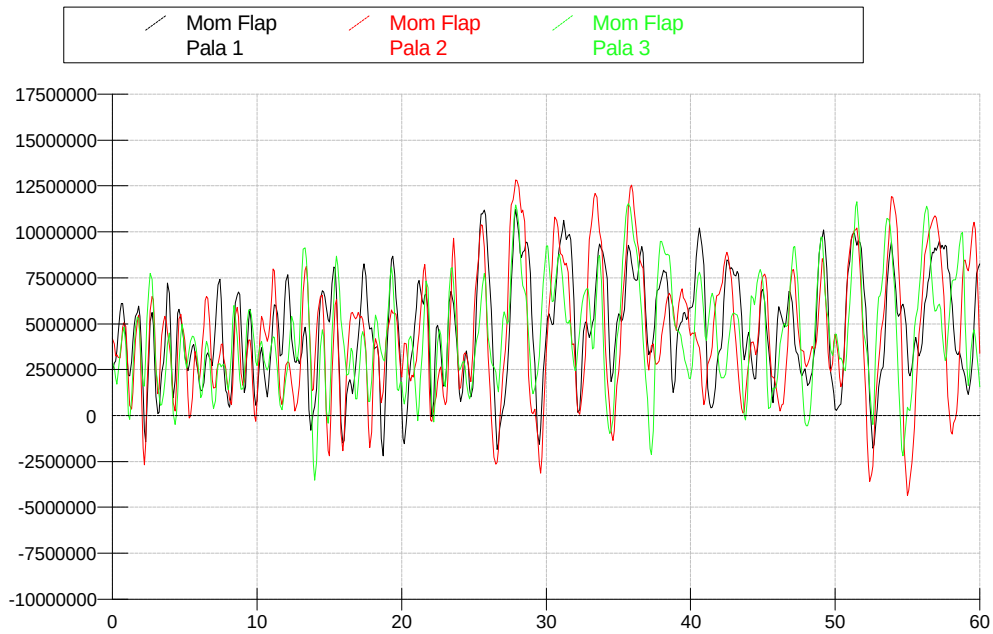



*Fig. 4 Evolution of bending moments on the three blade roots*

These loading sequences are obtained for different wind speeds in the operation range of the wind turbine. Next a Weibull distribution like the following one is used (figure 5).

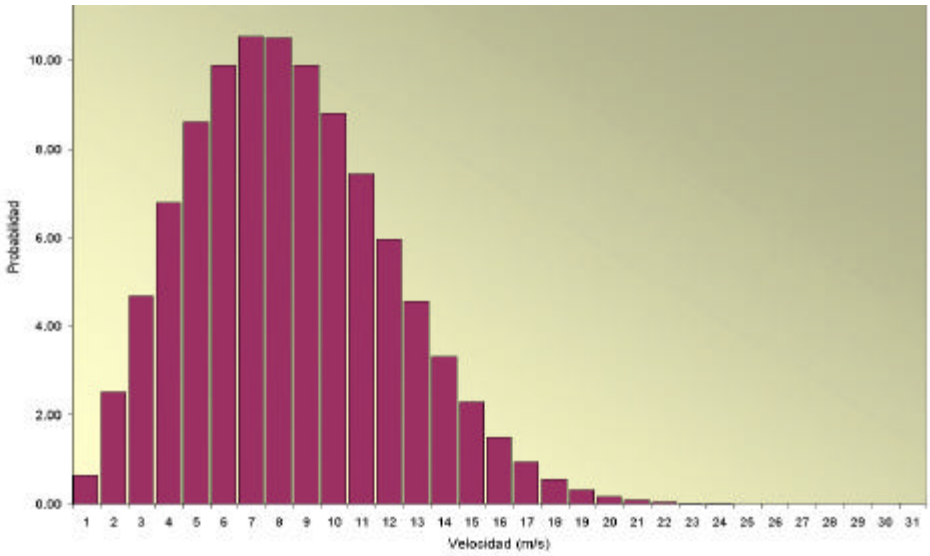



*Fig. 5 Typical Weibull distribution for a windfarm location*

Weibull distribution characterizes the wind over long periods (usually one year that is an standard in wind resource assessment). Weibull function is the following one:

$$F(V) = 1 - e^{-\left(\frac{V}{c\overline{U}}\right)^k}$$

This is a probability function where c and k are parameters to be adjusted for each location, V is the wind speed for which probability is to be calculated, and is the mean wind speed over the period evaluated.

Once the probability distribution of the wind speed is known, joining it with the loading sets already calculated, a rainflow [9] analysis of the loads for a determined life of the wind turbine can be calculated. Next figure shows one of the diagrams obtained from a rainflow cycle counting.

From this point, detailed analysis and calculations of the different parts of the structure of the wind turbine can be performed.

The first part to be analyzed is the blade. In this case the windmill blade differs substantially to other conventional blades. Here, carbon fiber is used for the main beam of the blade while having glass fiber for the outer skins as they are not given any structural responsibility. Second, in this case there are a number of ribs that connect the outer skins to the main carbon fiber beam. The ribs are made in a mixture of glass fiber and carbon fiber to have the lowest warping and twisting possible when they are extracted from the molds. The reason for this distortion in that these ribs are not symmetrical with respect to their mean plane so out of plane displacements appear when cooling down the prepregs in the mold. The effects of this twisting have been studied by means of nonlinear finite element calculations using MARC to find out the best fiber orientation compromise between mechanical requirements and heat distortion in curing process. Next figure (figure 6) shows one of the calculations performed.

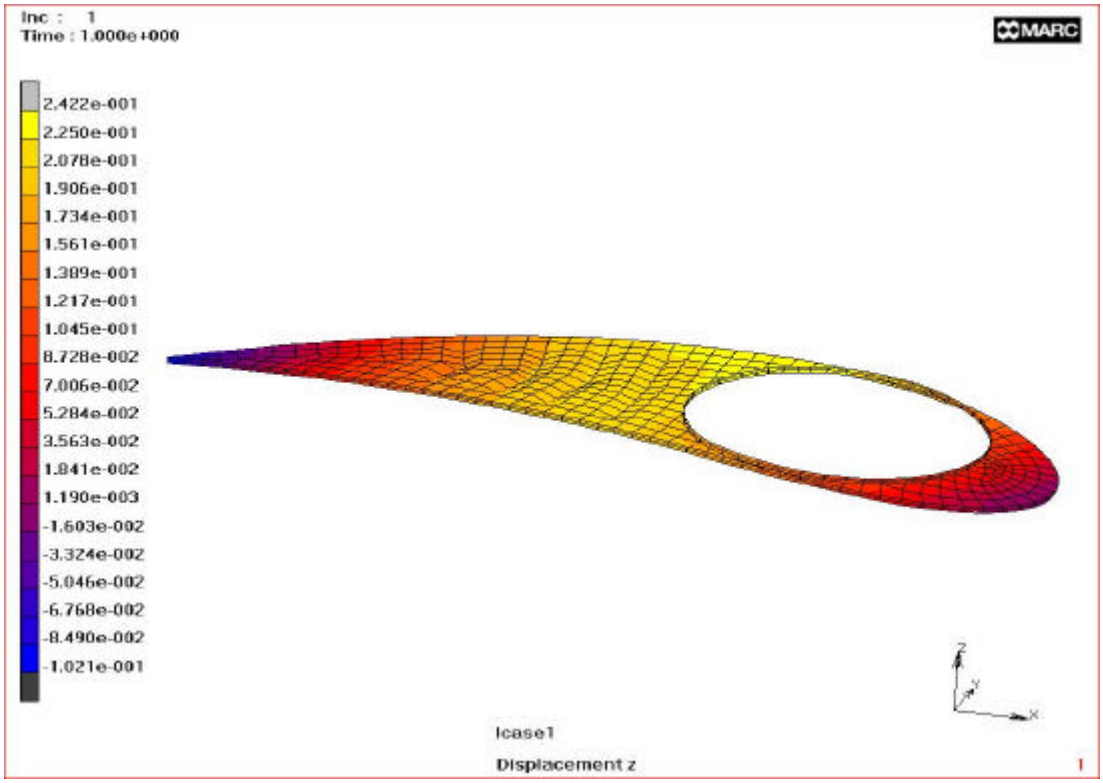



*Fig. 6 Out of plane deformation of a rib caused by the curing process*

Another interesting issue of the ribs is that they are made in one shot with a hot forming process starting from unidirectional prepregs. The fine tuning of this process lets to obtain the final part without additional finishing operations.

Once selected fiber orientation required by manufacturing aspects, the whole blade is calculated to check the behavior of the different laminates and materials used in each part of it. Classical cuadratic failure criterion (Tsai Wu [2]) is used to check for possible failures of the blade. Also some more particular failure criteria are used for ensuring that delamination does not appear as a result of interlaminar normal and shear stresses. Finite element MARC approach has demonstrated to be accurate enough in predicting these stresses (interlaminar shear stresses) [10] even from a laminated shell element approach. Formulation implemented in MARC for Tsai Wu interlaminar failure criterion is the following one [11].

$$\left(\frac{1}{Z_t Z_c}\sigma_z^2 + \frac{1}{R^2}\tau_{xz}^2 + \frac{1}{Q^2}\tau_{yz}^2\right)R^2 + \left[\left(\frac{1}{Z_t} - \frac{1}{Z_c}\right)\left(\frac{1}{h_0}\int_0^{h_0}\sigma_z(y,z)dy\right)\right]R - 1 = 0$$

In this formula, $Z_i$ are strengths in traction and compression, and $h_0$ is the thickness of each lamina. Next figure (figure 7) shows one of the results of a failure criterion calculation on the outer skin of the blade.

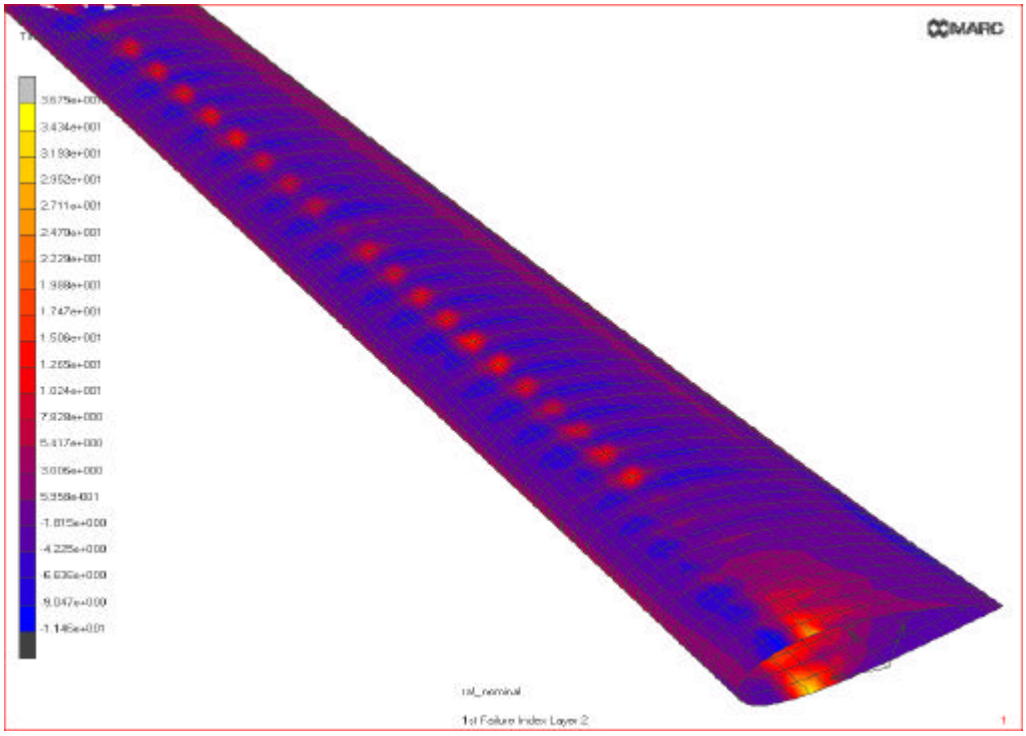



*Fig. 7 Failure criterion contour plot on the outer skin of the blade*

Finally, it is clear that all the loads introduced in the blade by the wing must be addressed to the blade root from which they are passed to the hub trough a blade bearing. Bending moments appearing in this blade root are really impressive as the wind turbine is rated at 1500 kW and it has a rotor diameter of 72 meters with a survival wind speed of 70 m/s. The blade itself has a length of 33 meter.

Blade roots are commonly made in conventional blades either by introducing a metallic part bonded to the glass fiber or by increasing the thickness of the laminate in the blade root to a dimension in which screws can be placed inside and locked to the fiber trough steel inserts.

In this case, a different approach is used as the blade root in this blade is built on the carbon fiber beam. This means that the thickness can not be increased too much in order to avoid high cost of the blade and aluminum can not be used as flanges due to galvanic corrosion between it and carbon fiber.

To overcome these problems the blade root has been designed with two steel flanges joined to the main carbon fiber beam with adhesives and rivets to increase the safety of this joint. One of the flanges is cobonded with the tube and the other is just bonded with a main purpose of avoid the collapse of the adhesive due to thermal stresses caused by the high coefficient of thermal expansion mismatch.

For designing this blade root, the adhesive layer has been simulated in the finite element code MARC, including both its nonlinear behaviour and a progressive failure approach that decreases the stiffness of the adhesive to 10% of the original one when an stress level is achieved.

Next figure (figure 8) shows a detail of the finite element model of the blade root (only one of the flanges can be seen).

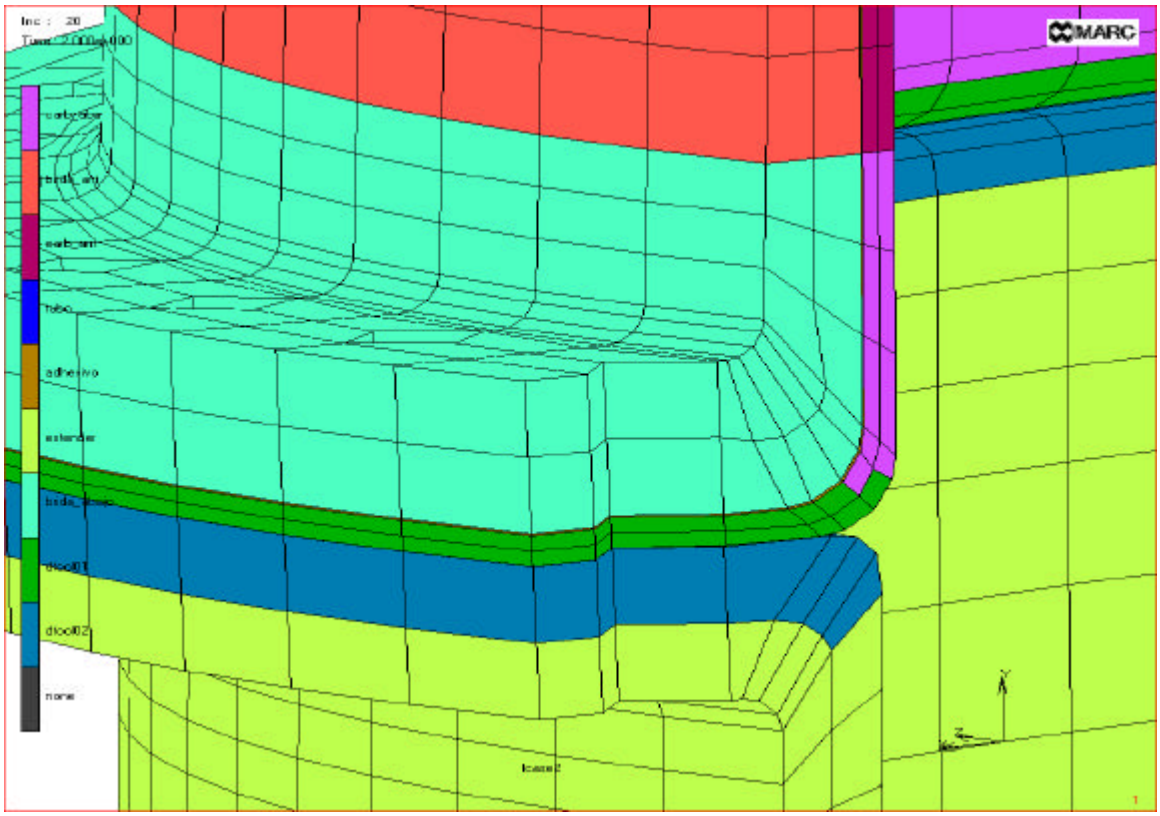


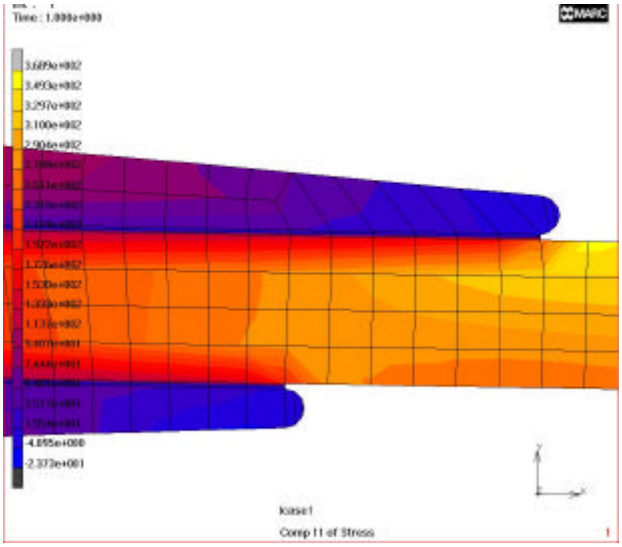

*Fig. 8 Detail of the blade root calculated*

For performing these calculations, the transition between shells and solids explained at the beginning of this article, as well as equivalent single layer theories have been used. This model includes more nonlinearities, specially in the joining with the extender of the blade and the blade itself as contact and nonlinear bolting is included to reproduce the real behaviour of the joint.

With this behaviour bolts only work when they receive tensile stresses but if the load is a compression one, bolts do not work and the load is transferred by the contact between the faces of the flanges.

Of course the mesh of the adhesive in this finite element model does not maintain the connectivity between nodes because this will imply to have very bad aspect ratio in these elements (quite large and high width for a thickness always lower than 0.3 mm. that is the total thickness of the adhesive layer). To avoid this necessity of maintaining connectivity between nodes, another very important feature of MARC finite element software is used. This feature is the glued contact option [12]. With this option a contact between model faces with different mesh densities is specified by means of a contact among them with a certain force to break this contact. Load transition between both parts is ensured so that the major problem with mesh densities is overcomed.

Next figure (figure 9) shows one stress distribution for one case of the blade roots analyzed seeing the whole three dimensional model. Also the same figure (figure 9) includes a detail of the results on a cross section of the blade root showing the adhesive layer between the outer steel flange and the carbon fiber main tube.



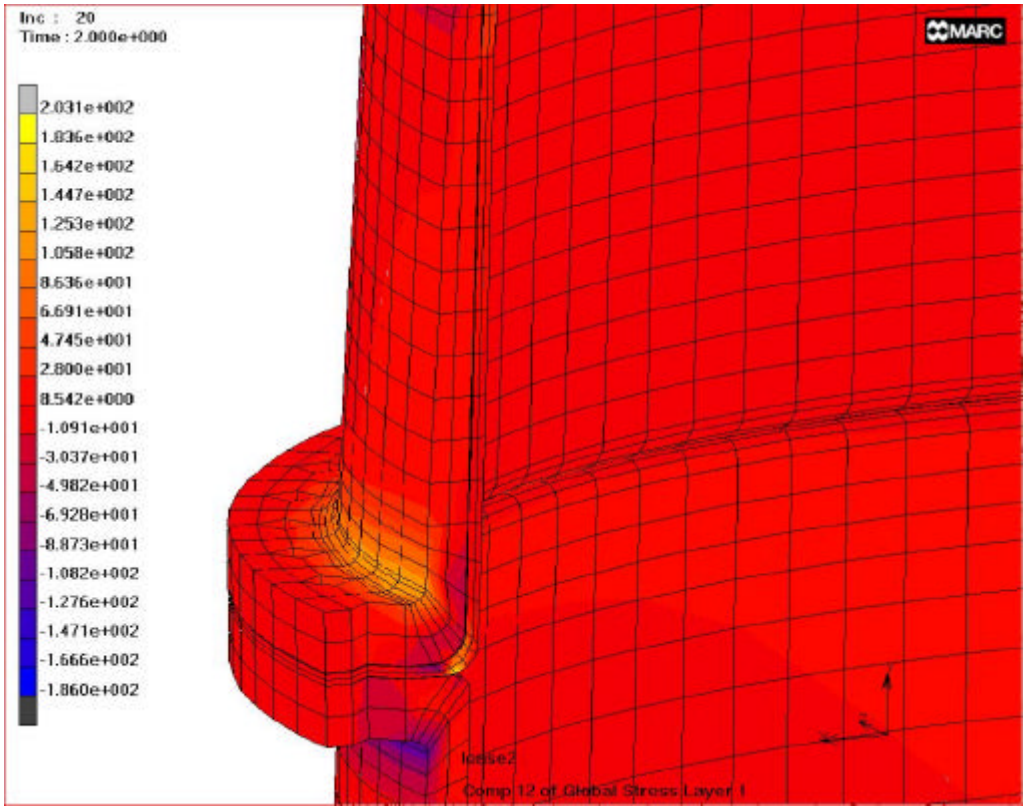




*Fig. 9 Detail of the stress distribution on the blade root*

So, as a summary of this blade root calculation, it is clear the high nonlinearity of the finite element model used. It includes non linear behaviour of the adhesive, progressive failure both in the adhesive as in the composite structure, glued contact between adhesive and flange and adhesive and carbon fiber, contact between flanges, and non linear user subroutine for simulating the bolts in the flanges.

## CONCLUSIONS

Along the present article, the procedure for performing calculations on a wind turbine blade has been presented. Non linear finite element calculations are presented for different parts of the blade. Dynamics and fatigue loading are introduced in the calculations to ensure a useful life specified by international regulations. Extensive use of computational tools (Bladed for

Windows and MARC for the simplified models from which dynamic loading is obtained and MARC for detailed non lineal finite element calculations on the composite parts) has been made to perform the design of the blade.

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