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W. EL MASNAOUI[1,2], A. DAIDIE[1], F. LACHAUD[1], C. PALECZNY[2]
[1]ICA, Université de Toulouse, UPS, INSA, ISAE-SUPAERO, MINES-ALBI, CNRS
3 Rue Caroline Aigle, 31400 Toulouse Cedex 4, France
Email: wafaa.el-masnaoui@insa-toulouse.fr
Email: alain.daidie@insa-toulouse.fr
Email: Frederic.Lachaud@isae-supaero.fr
[2]SAFRAN Aircraft Engines Villaroche, SAFRAN Group
Rond-point René Ravaud, 77550 Moissy Cramayel, France
Email: christian.paleczny@safrangroup.com

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Abstract

This research work deals with development of a simplified beam model for 3D woven-metal bolted joints behavior. An analytical model called “bending beam model” applied to homogenous materials has been identified and investigated. Its adjustability to our application was confirmed through comparisons with Finite Element models on equivalent geometric configurations and several load cases. Then, a strategy is set in order to extend the model to composite joining behavior based on beam theory. The main idea consists on searching an equivalent stiffness able to describe the global composite behavior under bending loading. Thus, bending stiffness of an orthotropic beam under plan stress is considered. Stiffness calculation is necessary in bolted joint design. Hence, A method is proposed to take into account the out-of-plan stiffness of the beam. Finally, the pseudo homogenous behavior of the composite based on equivalent stiffness formulation is implemented in the analytical model and composite beam theory is applied. Results are compared to a 3D Finite Element model.

1 Introduction

Composites materials are increasignely being used in many engineering applications due to their high strength to weight ratio. One of major applications of composite materials is in aerospace structures where the weight reduction is a challenging issue. Aircraft structures are made up of several components which should be assembled. Thousands of fasteners are used to join components together and bolts are a type of mechanical fasteners.

This study is motivated on application of 3D woven composites on aircraft engines casing. The main function of the Fan case is to ensure blade containement in case of a Fan Blade Out called commonly FBO. Assembly of the casing line is made up of bolts, joining it to the inlet. Methods are needed to design this joints accounting complexity of woven composites behavior and bolt interaction with composite flanges. However, time and design cost must be reduced.

Field of bolted joint analytical design tools is large, VDI 2230 [1] recommendation is the most known in bolted joints design. However, it is limited to bolted assemblies subjected to axial or weakly eccentric loading since for high bending the hypothesis of peeling off at the contact interface is no more accurate. Agatonovic [2] developed a model counting the bending stiffness of subassemblies, hence it is adapted to bolted joints subjected to high eccentric loads. One of important elements of bolted joint design is compressive and bending stiffness of assembled parts. Rassmusen [3] proposed an analytical method to calculate the equivalent clamped area of cylindrical joined parts, Alkatan [4] extended the method to prismatic joined parts. VDI 2230 considers a cone deformation zone

geometrically calculated and Rotcher [5] cone based. One can note that all previous studies are developed for homogenous and isotropic materials.
In company with increasing composites use, the investigation of composite bolted joints behavior is unavoidable. Several experimental and numerical studies deal with composite bolted joints, Warren, et al [6] experienced the behavior of a 3D orthogonal woven composite bearing under single and double shear joints. Results show that composite with three dimentionnal fiber reinforcement exhibit exceptional strength when subjected to off-axis loading. Zlatan, et al [7] investigated the behavior of fastened composite-aluminium joint. They conclude that with a non linear finite element model results are in better agreement with experiments, damage and progressive failure must be considered. McCarthy and McCarthy [8] developed a finite element model to study the effect of bolt hole clearance in single bolt single-shear joints. One of the important conclusions was that as the bolt hole clearance became larger, the joint stiffness was reduced. Chen and Lee [9] proposed a 3D contact modelling strategy for composite laminate with bolted joints under bending loading. Many other analysis were performed, but analytical tools for bolted joint design ramain less numerous. Madenci, Barut and Guven [10] performed an analytical method for stress analysis of composite bolted joints under multiaxial loading based on a combined complex potential and variational formulations but no stress analysis are conducted for bolt stress.

In this paper, an analytical model for bolted joint behavior of a 3D woven composite considered as an orthotropic materials in a microscopic scale where both plates and bolt stress are analysed. The model is based on Agatonovic bending beam model and adjusted to orthotropic materials by integrating bending and compressive stiffness of an orthotropic beam under plane stress. In fact, several beam theories have been developed for predicting response of composites beams. Ghugal and Shimp [11] classified beam theories into three main categories: the classical beam theory (CBT) known as Euler beam theory applicable to slender beams. For deeper beams, it underestimates deflection in case of static loads due to ignoring transverse shear effects. This problem is overcome with First Order Beam Theory (FOBT) know as Timoshenko beam theory. However, FOBT suppose that shear strain and stress is constant along beam thickness and a shear correction factor is introduced. The High Order Beam Theory (HOBT) is developed to avoid the use of shear factor and based on higher order variation in-plane and transverse displacement. Although the HOBTs offer an improvement in accuracy compared to FOBT, they are computationally more demanding.

2 Bending beam model for isotropic materials

2.1 General presentation of the model

Developed first by Agatonovic, the analytical description of the model is shown in Figure 1. The assembled plate is modeled by a supported beam subjected to an eccentric axial load F_e and/or moment M_e at a distance *m* from the bolt axis. The two reactions are modelling bolt load F_b and contact force resultant F_c situated at a distance *s* from bolt axis. This parameter *s* – eccentricity of contact force resultant- is governing equations to find finally tensile and bending bolt stress.
The strength of this model appears in considering bolt preload and a highly excentric external load. Therfore, simulations run in many studies shows a weakness of the model in case of high preloading level. This point was the subject of improvement performed by Guillot [12] who introduces a parameter γ which relies the plate stiffness under bolt head to the plate stiffness at the joint interface. It has been prooved that this parameter depends on preload value. In addition, beam model supposes a good-enough support area behind the bolt head. Bakhiet [13] developped a new model based on Agatonovic model accounting any support area behind the bolt head.

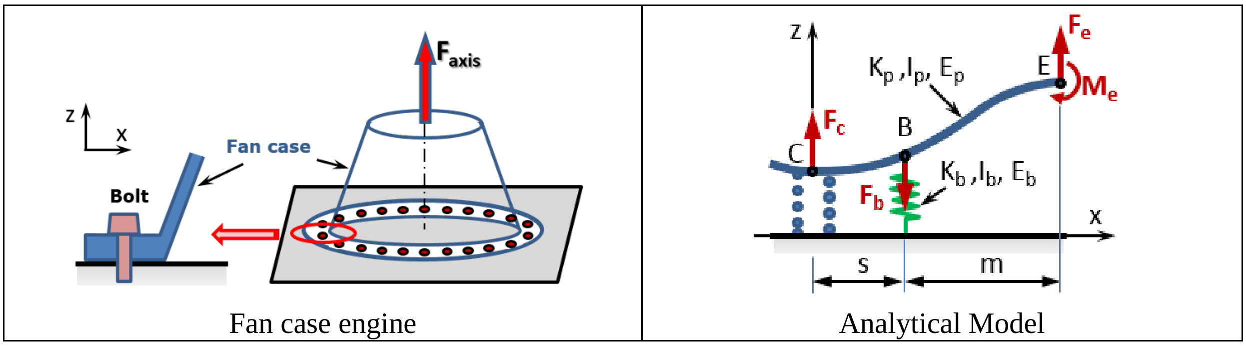

Figure 1. Analytical description of the generalized beam model

2.2 Governing equations

After writing equilibrium equations, displacement compatibility equations and deflection expression at point B according to Euler-Bernoulli beam theory [12], the characteristic equation of bending beam model is given by (Eq.1) as:

$$C_1 s^3 + C_2 s^2 + C_3 s + C_4 = 0 \tag{1}$$

$$C_1 = \frac{mK_pK_b}{6\, E_pI_p(K_p + K_b)}$$

$$C_2 = \frac{\emptyset}{m + 2\, E_pI_p\, (q_1 + \frac{q_2}{m})}$$

$$C_3 = \frac{2\, m + 2\, E_pI_p\, q_1}{m + 2\, E_pI_p\, (q_1 + \frac{q_2}{m})}\emptyset$$

$$C_4 = -m\ \&\ \emptyset = \frac{Q}{F_e} + \gamma\frac{S_p}{S} - 1$$

Where E_p is Young Modulus of the plate, I_p moment of inertia, K_p plate stifness, K_b bolt stifness. q_1 and q_2 are two parameters depending on local stiffness of the plate and depends on joint type (flanged connection, T joints, …), F_0 is the preload.

There are many ways to calculate bolt and plate stiffness. In this work, the two prameters are calculated as detailed in [4]. γ is calculated within the (Eq.2) and it is set to 0 if negative where h_p is the plate height.

$$\gamma = 2{,}2\ (\frac{s}{h_p}) - 0{,}6 \tag{2}$$

In case of prismatic plates bolted joints, as investigated in this paper, C_1 and C_2 are null and resolution of the characteristic equation leads to bolt tensile load and bending moment given by (Eq.3) and (Eq.4) as :

$$F_b = F_e \left(1 + \frac{m}{s}\right) \tag{3}$$

$$M_{fb} = \frac{s^2}{2\,h_p} \frac{E_b I_b}{E_p I_p} F_c \tag{4}$$

3 Bending beam model adjusted to 3D woven composite

3.1 Method

The model described above proved its ability to predict bolted joint behavior of homogenous plates in previous studies that are not detailed in this paper. The purpose of this study is to extend the model to 3D woven-metal composite bolted joint. As this one is based on beam theory, on can apply beam theory of orthotropic materials under plane stress considering a macroscopic behavior of the composite.
The elastic behavior of orthotropic beams under plane stress is given as:

$$\begin{Bmatrix} \sigma_{xx} \\ \sigma_{xz} \end{Bmatrix} = \begin{Bmatrix} \dfrac{E_x}{1 - v_{xz}v_{zx}} & 0 \\ 0 & G_{xz} \end{Bmatrix} \begin{Bmatrix} \varepsilon_{xx} \\ \gamma_{xz} \end{Bmatrix} \tag{5}$$

Where σ_{xx} and ε_{xx} are the tensile stress and strain respectively, σ_{xz} and γ_{xz} are the shear stress and strain respectively, E_x the Young Modulus in x direction, v_{xz} is the Poisson coefficient in (x,z) plan and G_{xz} is the shear modulus in (x,z) plan.

In order to simplify the study, Euler-Bernoulli beam theory is considered where the shear effect is neglected, i.e. :$\gamma_{xz} = 0$ as supposed in Agatonovic bending beam model. Thus, the classical beam theory is applied. This assumption is not properly established due to the span to depth ratio of the considered beam in this paper which is quite important. In addition, concentrated loads seems to be increasing the shear effect in orthotropic beams more than in isotropic ones [15]. Therefore, in a preliminary design context, this type of assumption can be accepted where not a high accuracy is expected.
Thus, shear stress is null and tensile stress is given by (Eq.3) as:

$$\sigma_{xx} = \frac{E_x}{1 - v_{xz}v_{zx}} \varepsilon_{xx} \tag{3}$$

From this relation, one can replace the Young Modulus E_p of isotropic material by Q_{xx} in case of orthotropic materials (Eq.4).

$$Q_{xx} = \frac{E_x}{1 - v_{xz}v_{zx}} \tag{4}$$

The second element of bending beam model extension to 3D woven composite is the softness method calculation. One can note that this parameter is set by considering the out of plan displacement of the plate's clamped part, consequently, we propose to consider the out of plan modulus Ez in stiffness calculation. In this way, the plate stifness is given by (Eq.5):

$$K_p = \frac{E_z\,h_p}{A_{eq}} \tag{5}$$

Where A_{eq} is the equivalent area given in [4] and h_p is the depth plate.

By including those two main modifications to the basic bending beam model, the orthotropic behavior of 3D woven composite is implented in the bolted joint model.

3.2 Application

In order to verify the relevance of suggested modifications for accounting the orthotropic behavior of 3D woven composites in the bending beam model, this one is applied on a bolted joint assembly as shown in Figure 2, which dimensions are given in Table 1. It is about a multimaterial assembly where the lower plate material's is isotropic and the upper one's is an orthotropic material which engineering constants are given in Table 2 [16]. The analytical calculation is run via MATLAB®.

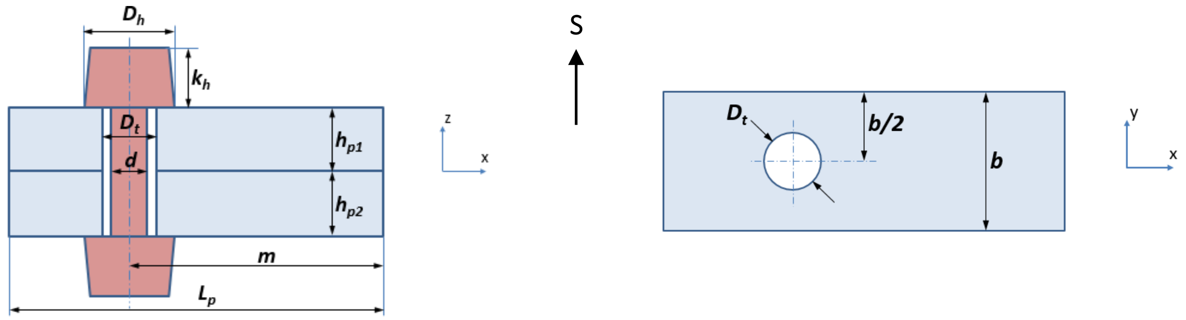

Figure 2: Description of bolted joint specimen

Table 1: Dimensions of bolted joint specimen

	D_h	D_t	k_h	d	m	L_p	hp_i	b
Distance (mm)	9	6.5	11	6	27.5	40	5	20

Table 2: Material properties of assembled parts

	E_x (GPa)	E_y (GPa)	E_z (GPa)	ν_{xy}	ν_{xz}	ν_{yz}	G_{xy} (GPa)	G_{xz} (GPa)	G_{yz} (GPa)	E (GPa)	ν
Plate1	54.25	54.25	12.59	0.309	0.332	0.332	20.72	4.55	4.55		
Plate2										210000	0.33
Bolt										210000	0.33

3.3 Finite Element simulation

A three dimensional (3D) Finite Element (FE) model is conducted using Abaqus® standard solver in order to compare results of analytical model to those given by the FE simulation. The bolt and two plates are modeled separately with C3D8I elements. Such elements are used to better capture bending, since the orthotropic plate is subjected to an eccentric axial loading that will cause a bending stress of both plate and bolt. The simulation is such that the assembly is preloaded at F_0 = 10kN at the first step, then, a concentrated load is applied at the left cross section mid-plane of the plate above, with kinematic coupling with the right face nodes. The isotropic plate2 is fixed and load is increasing with a step of 0.2kN till 5kN. Contact between bolt and the isotropic plate is set to be with friction where friction coefficient is 0.1, while contact between isotropic and orthotropic plates is frictionless, and a hard contact formulation is choosen.

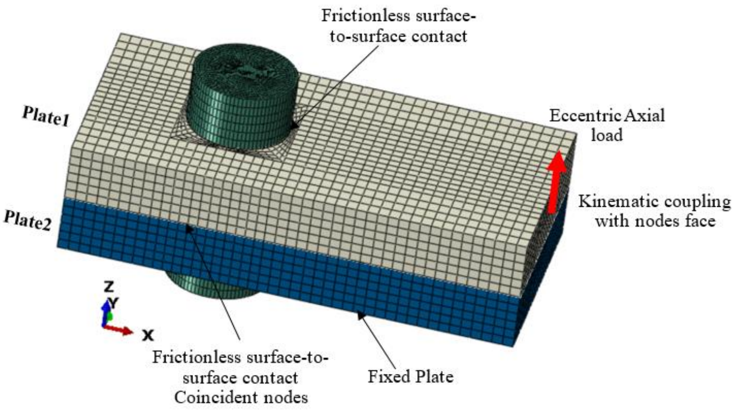

Figure 3: 3D Finite Element Model description

3.4 Results discussion

Analysis of analytical and numerical results leads to compare four main elements. First the excentricity ***s*** of contact force resultant, tensile and bending bolt stress and finally, bending stress of the plate.

Results comparison shows that the analytical model integrating plate's orthotropic behavior under Euler-Bernoulli beam theory shows a reasonable agreement with FE simulation. Figure 4-a shows evolution of bolt tensile stress with increasing load Fe, results are showing a max difference of 6% which means that the present analytical model can well predict bolt tensile in a composite bolted joint considering an orthotropic behavior. However, bending stress results are diverging as shown in Figure 4-d, the analytical model seems to over estimate bolt bending. This observation may be explaned by neglecting the shear effect under head bolt of the orthotropic beam. Bending moment of the beam at bolt axis is also compared between FE and analytical model in Figure 4-b. This one is calculated according to beam theory as well:

$$M_{fp} = \frac{F_b\, m\, s}{m + s} \tag{6}$$

Analytical results of plate bending moment are showing the same difference as for bending bolt stress. Another reason can be given for the differences observed, in fact, stiffness method calculation may not be accurate for composite bolted joint, a new method must be developed in order to consider stiffness in two principal axis of the orthotropic beam.

Values of ***s*** given by analytical model are closed to FE results till 2 kN to 2.6 kN as shown in Figure 4-c. From these loading values, all external loads are supported by the bolt and the peeling off happens, Figure 5 shows contact status with increasing Fe where after Fe=2.6 kN the contact area outstrip the bolt head. After peeling off, the analytical model is known to be less accurate as the plate rotation is not considered which explains the difference observed. One can note that in case of isotropic materials, the max gap on *s* between FE and analytical models and before peeling off is 0.5%, here it presents 2%.

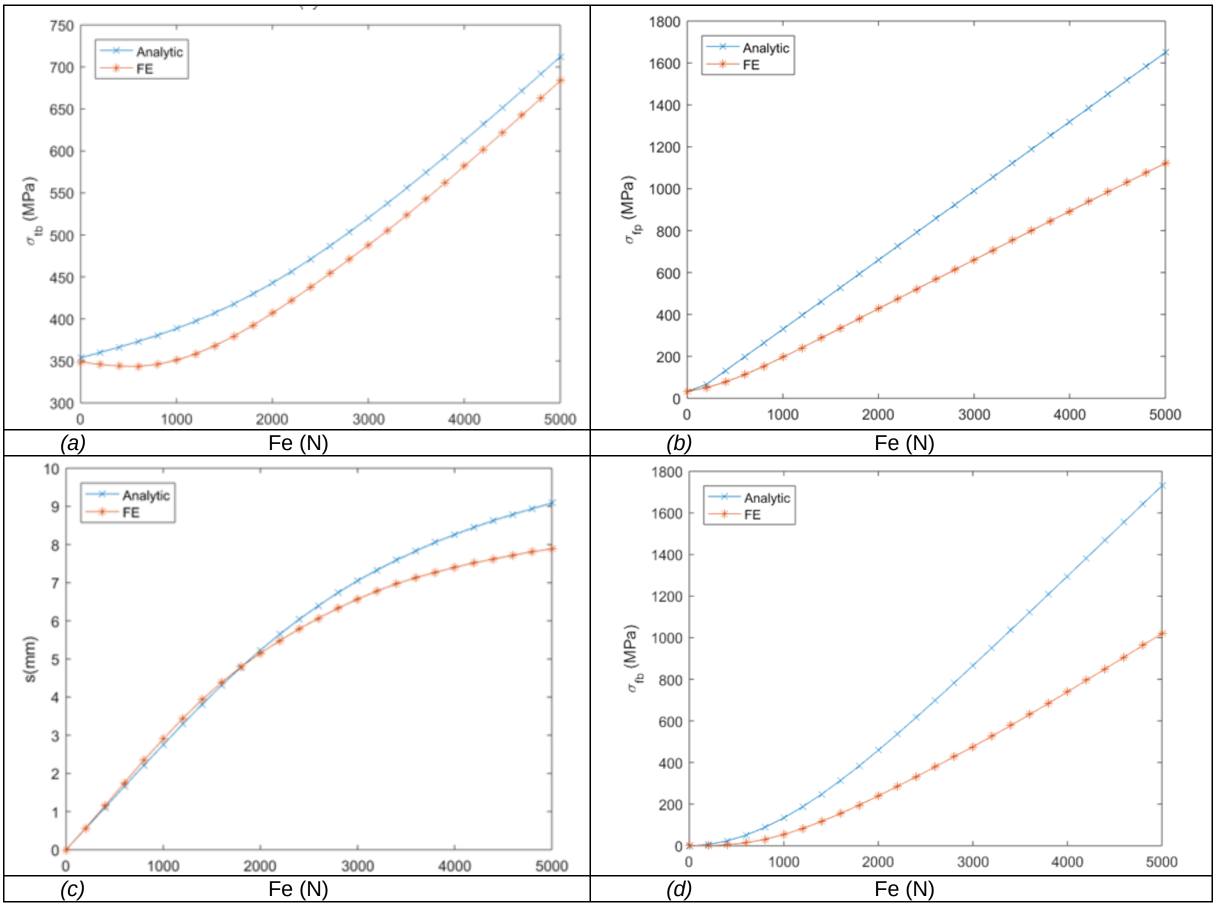

Figure 4: FE vs analytical model for (a) contact force resultant eccentricity s (b) plate bending stress $\boldsymbol{\sigma_{fp}}$ (c) bolt tensile stress $\boldsymbol{\sigma_{tb}}$ and (d) bolt bending stress tensile stress $\boldsymbol{\sigma_{fb}}$

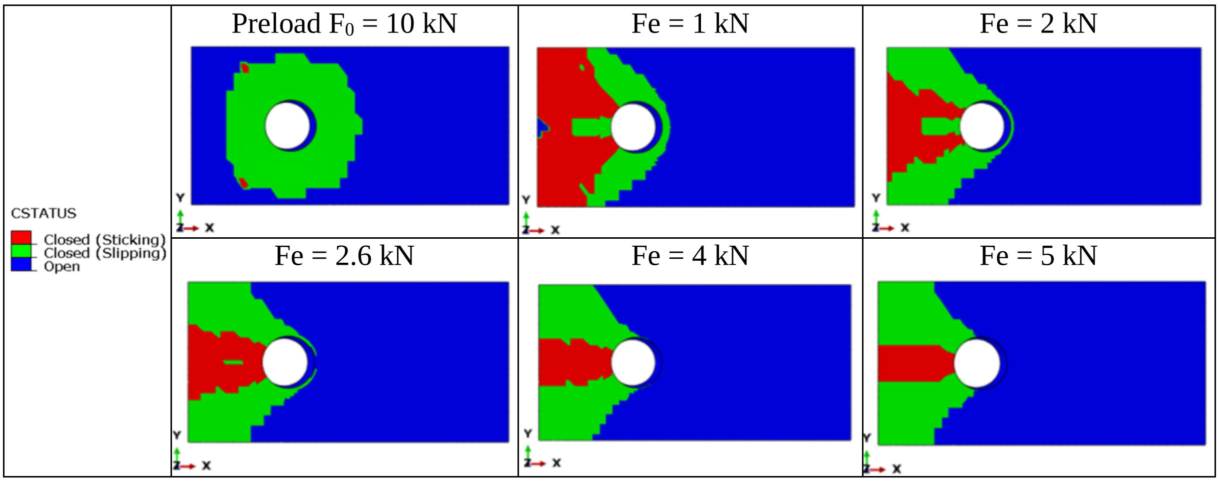

Figure 5: Contact status with increasing of axial load Fe

4 Conclusion

An analytical beam model of isotropic bolted joints assemblies have been adjusted to integrate orthotropic behavior of 3D woven composite in a metal-composite bolted joints. It is an Euler-Bernoulli beam theory based model. Results show that the suggested method to extend the model to orthotropic joints is good enough in a prelemenary design context for tensile bolt stress and contact force resultant position. Bolt and plate bending stress are less accurate. The model must be more improved in order to resolve bending stress results divergence from FE calculation. This can be conducted by considering shear effect and apply FOBT. Developing a new method for through-the-thickness stifness calculation of the assembled parts is also a hopeful way to improve results accuracy.

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