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FINITE ELEMENT ANALYSIS OF THE EFFECTS OF CLEARANCE ON SINGLE-SHEAR, COMPOSITE BOLTED JOINTS

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ABSTRACT

A three-dimensional finite element analysis on the effects of bolt-hole clearance in composite bolted joints is presented. Both single-bolt and multi-bolt single-shear joints were modelled and the results were compared with those from a parallel experimental programme. The specimens studied were made from graphite/epoxy HTA/6376, with quasi-isotropic lay-ups. Protruding head bolts of 8 mm diameter and torqued to finger-tight conditions were used. The models showed excellent capability to quantify the effects of increasing clearance, which included reduced contact area and overall stiffness in the single-bolt case, and substantially changed load distribution in the multi-bolt case.

Keywords: Bolted Joints, Composites, Clearance, Load Distribution, Finite Element Analysis

1. INTRODUCTION

Mechanical fastening remains a critical aspect of designing aircraft structures from composite materials. Current design methods are largely empirical in nature. To use them in the primary structure of commercial aircraft, involving new materials and more heavily loaded configurations, will require extensive testing involving thick (and therefore expensive) specimens. Even then, designs may not be optimal, which may erode the potential benefits to be gained by using composites. Even though composite bolted joints have been the subject of academic study for thirty years, there is still a need for improved understanding of composite bolted joint behaviour, and analytical methods need to be improved and validated, so they can find increasing use within the industry. The EU research project “BOJCAS – Bolted Joints in Composite Aircraft Structures” [1], aims to produce improved analysis techniques of two types: global analysis methods for fast preliminary analysis and detailed analysis methods for detailed design of critical joints. Detailed analysis methods will involve three-dimensional finite element analysis, and the University of Limerick is involved in this aspect of the work.

The aims of this work are to produce validated models and evaluate their capability to predict behaviour in situations where three-dimensional effects are important. It is also desired to determine procedures required to produce efficient models, which can be incorporated into a semi-automated model-creation tool which is being developed in parallel [2]. As a suitable case study, it was decided to perform a combined experimental/analytical study on the effects of clearance in single-shear joints. The loose-fit clearances and single-shear configuration in the study can be expected to result in variable three-dimensional stress distributions due to bolt tipping, which the models should be capable of capturing.

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Previous studies on the effects of clearance were two-dimensional. Crews and Naik [3] performed a two-dimensional finite element study of specimens loaded in double shear. Hyer, Klang and Cooper [4] performed an analytical, complex variable study, which included consideration of variable clearance, again on double-shear specimens. Di Nicola and Fantle [5] performed an experimental and (two-dimensional) finite element study on pin-loaded specimens in double-shear. Lanza Di Scalea, Cappello and Cloud [6] performed a detailed two-dimensional finite element study of pin-loaded joints with much larger clearances than in any of the above papers. Finally, Fan and Qiu [7] performed a purely analytical study of the effects of clearance in *multi*-bolt joints.

A drawback with two-dimensional analytical or finite element methods is that effects such as clamp-up, secondary bending and varying through-thickness stresses cannot be accounted for. Di Nicola and Fantle [5] concluded that two-dimensional finite element analysis was inadequate to represent bearing strength because it could not account for delamination buckling which they found to govern failure of neat-fit fastener holes.

Recently, three-dimensional finite element models of bolted joints have appeared in the literature. Camanho and Matthews [8, 9], using a pin-lap configuration with a rigid pin model, studied the progression of damage and delamination onset around the bolt-hole interface. Lin and Jen [10] investigated bolted joints with a bonded interface. Ireman [11] modelled a single-shear, single-bolt joint using solid elements to investigate through-thickness stress distributions in composite to metal joints. Thus far, no attempt has been made to quantify the three-dimensional effects that bolt-hole clearance may have on the characteristics of single-shear composite joints. This paper thus presents finite element results on this topic with comparison with results from experiments.

2. MODEL DEVELOPMENT

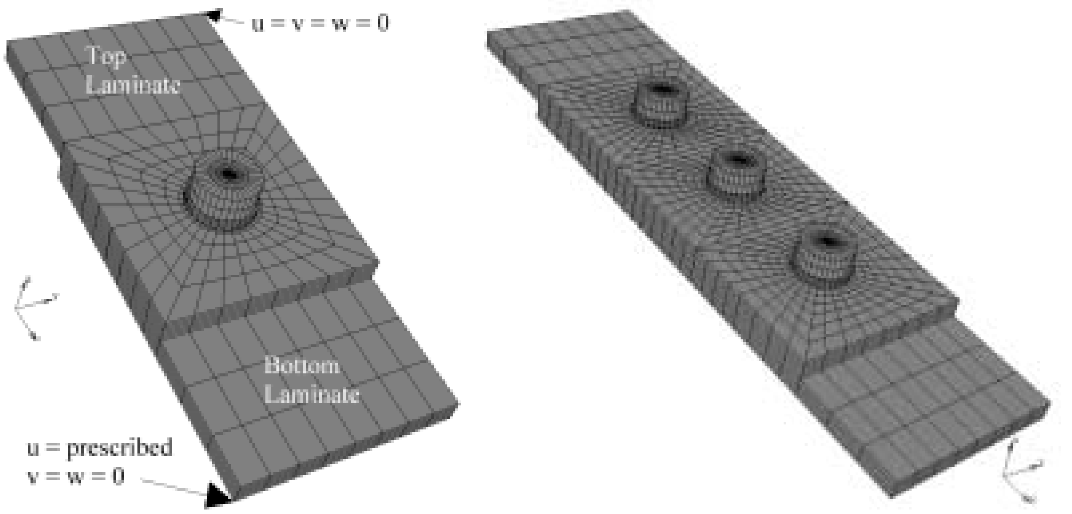
The dimensions of the single and multi-bolt joints modelled here are identical to those of the joints experimentally tested in [12] and [13]. Four different clearances were tested in the single-bolt case (see Table 1), and clearances were varied in each hole in the multi-bolt joints according to a comprehensive test matrix [13]. Clearance was obtained by reaming the laminates with different size reamers, and ranged from neat-fit (coded "C1"), to slightly larger than those found in aerospace primary structures [5] (coded "C4"). Clearances in the experiments could not of course be exact due to manufacturing tolerances so nominal clearances were used in the FE models, as shown in the last column of Table 1.

Typical finite element models for the single-bolt and multi-bolt joints are shown in Fig. 1. The meshing of the laminates is similar to that used by Ireman [11] with relatively high radial mesh density near the hole and under the washer, where high strain gradients exist. Differently from Ireman, the washer was modelled separately. Eight-noded isoparametric hexahedral elements with a full integration scheme were used in the analysis. Wedge elements were used in the centre of the bolts.

Table 1 Clearances used in experiments and FE models

Clearance Code	Reamer Min (mm)	Reamer Max (mm)	Bolt Min (mm)	Bolt Max (mm)	Min Clearance (µm)	Max Clearance (µm)	Clearance used in FE Model
C1	7.985	7.994	7.972	7.987	-2	22	10
C2	8.065	8.074	"	"	78	102	80
C3	8.14	8.149	"	"	153	177	160
C4	8.225	8.234	"	"	238	262	240

The boundary conditions for both models were identical. The left end of the top laminate was given fixed displacement boundary conditions as shown in Fig. 1a. Load was introduced by applying a prescribed displacement in the *x*-direction to the rightmost end of the bottom laminate to mimic a quasi-static loading in the experiments. In order to avoid potential rigid body modes, light springs were applied to the components that were not fully constrained, such as the bottom laminate, bolt(s) and washers. To simulate bolt axial pre-stress due to applied bolt torque, orthotropic thermal expansion coefficients (allowing thermal expansion/contraction only in the direction of the longitudinal axis of the bolt) were applied to the washers on one side of the joint. They were then subjected to a positive temperature differential prior to mechanical loading which had the effect of stretching the bolt and clamping the laminates, which is essentially what happens experimentally. In all the tests in the present study, a torque value of 0.5 Nm was applied to the bolts, representing the lowest repeatable value that could be obtained experimentally, i,e, so-called “finger-tight” conditions. The bolt axial pre-stress at this torque level was determined to be 7.2 MPa using instrumented bolts, as described in [13].



(a) Single-Bolt FE model **(b) Multi-Bolt FE model**

Fig. 1 FE models of single-bolt and multi-bolt joints with boundary conditions

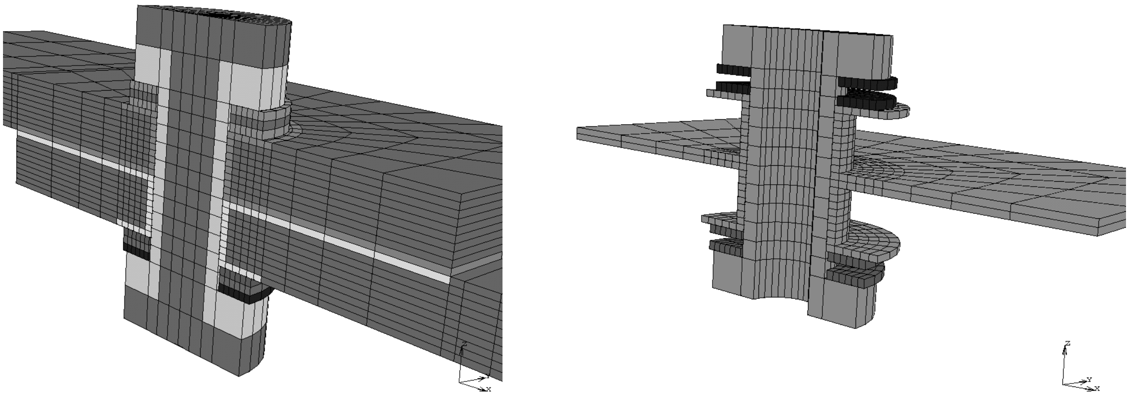
The laminates were made from HTA/6376 (high tensile strength carbon fibre, toughened epoxy) and the material properties are listed in Table 2. The stacking sequence was quasi-isotropic, $[45/0/-45/90]_{\text{Rë}}$, with 40 plies in each laminate, giving a total laminate thickness of 5.2 mm. The bolt was titanium alloy 6Al-4V, the nut was steel, and steel washers were used on nut and head side. The laminates were modelled with ten elements in the thickness direction allowing each element to represent four layers of the composite material, using the layered continuum composite element available in MSC.Marc (Element 149). To allow easier understanding of the model results, and hence clearer determination as to whether contact was working correctly, "equivalent" homogeneous, orthotropic material properties for the laminates were also developed. These were generated by performing a series of tensile and shear numerical experiments on a block of layered material and the in-plane properties were validated against classic laminate theory.

Contact in MSC.Marc requires the definition of "contact bodies", i.e. bodies which potentially may come in contact with each other. Contact bodies can be the physical bodies themselves (e.g. the two laminates, the bolt(s), and the washers). However it is more efficient to select subsets of the physical bodies which are likely to be involved in contact (see Fig. 2) since less checking for contact is required at each solution step.

Table 2: Lamina/Equivalent laminate properties for quasi-isotropic lay-up
*** - verified by laminate theory**

	E_{NN} (GPa)	E_{OO} (GPa)	E_{PP} (GPa)	G_{NO} (GPa)	G_{NP} (GPa)	G_{OP} (GPa)	μ_{NO}	μ_{NP}	μ_{OP}
Lamina Properties	140	10	10	5.2	5.2	3.9	0.3	0.3	0.5
"Equivalent" Properties for QI Lay-up	54.25*	54.25*	12.59	20.72*	4.55	4.55	0.309*	0.332	0.332

Efficiency can also be improved by the use of "contact tables". Contact tables define which contact bodies are likely to contact each other. For example, it is known *a-priori* that the two washers will never come into contact, so the contact table can be set up to eliminate checking for this possibility. Fig. 3 shows the contact table for the single bolt joint case. It can be seen that the contact table is quite sparse indicating an efficient contact definition. The leading diagonal has no entries which eliminates any checking for self contact, and the lower left area of the contact table is deactivated by selecting a single-sided contact definition which ensures that the contact algorithm only checks if nodes from one body (defined first) contact segments on the other body (defined second). This has been found to work much better than so-called "double-sided" contact, in which the nodes from both bodies are mutually checked against the segments of both bodies.



(a) Section through single-bolt model highlighting contact bodies

(b) Contact bodies isolated

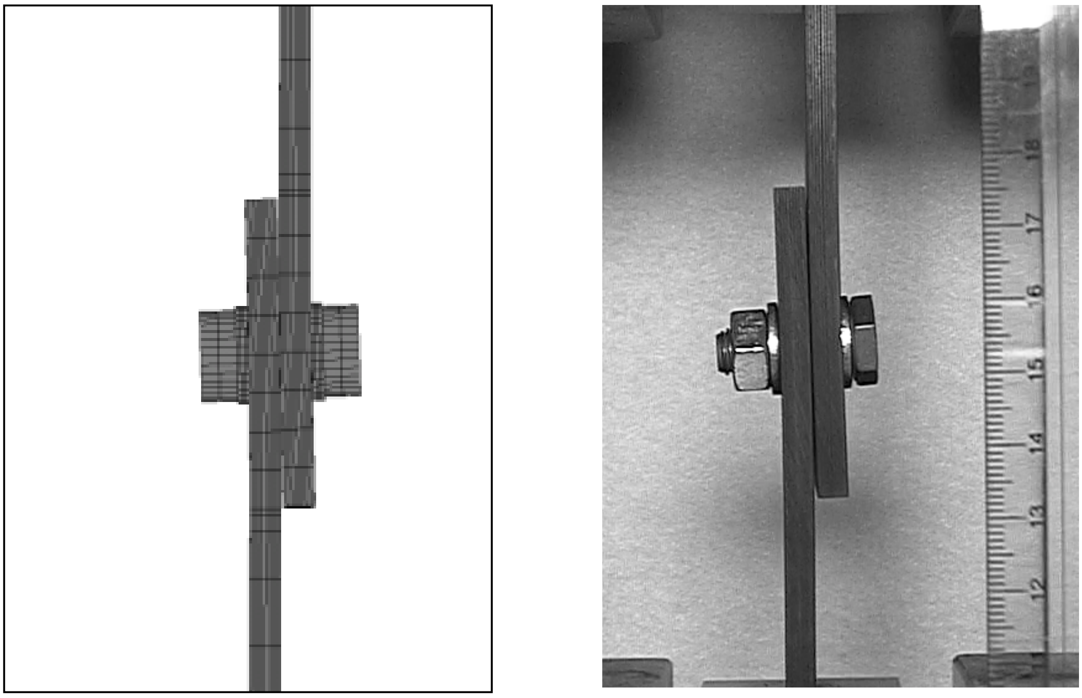
Fig. 2 Contact bodies defined by possible contacting elements only

	BODY NAME	BODY TYPE	1	2	3	4	5	6	7
1	Top_washer_C_lap	deformable					T		
2	Top_washer_c_bolt	deformable							G
3	Bottom_washer_C_l	deformable						T	
4	Bottom_washer_c_b	deformable							G
5	lap1	deformable						T	T
6	lap2	deformable							T
7	bolt	deformable							

Fig. 3 Sample contact table defined in MSC.Marc/Mentat for the bolted joint model
‘T’ indicates touching contact between two bodies
‘G’ indicates glued contact between two bodies

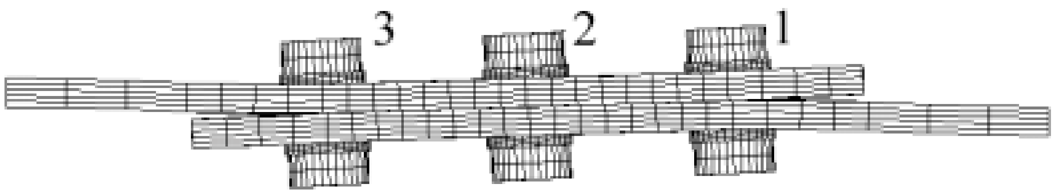
3. RESULTS AND DISCUSSION

The deformed shape of the single-bolt joint model is shown together with the actual deformation in Fig. 4. It can be seen that the finite element model shows similar deformation characteristics to the experiment, including bolt tilting, and secondary bending of the laminates, which is caused by an eccentric load path through the joint.

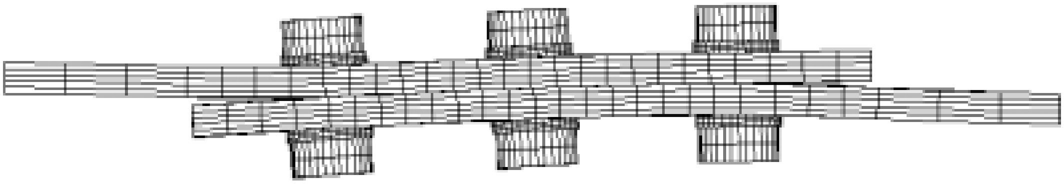


(a) FE Prediction **(b) Experiment**

Fig. 4 Deformation characteristics of a C1 (neat-fit) clearance single-bolt joint



(a) C1_C1_C1 deformed shape, showing anti-symmetric deformation



(b) C3_C1_C1 deformed shape, showing that anti-symmetry is lost

Fig. 5 Deformed shape of multi-bolt models

Fig. 5 shows the deformed shape of two multi-bolt joints, one with all three bolts having a neat-fit entitled C1_C1_C1 and the other with a large clearance (180 microns) in Hole 1 entitled C3_C1_C1. Both deformed shapes are plotted at a magnification factor of two, and the bolts are numbered from right to left as in the figure. Model C1_C1_C1 shows an anti-symmetric deformation, but anti-symmetry is lost in the C3_C1_C1 model. In the C1_C1_C1 model, all three bolts are tilted by approximately the same degree of rotation, but in the C3_C1_C1 joint, Bolt 1 does not rotate as much as Bolts 2 and 3. This is because Bolt 1 is not contacted and hence, not rotated by the laminate, until the clearance at that hole is taken up. This has a significant effect on the load distribution as will be shown later.

Fig. 6 shows the load-deflection curve for C1 and C4 clearance single-bolt joints. The C4 curve shows a delay in load take-up approximately equal to the clearance, which concurs with the experiments in [12]. Attempts at best-fit straight lines are also shown. From these it can be seen that both curves show some initial non-linearity, but after this, the C1 curve is essentially linear. In contrast, the C4 curve shows a slight tendency to stiffen with increasing load. Finally, comparing the two best-fit lines, it can be seen that increasing clearance results in a drop in stiffness of the joint. These findings are all consistent with those reported for the experiments in [12].

The explanation for these variations in stiffness lies in the development of the contact area between the bolt and the laminate. Fig. 7 shows the growth of the contact area between the bolt and the bottom laminate in the lower clearance joint. It can be seen that the contact area gets up to its final value quite quickly, with a contact angle of $160° - 170°$ which is fairly constant through the thickness. In the experiment, the bolt was found to leave a silver-coloured imprint on the inside of the hole as shown in Fig. 7 (highlighted for clarity). This is a clear indication of the maximum contact area that developed during the experiment and can be seen to closely match that predicted by the finite element analysis.

In contrast, in the C4 joint, shown in Fig. 8, significant contact is not made until clearance is taken up, and initial contact is over a very small contact arc. As the load increases, the bolt tilts, and the contact area grows quite gradually. Even at high loads, the contact area is still much less than in the C1 joint. Note that the contact area in the model again agrees well with the imprint left by the bolt in the experiment. The gradual nature of the increase in contact area explains the continuing stiffening of the C4 joint with increasing load, while the lower final contact area explains the lower stiffness of the C4 joint compared to the C1 joint.

To compare models and experimental results for different clearances, a measure of joint stiffness was taken between 2kN and 7kN, for which the experimental load-deflection curves were essentially linear. Above this value, damage began and the curves became non-linear. Table 3 shows the change in stiffness as the clearance is varied from neat fit (C1) to 240 microns (C4), for both the experiments and the finite element models. It is evident that as clearance increases, joint stiffness decreases, and it is also evident that the finite element models provide an accurate prediction of this loss in stiffness.

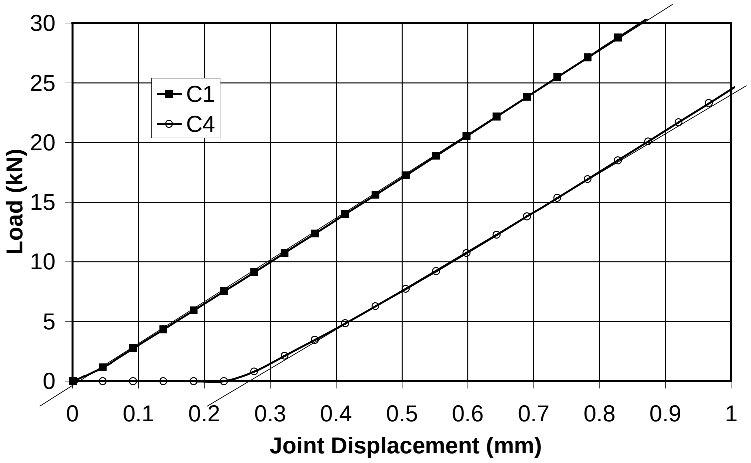
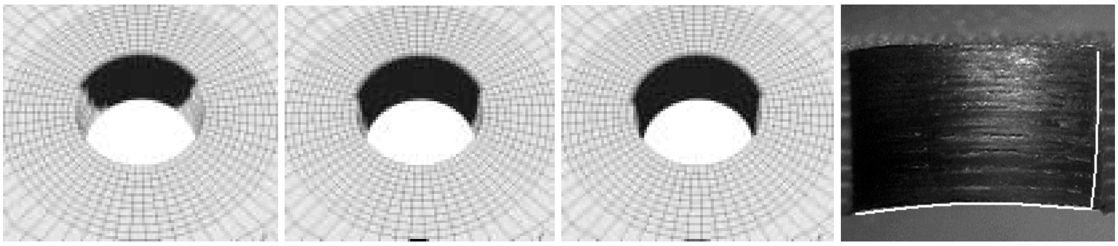
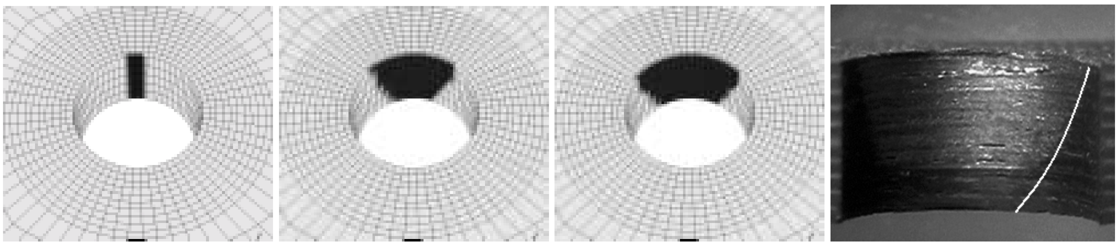


Fig. 6 FE predictions of load-deflection curve for C1 (neat-fit) and C4 (240 micron) clearance single-bolt joints



(a) Initial Contact (b) Intermediate (c) Final Contact (d) Experiment

Fig. 7 Development of contact area in C1 (neat-fit) clearance single-bolt joint

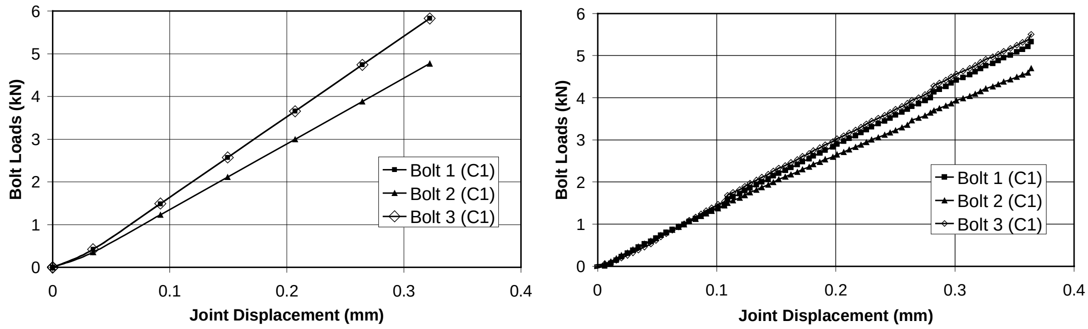


(a) Initial Contact (b) Intermediate (c) Final Contact (d) Experiment

Fig. 8 Development of contact area in C4 (240 micron) clearance single-bolt joint

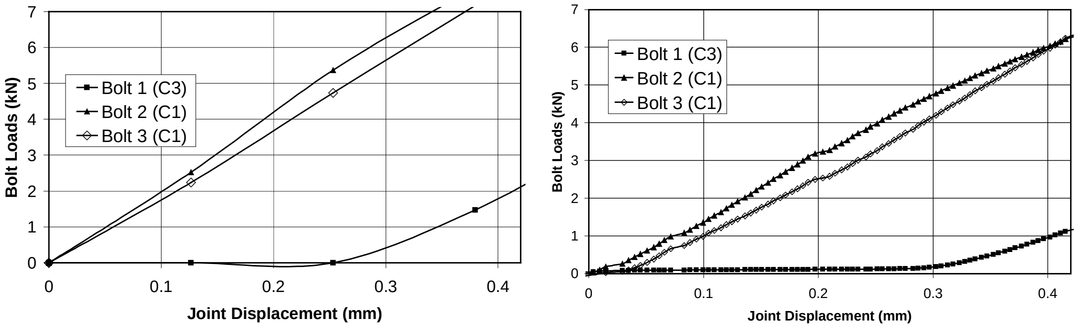
Table 3 Reduction in joint stiffness as a function of bolt-hole clearance – simulations versus experiments –

	Neat Fit (C1)	80 microns (C2)	160 microns (C3)	240 Microns (C4)
Percentage change from C1 (Models)	**-**	**-4.2%**	**-8.5%**	**-11.7%**
Percentage change from C1 (Experiments)	**-**	**-1.9%**	**-7.3%**	**-10.4%**



(a) FE prediction **(b) Experimental results**

Fig. 9 Load distribution in a C1_C1_C1 clearance multi-bolt joint



(a) FE Prediction **(b) Experimental results**

Fig. 10 Load distribution in a C3_C1_C1 clearance multi-bolt joint

Of interest in design is the load taken by each bolt in a multi-bolt joint, so the bolt with the highest load can be identified and analysed for failure. Current design rules indicate that the two outermost bolts in the joint in Fig. 5a would carry the highest loads. Fig. 9 shows that this was confirmed by both the models and the experiments (using instrumented bolts - see [13]) for a multi-bolt joint with all three holes having a neat-fit clearance. The distribution of the loads is seen to be similar in the experiments and the models. The bolt load in question here is the shear load on the bolt and is found from the models by summing contact forces on each hole in the laminate, and from the experiments by measuring the shear strain from $\pm 45°$ gauges placed on the bolt at the faying surface of the joint, and relating this to load via a calibration curve obtained from single-bolt joint tests (see [13]).

To see the effect of clearance in multi-bolt joints, a number of experiments involving different combinations of clearance were carried out [13] and modelled. To illustrate the results, only the C3_C1_C1 case involving a clearance of 180 microns at Hole 1 (with neat-fits at the other two holes) is shown here. Fig. 10 shows the results from the finite element analysis and the experiment for this joint, and it can be seen that Bolt 1 carries no load until clearance is taken up at that hole. During this period, Bolts 2 and 3 take all the load, and interestingly Bolt 2 takes a slightly higher load than Bolt 3. Once Bolt 1 starts to pick up load, the percentage load in Bolt 2 starts to fall back. Thus, the usual assumption that Bolt 2 is not under threat of failure is seen to be questionable with quite small variations in the clearance conditions. Of particular interest here is the close agreement between the instrumented bolt results and the finite element results, which is a validation of both the models and the experimental technique.

4. CONCLUSIONS

Finite element models of single-bolt and multi-bolt, single-shear joints were developed in which the bolt-hole clearance was varied from a “neat-fit” to slightly larger than is found in aerospace primary structures. It was found that increasing clearance had the effect of reducing the stiffness of single-bolt joints and changing the load distribution in multi-bolt joints. Results were compared with experiments and excellent agreement was generally found. Moreover, the finite element models provide insight into the reasons for the effects seen in the experiments.

Further work is currently being carried out on software to automate the quite time-consuming process of producing three-dimensional bolt models, so that in the future such models can become a routine part of the design process. Work is also being performed on incorporating progressive damage models, in order to predict joint failure.

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