INFLUENCE OF BEARING DEGRADATION ON THE BEHAVIOUR OF MULTI-BOLT COMPOSITE JOINTS WITH HOLE-LOCATION ERROR

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Abstract

In the aircraft industry, the method for designing metal-composite joints is mainly based on conservative metallic calculation rules and on applying high safety factors. The reason is that the influence of errors due to manufacturing is not well-known, and few studies deal with this issue. This study first presents the evolution of load transfer inside a double-lap joint with two bolts, in presence of bolt-hole clearance and position error. Then, a 2-D analytical numerical model is detailed, in order to evaluate the load distribution in multi-bolt shell structures.

1. Introduction

The increasing use of composite materials in the aeronautical field leads to metal-composite joints with a large number of fasteners. The choice of joint dimensions, fastener diameter, number of fasteners, spacing between rows of fasteners, is well understood now, thanks to many studies about their influence [1], [2], [3]. However, the effect of hole-location errors on the assembly mechanical performance is less well controlled. To reduce location error and thus ensure mechanical performance, fastener holes are made in a single drilling operation which requires a complex flow-process grid. This assembly process is incompatible with cost-efficient interchangeability rules and contributes in the increase in joint manufacturing and maintenance costs.

Few studies have been published dealing with the influence of geometrical errors on the mechanical behaviour of joined structures. This can be explained by the fact that this issue needs an accurate stiffness model for each constituent of the assembly (joined parts, fasteners) which controls load distribution between fasteners. Moreover, even for low hole-location errors or clearance, load distribution between fasteners is controlled by non-linear behaviour of materials, which generates a local softening [4], [5]. Another problem, related to experimental validation, is found in introducing a controlled flaw into samples and developing specific instrumentation to analyse the effect of the flaw [4].

The load distribution in bolted joints has been investigated with different levels of complexity. Some studies take the fasteners as rigid bodies and define the material behaviour of structures as elastic linear [6], others integrate nonlinearities due to contact issues (clearance, friction, etc.) and also bolt stiffness [7], [8], [9], [10]. These models do not usually include nonlinear behaviour due to material damage in the fastener or around the hole. Some studies take into account material damage [5], [11], [12], [13], [14], [15], [16], [17], [18]. In [5], [11], [13], [18], a 3-D finite element model is used to study the progressive degradation implied by bearing loading. Thus several damage mechanisms are introduced in this model, such as delamination, matrix cracking and fiber compressive failure. However, such a complex model cannot be applied to a large structure with multi-bolt joining with a view to study the effect of design parameters. In [12], the most critically loaded fastener is first determined using a linear analysis, and then a specific study including damage mechanisms focuses on this location. Nevertheless, the effect of local softening on load redistribution between fasteners is not represented. In [15], [17], a 1-D analytical model is proposed in which each bolt is modelled by a spring with a nonlinear force displacement law in order to represent the local softening due to bearing damage. The evolution of load transfer rate during loading is thus more easily predicted according to joint dimensions and material behaviours. It should be noted that clearance can be included in the spring force displacement law. Gray and McCarthy [16] included nonlinear bolt behaviour previously proposed [15] in a user-defined 10-node super finite element which can be used with shell elements to simulate large-scale structures. Results on 20-bolt joints are presented and compared with experimental data in terms of strain distribution. Although good agreement is found, the effect of nonlinear behaviour is not addressed.

The approach proposed in this work consists in developing two models with different levels of complexity. The first one is a 1-D analytical model which is dedicated to study the influence of joint parameters in preliminary design stages. The second model is a 2-D analytical numerical model which is dedicated to evaluate the mechanical performance of actual multi-bolt joint by taking into account joint and part architectures. In these two models, bolt compliance, nonlinear bolt behaviour (i.e. contact with clearance and damage), and hole-location error are taken in account. While nonlinear bolt behaviour is introduced at the bolt scale in the 1-D analytical model under the form of a force-displacement response, it is explicitly introduced in the 2-D model with local behaviours.

2. Description of the 1-D analytical model

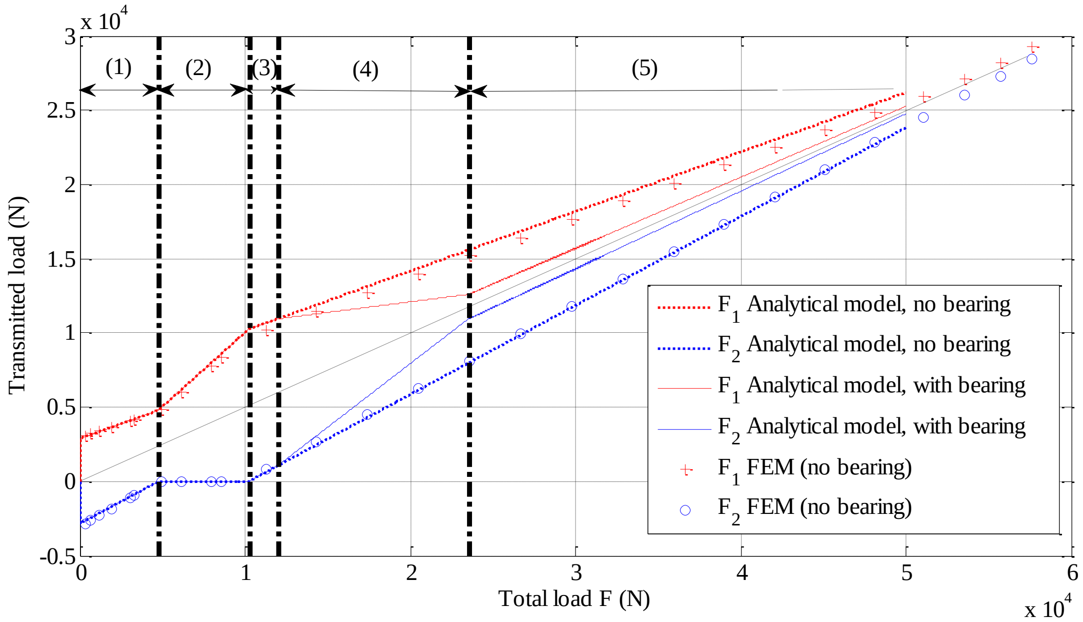

Figure 1. Loads transmitted by the bolts versus applied load for Cgeom = 0.025 mm and δp = 0.150 mm.

The model consists in dividing the n-bolt joint in n+1 sections. Each adherent section and each bolt are considered as a spring. Elastic linear behavior are assumed for adherent section springs while a non-linear behavior is used for bolt spring in order to account clearance, progressive contact establishing and composite damage. Hole-location errors are introduced in term of initial displacement conditions. The model statement and identification are fully described in [19].
The model was validated by a 3D finite element model and experimental tests on double-lap aluminum-composite joints with two bolts. The evolution of the load transmitted by each fastener is plotted in Figure 1 for both the analytical model and the FEM for a clearance of 0.025mm and a location error of 0.150mm. Several stages can be identified along the curves obtained with the analytical model. Preload caused by hole-location error generates two opposite loads on bolts. During the first stage of external load application (1), the second bolt is unloaded while the first bolt is loaded more. When the second bolt is totally unloaded, the clearance recovery stage (2) starts. During this stage, the first bolt transfers the whole applied load. Once the clearance has recovered, the second bolt is in contact with the adherent holes again, and both bolts can be progressively loaded simultaneously (3). Bolt #1 reaches the bearing degradation threshold F_b, first, which implies a decrease in load transfer rate (4). Consequently the load is transferred to the second bolt, until it reaches F_b in its turn (5). Furthermore, from Figure 1 it can be concluded that the analytical model is able to replace a complex finite element model for load transfer estimation in a multi-bolt joint with clearance and location error. The FEM, which contains 760,000 degrees of freedom, needs about 12 hours of calculation time, whereas the analytical model only takes a few seconds.

This analytical model is quite simple and takes a lot of parameters and phenomena into account, but it can only be applied to joints with one row of bolts loaded in the bolts line.

3. Description of the 2-D model

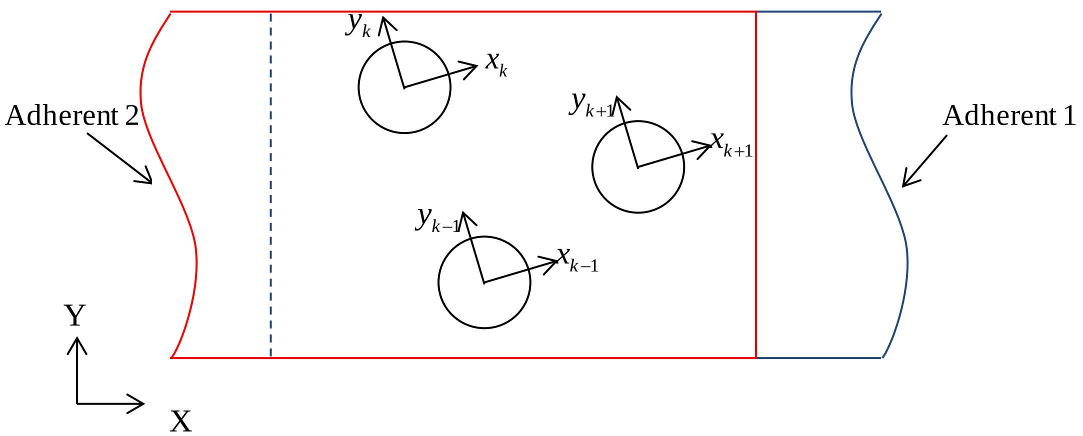

Figure 2. Description of a multi-bolt joint with coordinates systems

3.1. Problem statement

We focus on a multi-bolt composite-composite or metal-composite joint, made of a part 1, with a thickness h_1 and called adherent 1, and a part 2, with a thickness h_2 and called adherent

2 (Figure 2). The global coordinate system is defined as $(\vec{X},\vec{Y})$. The Figure 3 shows the local polar coordinate systems $(\vec{x}_k,\vec{y}_k)$ and $(\vec{u}_k,\vec{v}_k)$, whose origins are located at the center of the fastener pin. $\vec{x}_k$ is defined as the direction of the loads $\vec{F}_k^{\,p}$ transmitted by the adherents *#p* to the fastener. The radius of the fastener *#k* is equal to $R_k = a_k - \lambda_k$, where a_k is the hole radius of adherents *#p* associated to fastener *#k* and λ_k is the radial bolt-hole clearance associated to fastener *#k*. For each bolt *#k*, a_k and λ_k are assumed equal for the two adherents.

Adherents 1 and 2 are modelled by continuum shell elements which allow representing complex geometry with a reasonable calculation time. Adherent holes are explicitly represented with a moderately refined mesh which allows accessing the stress field around the hole. Hole-location error can then be directly integrated on adherent geometry. Bolt and adherent-bolt contact models are based on the Madenci's model proposed in [8].

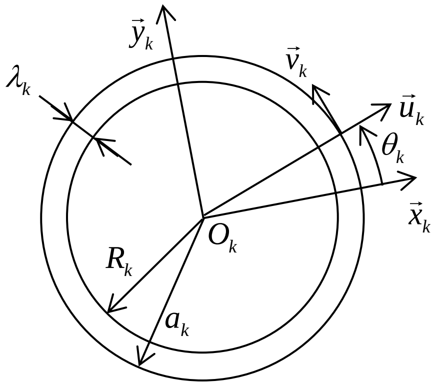

Figure 3. Definition of polar coordinate systems and holes dimensions for the fastener *#k*

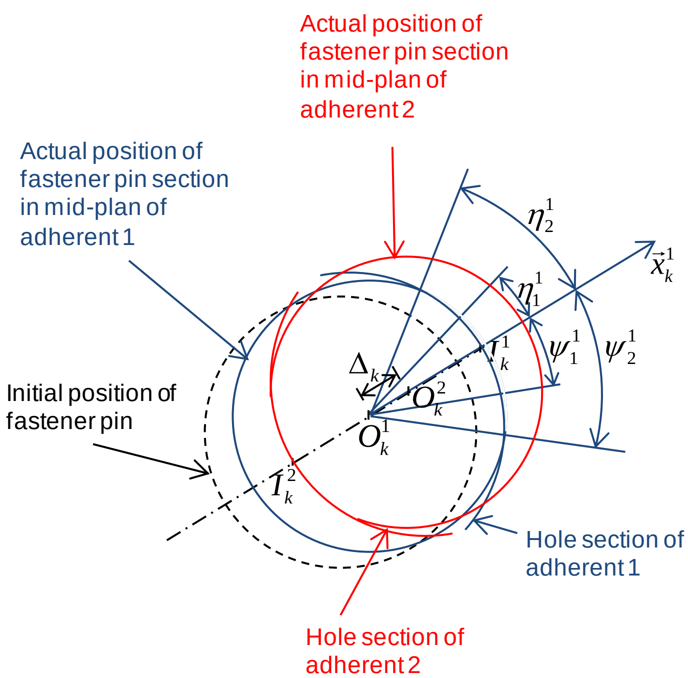

Figure 4. Description of contact between fastener *#k* and hole boundary

As proposed by Xiong and Poon [9] by modelling the fastener as a short beam, the compliance S_k of fastener *#k* can be calculated as follow:

$$S_k = \frac{\Delta_k}{F_k} = \left(\frac{2R_k}{h_1}\right)^2 \frac{\left(h_1 + h_2\right)^3}{12E_k I_k}\left[1 + \frac{12\kappa_k E_k I_k}{G_k A_k \left(h_1 + h_2\right)^2}\right] \quad (1)$$

where $F_k = \overrightarrow{F_k^p} \cdot \overrightarrow{x_k^p}$ is the amplitude of the load transmitted by fastener *#k* and Δ_k is the relative displacement between the two fastener pin sections in mid-plane of each adherent as illustrated in Figure 3. κ_k is a shear coefficient parameter which is equal to 1.33, $E_k I_k$ is the bending stiffness and $G_k A_k$ is the shear stiffness of the fastener pin.

The relative normal displacement ΔU_k applied to adherents between contact center points I_k^1 and I_k^2 defined in Figure 4 are then related to the transmitted load by the following equation:

$$\Delta U_k = u_k^2\left(a_k, \theta_k = 0\right) - u_k^1\left(a_k, \theta_k = 0\right) = 2\lambda_k + S_k F_k \quad (2)$$

where $u_k^p\left(a_k, \theta_k\right)$ is the normal (i.e. on $\vec{u}_k$) displacements of adherent *#p*. The bolt load is transmitted through the contact zones between fastener and adherent holes. In presence of friction between fastener pin and adherent holes, the contact zone in each adherent *#p* for each fastener *#k* is divided in no-slip and slip zones, delimited by angles η_{1k}^p, ψ_{1k}^p, η_{2k}^p and ψ_{2k}^p (Figure 4).

The boundary conditions along the fastener holes can then be defined as:

$$u_k^p\left(a_k, \theta_k\right) = u_k^p\left(a_k, \theta_k = 0\right)\cos\left(\theta_k\right), \text{ with } \theta_k \in \left[-\psi_{2k}^p, \eta_{2k}^p\right] \quad (3)$$

$$v_k^p\left(a_k, \theta_k\right) = u_k^p\left(a_k, \theta_k = 0\right)\sin\left(\theta_k\right), \text{ with } \theta_k \in \left[-\psi_{1k}^p, \eta_{1k}^p\right] \quad (4)$$

$$\begin{aligned} \tau_k^p\left(a_k, \theta_k\right) &= f\left|\sigma_k^p\left(a_k, \theta_k\right)\right|, \text{ with } \theta_k \in \left[-\psi_{2k}^p, -\psi_{1k}^p\right] \\ \tau_k^p\left(a_k, \theta_k\right) &= -f\left|\sigma_k^p\left(a_k, \theta_k\right)\right|, \text{ with } \theta_k \in \left[\eta_{1k}^p, \eta_{2k}^p\right] \end{aligned} \quad (5)$$

$$\sigma_k^p\left(a_k, \theta_k\right) = \tau_k^p\left(a_k, \theta_k\right) = 0, \text{ with } \theta_k \in \left[\eta_{2k}^p, -\psi_{2k}^p\right] \quad (6)$$

where v_k^p is the tangential (i.e. on $\overrightarrow{v_k^p}$) displacement of adherent *#p*, f is the friction coefficient, σ_k^p and τ_k^p are respectively the normal and shear stress in the adherent *#p*. The continuity equations which define the contact angles give:

$$\sigma_k^p\left(a_k, \theta_k\right) < 0, \text{ with } \theta_k \in \left[-\psi_{2k}^p, \eta_{2k}^p\right] \quad (7)$$

$$\sigma_k^p\left(a_k, -\psi_{2k}^p\right) = \sigma_k^p\left(a_k, \eta_{2k}^p\right) = 0 \quad (8)$$

$$\tau_k^p\left(a_k,\eta_{1k}^{p-}\right)=\tau_k^p\left(a_k,\eta_{1k}^{p+}\right) \tag{9}$$

$$\tau_k^p\left(a_k,\psi_{1k}^{p-}\right)=\tau_k^p\left(a_k,\psi_{1k}^{p+}\right) \tag{10}$$

The equilibrium of the fastener #*k* gives:

$$F_k^1+F_k^2=0 \tag{11}$$

The last equation gives the relation between the force transmitted by the adherent #*p* to fastener #*k* and the stress field in adherent #*p*:

$$h_p a_k \int_0^{2\pi}\left[-\sigma_k^p\left(a_k,\theta_k\right)\cos\theta_k+\tau_k^p\left(a_k,\theta_k\right)\sin\theta_k\right]d\theta_k-F_k^p=0 \tag{12}$$

3.2. Solving method

For solving this problem, Madenci [8] uses the boundary collocation technique, made possible by the use of the complex analytic functions introduced by Lekhnitskii [20] to represent the adherent elastic deformations.
In the present study, an iterative procedure is proposed to solve both the analytical bolt and adherent-bolt contact formulation and a finite element model (FEM) is used for the adherents. At each iteration, the stress field obtained from the FEM is used to calculate the load transmitted by each fastener (Equation (12)) and to readjust the contact angles according to Equations (7) to (11). Equations (1) to (6) allow then to update the boundary conditions on each adherent hole. Without hole-location error, bolt and adherent holes being initially coaxial, the clearance disables the contact. Contact angles are thus initially taken equal to zero. Hole-location error can otherwise be introduced progressively in a first step by decreasing a virtually augmented clearance up to the actual value. This iterative procedure can be implemented in an implicit or explicit scheme. The user-subroutine facilities of Abaqus software were exploited to combine the analytical equations with the FEM.
First simulations were dedicated to double-lap aluminum-composite joints with two bolts in order to compare the 2-D model to 3-D FEM and 1-D model. Comparison mainly aims at evaluating the relative performances in term of calculation time since the phenomena implied in the three different models are identical. First results are encouraging even if the calculation time proves to be quite dependent to the iterative procedure algorithm.

4. Conclusions

Two models with different levels of complexity were developed in order to study the effect of hole-location error on the load distribution in multi-bolt joints. The work puts emphasis on non-linear behaviour generated by friction phenomena and clearance in bolt-adherent contact and material damage.
An analytical model was devoted to double-lap joint with several bolts lined up in a row. This time-efficient model allows parameters influence studies, but cannot be applied to complex structures. The 2-D model elaborated here is balanced between a numerical part, to represent the geometry of assembled parts including hole-location error, and an analytical part, for contact issues, bolt-hole clearance and bolt pin deformation. This model makes possible to obtain the load in each bolt and plan stress field of composite parts while avoiding the

meshing of fasteners. The compromise between calculation time and result accuracy appears very interesting.

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