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A reduced fastener model using Multi-Connected Rigid Surfaces for the prediction of both local stress field and load distribution between fasteners

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ABSTRACT

This paper describes the development of a reduced model of a fastener using Multi-Connected Rigid Surfaces (MCRS). The stiffness of the connectors is determined, based on a physical approach, considering different deformation modes of the bolt. The reduced model is constructed and identified from a numerical simulation of a single lap reference joint under tensile load, with the adherent parts and bolts represented by 3-D solid elements. A single simulation with a given clearance, axial preload and friction coefficient is used to identify equivalent stiffnesses. The reduced model is then compared with the 3-D solid elements model in a two-fastener configuration for different values of clearance, preload and friction coefficient. The comparison covers overall response in terms of stiffness and load distribution between fasteners, local response in terms of stress fields and calculation times. Results show that the reduced model proposed here is able to reduce calculation times while still providing a good estimate of the mechanical quantities needed for the study and dimensioning of multi-fastener joints.

1. Introduction

Because of the costs and delays incurred in experimental studies, numerical simulation is an essential tool when designing assembled structures. The design of fastened joints is based on making a good estimate of the distribution of loads between fasteners. This load distribution depends on the stiffness of the fastener, the stiffness of the parts, the way in which external forces are introduced, and also clearances and geometrical defects. Note that by stiffness of the fastener, we refer to the ratio of the load borne by the fastener to the relative displacement, one against the other, of the parts adjacent to the fastener. Depending on whether the load is transmitted by adherence or by fastener pin-hole contact, this stiffness will be very different. In the former case (load transmitted by adherence), the role of the axial preload is fundamental; in the latter case (load transmitted by contact), the presence of clearance results in contact occur gradually and hence stiffness evolves. For a composite structure joint, the two modes of load transfer combine since the material is unable to bear a high preload due to its low resistance in the out-of-plane direction. In addition, the materials in contact tend to be damaged as a result of bearing stresses, which also leads to a change in stiffness. Generally, with a single lap joint, the rotation of the parts caused by the secondary bending moment is directed by the bending stiffness of the fastener heads. This is an important phenomenon since it gives rise to a non-uniform distribution of contact pressure between the pin and the two holes.

The quantity and complexity of the phenomena involved lead us naturally to model the fastened joints using a refined mesh of 3-D solid elements. This type of model was developed in order to study the effect of the friction coefficient [1], clearance [2], location error [3] or preload [4]. As well as predicting load distribution between the fasteners, these models also provide accurate results for the stress fields in the high gradient regions around the hole. They are also known for their ability to take material nonlinearities into account, such as damage [2,5,6]. Several modes of degradation can be introduced, such as delamination, matrix cracking or fibre failure. Despite these performances, models using 3-D solid elements are still very costly in terms of calculation time and they require specifically adapted calculation strategies, notably parallel computing by domain decomposition to process assemblies with several dozen fasteners or to carry out parametric studies on several hundred configurations [7,8].

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Thus in order to deal with assemblies with a very large number of fasteners, or to carry out extensive parametric studies, analytical or semi-analytical models have been developed. The 1-D models proposed in the literature can predict load distribution between fasteners in a wide variety of configurations: bolted [3,9–12] or bolted/bonded [13], with clearance and location errors [3], taking into account loss of stiffness due to bearing damage [3,9,13,14]. However, these models are limited to one-dimensional plane geometries and loadings.

Several approaches have been developed in recent years to simplify the way fasteners are represented in finite element models. The first consists of representing each fastener by a special connection element, available in most finite element analysis software packages, which links a single node from each adherent part, potentially incorporating various non-linear behaviours [15–17]. In this case, the holes and the contacts are not represented. This is a very interesting approach in terms of calculation time, but it requires a specific identification procedure to include the role of clearance, friction and material non-linearities overall in the behaviour of the connector. Moreover, the stress field around the fasteners is not represented, and thus macroscopic failure criteria must be used for the assembled parts [18,19].

Gray et al. [20] have modelled the adherent part with shell elements and the bolts with two rigid cylindrical surfaces connected by a beam exhibiting elastic behaviour. The heads were replaced by coupling the movement of the beam ends with the adherent parts. The model is validated for a single lap joint with one bolt and three bolts, including clearance and preload. The problem with these approaches remains the identification of the equivalent stiffnesses associated with the connection elements. A simple estimate of shear stiffness can be made using various analytical models to be found in the literature [21,22], but the values obtained can vary by a factor of 1 to 5.

In this paper, a reduced model of a fastener using Multi-Connected Rigid Surfaces (MCRS) is proposed. The number of rigid surfaces and the stiffnesses introduced between them were selected by analysing deformation modes of the fastener in a single-lap configuration obtained from a reference model made up of 3-D solid elements. From this decomposition, 4 rigid surfaces and 4 stiffnesses were defined. The procedure for identifying these equivalent stiffnesses is also based on the 3-D reference model in a fixed configuration (clearance, friction coefficient, preload). The efficiency of the model was assessed on the basis of global criteria (at the scale of the joint) and local criteria (at the scale of the materials). The validity domain of the identification was studied by varying the clearance, friction coefficient and preload.

2. Analysis of deformation modes of the fastener

In this section, an analysis of deformation modes of the fastener is proposed and in particular the relative movement between the functional surfaces of the fastener. Studies referred to in the bibliography [1,3,21,23] show that deformations in the bolt can be decomposed into 4 main modes: tension and bending along the fastener pin axis, shearing of the pin and bearing–compression of the contact semi-cylinders. A finite element model where the fastener is represented by 3-D solid elements is proposed to extract quantitatively these different deformation modes. This analysis is used as a basis for building and identifying the reduced model.

2.1. Description of the reference joint

The reference configuration used to study deformation modes in the fastener is a single lap joint made of aluminium/composite material with two bolts, as shown in Fig. 1. The composite material is a carbon fibre laminate with unidirectional ply and a thermoset matrix T700GC/M21. The stacking sequence for the composite adherent is $[90/0/-45/+45/0]_s$ and ply-thickness is 0.25 mm. The second adherent part assembled is made of aluminium alloy 2024. The two steel bolts, diameter 6.35 mm, are fixed with a radial clearance of 10 μm and axial preload of 3500 N generating an average contact pressure under the head of about 75 MPa. All material properties are listed in Table 1.

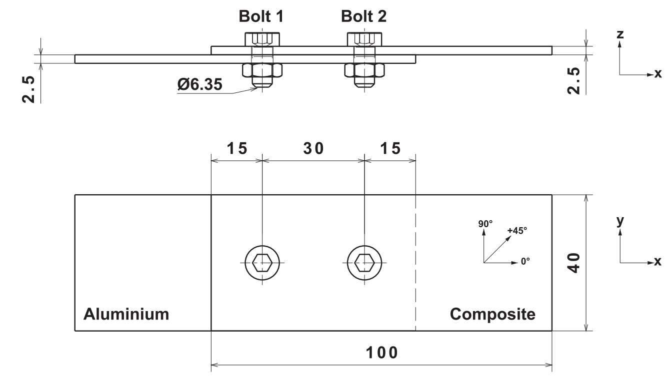

Fig. 1. Geometry of the joint.

2.2. Analysis method of deformation modes of bolt

The model created with Abaqus code uses 3-D solid elements with reduced integration (C3D8R). The threaded junction between the nut and the screw is not modelled. The bolt is therefore represented as a single part. The flexibility resulting from the threaded junction between the nut and bolt is therefore neglected in the reference model and hence in identifying stiffnesses in the MCRS model. According to the literature, this flexibility can be estimated analytically and thus incorporated afterwards into the MCRS model [23,24]. Preload is modelled by initial penetration of 15 μm of each head into the adherent parts giving an intended axial preload of 3500 N. A contact algorithm using the penalty method with a friction coefficient of 0.1 is also introduced to model contact between the bolts and the adherent parts. Deformation modes in the bolt are analysed with a tensile load of about 15 kN, which corresponds to the initiation of bearing damage obtained experimentally.

A section S of a bolt is defined as a set of nodes having initially the same coordinate Z. To represent the kinematics of the bolt, for each section S, initially written S_0 with centre M_0, we define a least-squares plane P from actual position of the nodes of section S. In the event of uniaxial loading along $\vec{X}$, plane P remains perpendicular to plane (XZ). An orthonormal coordinate system $(M, \vec{t}, \vec{y}, \vec{n})$ can then be assigned to each deformed section, as shown in Fig. 2. M is defined as the intersection of plane P and the curve representing the deflected shape of the bolt axis. For the centre section O (O being the intersection of the axis of the bolt in its deformed state and the overlap plane of the joint) the associated coordinate system is written $(O, \vec{x}, \vec{y}, \vec{z})$.

In order to obtain a displacement field that can be exploited to analyse the different deformation modes in the bolt, a rigid body field displacement was subtracted from the calculated displacement field so that coordinate system $(O, \vec{x}, \vec{y}, \vec{z})$ and $(O_0, \vec{X}, \vec{Y}, \vec{Z})$ coincide. A column matrix of small displacements $[D]$ is then defined to represent the displacements of translations u_x and u_z, and also rotation $\theta_y = (\vec{x}, \vec{t})$ for each section, as shown in Fig. 2:

$$[D] = \begin{bmatrix} u_x \\ u_z \\ \theta_y \end{bmatrix}. \tag{1}$$