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MEASUREMENT OF LOAD DISTRIBUTION IN MULTI-BOLT, COMPOSITE JOINTS, IN THE PRESENCE OF VARYING CLEARANCE

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ABSTRACT

An experimental study is presented on the effects of bolt-hole clearance on the load distribution in multi-bolt composite joints. Single-shear, three-bolt joints were studied. The specimens were manufactured 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. Different combinations of clearance were obtained by using four different reamers and a purpose-designed high-precision drilling jig.

Instrumented bolts with strain gauges attached for measurement of shear strain at the shear plane, were used to measure bolt load distribution. Prior to use in the multi-bolt joints, the bolts were calibrated in single-bolt joints. For assembly of the multi-bolt joints, a special jig had to be designed to allow simultaneous aligning of the bolt along the axis of the joint, centring of each bolt in its hole, and torquing of the bolts to a prescribed level.

The results of the study show that relatively small amounts of clearance can have substantial effects on load distribution. As an example, the middle bolt in a joint of this type is normally assumed to carry less load than the outer bolts, and is therefore considered not to be under threat of failure. But in the presence of clearance the load can shift to the middle bolt, potentially causing an unexpected failure mode.

Keywords: Bolted Joints, Composites, Clearance, Load Distribution, Instrumented Bolts

1. INTRODUCTION

Joints in composite aircraft structures can be obtained by bonding, mechanical fastening or a combination of the two. While bonded joints have advantages such as low weight, distributed load transfer, and perfect sealing of the structure, the low resistance of composites to interlaminar stresses limits the loads that can be transferred. Bolted joints are thus preferred for transferring high loads and have particular relevance for future primary structures. They are also preferred in situations where disassembly is required for inspection or repair.

The analysis of multi-fastener composite joints is typically done in two steps: firstly a determination of load distribution between fasteners; then a more detailed analysis of the most highly-loaded bolt to ensure a safety margin for all modes of failure.

Load distribution analyses are typically done with quite simple models, e.g. spring models or two-dimensional finite element models. These models typically assume idealised conditions such as neat-fit bolt-hole clearances and finger-tight torque conditions, and very little has been published regarding experimental verification of their accuracy.

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This paper presents an experimental study of the load distribution in multi-bolt composite joints. To demonstrate the variations that can exist with "non-idealised" conditions, the tested joints contained variable bolt-hole clearances. The effects of clearance in multi-bolt joints were studied previously by Fan and Qiu [1] in a purely analytical study. The configuration was a four-fastener, single-shear joint made of carbon–fibre laminates and the analysis was two-dimensional. Clearances examined included neat-fit, 30 μm and 60 μm, in 5 mm diameter holes. They found quite significant effects. For example, with zero clearance on all holes, the outer two bolts took up higher loads (about 55%) than the inner two (about 45%). However, when the inner two bolts had a 60 μm clearance, the unevenness in load greatly increased, with the inner bolts taking only about 25% of the load. It was also found that the load distribution in this case became more even as the load increased. However no experimental verification was performed.

The present study involved single-shear, three-bolt specimens, with six different clearance conditions. Load distribution was measured using specially manufactured instrumented bolts. 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. The data produced will be used to evaluate load distribution analysis methods, especially three-dimensional finite element analysis methods that should be capable of capturing such effects. The work forms part of the EU research project "BOJCAS - Bolted Joints in Composite Aircraft Structures" [2].

2. EXPERIMENTAL METHODS

The specimen geometry is shown in Fig. 1. The specimen is a multi-bolt version of a single-bolt joint for which the effects of clearance were studied in [3, 4]; the single-bolt joint geometry was based on ASTM standard D5961/D5961M – 96, [5]. In the present configuration, ratios of width, edge distance, and bolt pitch to bolt diameter were respectively w/d = 6, e/d = 3, and p/d = 4.5, while the diameter to thickness ratio was d/t = 1.6.

The carbon fibre/epoxy material used was HTA/6376, manufactured by Hexcel (UK), a high-strength material currently used in the aircraft industry. The laminate stacking sequence was balanced, symmetric and quasi-isotropic consisting of forty plies ($[45/0/\text{-}45/90]_{5s}$), and yielding a nominal laminate thickness of 5.2 mm when cured. The bolts used were aerospace grade fasteners of a protruding head configuration. They were made from a Titanium alloy, with nominal diameter 8 mm, to an f7 ISO tolerance. A steel nut was also used, together with steel washers at both the head and nut side of the joint.

The codes for the six different clearance conditions are given in Table 1. In the first four cases, labelled C1_C1_C1 to C4_C1_C1, the clearance is varied in Hole 1 only from neat-fit up to a maximum of 240 μm – with neat-fit clearances in the other two holes. In the remaining two cases, the clearance varies in the middle hole as well. The variable bolt-hole clearances were obtained by using (nominally) constant diameter bolts and variable hole diameters. The tooling used for drilling the holes was made of solid carbide and was manufactured specially for the project by an aerospace supplier (Mohawk Europa Ltd). Four different sized reamers used to finish the holes were manufactured to a tight (h6) tolerance. The C3 clearance represents the upper range of clearances found in aerospace structures according to Di Nicola and Fantle of United Technologies-Sikorsky Aircraft [6]. The C4 clearance was studied to examine an out of tolerance situation.

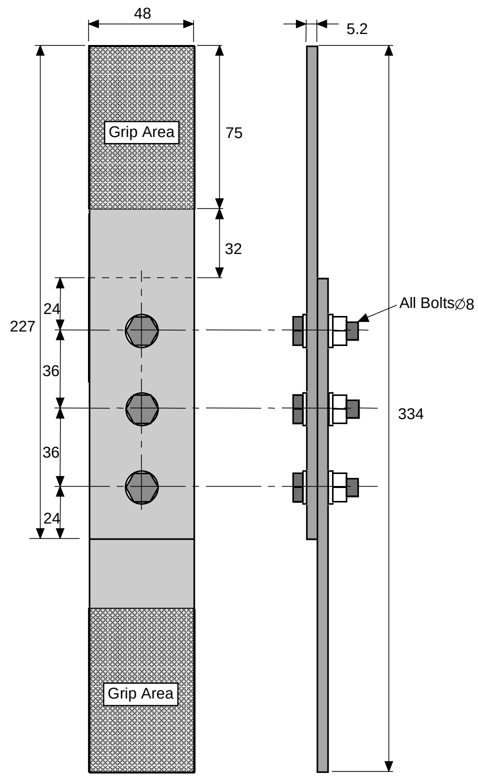

Fig. 1 Single-lap multi-bolt specimen geometry (all dimensions in mm)

Table 1 Nominal Joint Clearances

Code	Nominal Clearance (μm)		
	Hole 1	Hole 2	Hole 3
C1_C1_C1	0	0	0
C2_C1_C1	80	0	0
C3_C1_C1	160	0	0
C4_C1_C1	240	0	0
C1_C3_C1	0	160	0
C3_C3_C1	160	160	0

To measure load distribution, two standard aerospace grade bolts were sent to an external contractor to be fitted with strain gauges as shown in Fig. 2. On either side of the bolt, a shallow slot was milled, into which an axial gauge was affixed to measure axial load, and $\pm 45°$ gauges were affixed to measure shear load. The shear ($\pm 45°$) gauges were located to be at the faying surface (i.e. shear plane) of the assembled joint, thereby measuring shear strain at that location. Lead wires were routed in small holes out through the head of the bolt to the data acquisition system. An etched line in the head of the bolt enabled alignment of the bolt with the direction of load. Theoretically, since the applied load does not cause lateral bending of the bolt, the readings from the axial gauges on either side of the bolt should be the same, so these readings were added together to provide a single (stronger) axial signal – the same was done for the shear gauges. The actual strains were thus half of the measured signals. It should be noted though that the strain field in such instrumented bolts is quite complex due to the presence of the milled slot, and the proximity of the gauges to bolt threads and the bolt head, which introduces St. Venant effects. Thus, the current accepted practice is not to convert to strain but to calibrate the raw voltage against the known applied load.

The external contractor performed calibrations for shear load using a single-lap, single-bolt joint of the same dimensions as used in [3, 4]. Calibrations were supplied for cases where the bolt was aligned with the loading direction, and $\pm 5°$ out of alignment with the loading direction (very little difference was obtained for these three cases). Further calibrations were performed at the University of Limerick using single-bolt joints with four different clearances (C1 to C4). Fig. 3 illustrates the shear load calibration results. Three repeats are shown at each clearance – each repeat involved removing the joint from the testing machine, disassembling and reassembling the joint. The intention was to derive four different calibrations for the four different clearances, but as can be seen there is considerable scatter among the four groups of curves. Eventually it was decided to use a best-fit line through all these curves to represent a single calibration for all clearances. The potential error in using such a calibration is seen from Fig. 3 to be approximately 0.25 kN.

Calibration for axial load was also performed to determine the pre-load to use in finite element models. All tests in this paper were performed with bolts tightened to 0.5 Nm which was regarded as the lowest repeatable torque obtainable, thus representing "finger-tight" conditions. Calibration for axial pre-load at this torque level involved two tests. In the first test, the bolt was tightened to 0.5 Nm and the signal from the axial gauges was measured after allowing some time for relaxation. In the second test, a special fixture employing a spherical joint was used to apply an axial load to the bolt and again the signal from the axial gauges was measured. From the two tests a value for axial pre-load at a torque of 0.5 Nm was obtained. Both tests were repeated several times to get an average value.

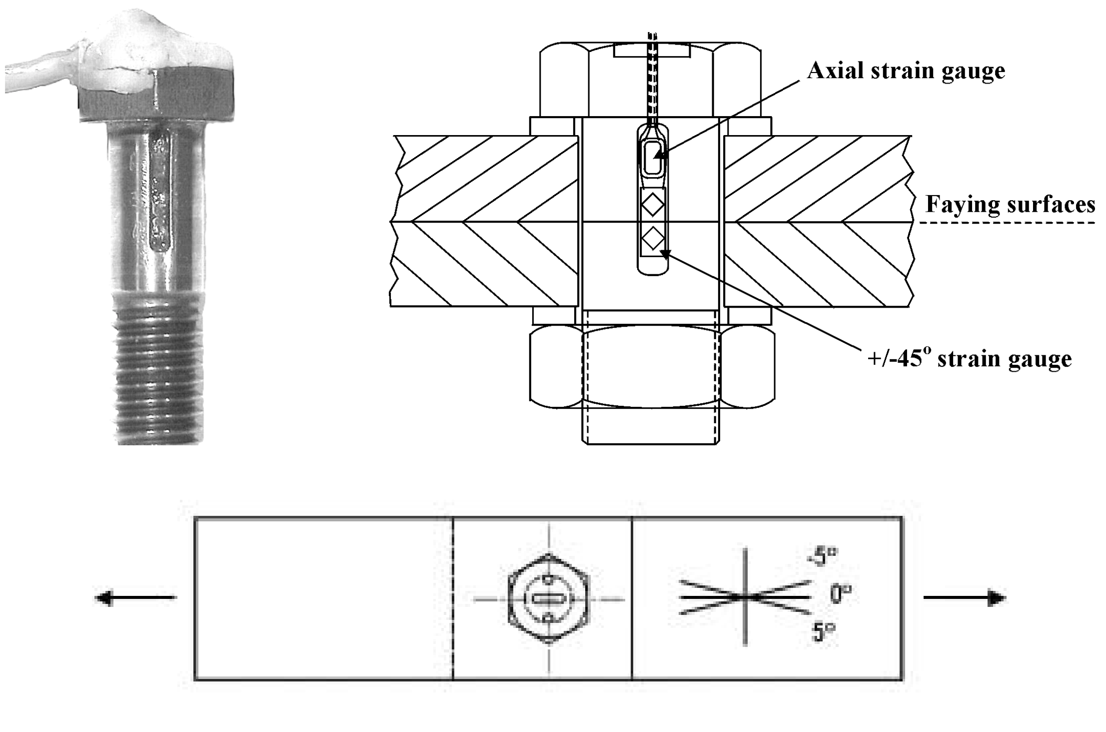

Fig. 2 Instrumented bolt used to measure load distribution

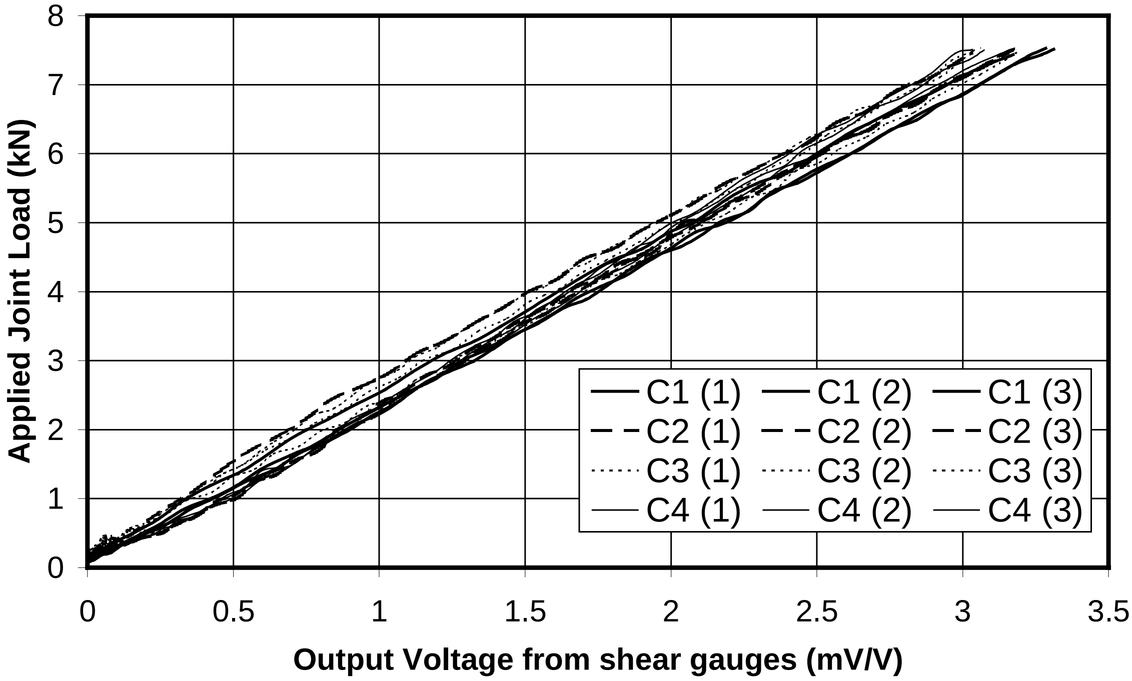

Fig. 3 Shear load calibrations from single-bolt joints – three repeats each of four clearances (C1: neat-fit, C2: 80 microns, C3: 160 microns, C4: 240 microns)

For the multi-bolt joint tests, a special jig was designed to centre the bolts in the holes. Clearly, in practice, bolts would not be centred in holes in an aircraft. However, because of the quite large clearances being used (by aeronautical standards), bolt position within the hole would have a strong impact on the results. To allow comparison with finite element models later, bolt position needed to be carefully controlled. A future study on the effects of bolt position would be possible with finite element analysis if the models could be validated.

The requirements for the design of this jig were quite complex, since three conditions had to be achieved simultaneously. Firstly, the instrumented bolt needed to be aligned to within ±5° of the major axis of the joint so that the gauges accurately measured the shear load. Secondly, each of the three bolts needed to be centred in its hole, and the jig had to achieve this while allowing for variable bolt-hole clearances. Finally, the bolts had to be torqued to a prescribed level of 0.5 Nm representing "finger-tight" conditions. Complicating matters, the wiring and protective cement around the balancing resistors on the head of the instrumented bolts (see Fig. 2) meant that the calibrated torque wrench could not grip the head, and instead needed to grip the nut.

The solution eventually found is illustrated in Fig. 4. A cavity was machined into a block of Delrin®, with the dimensions of a bolt head and washer, with a tight fit. Two other "half" cavities (equivalent to half a bolt head and washer) were machined either side of this central cavity at a distance equal to the fastener pitch in the joints. Half cavities were machined in order to accommodate the wiring and cement on the instrumented bolts (three sides of the instrumented bolt heads were not hindered by these obstacles). The half cavities also aligned the heads, thus strain gauges, in the desired orientation.

Fig. 4 Multi-bolt joint assembly jig used in the instrumented bolt tests to simultaneously centre the bolts in the holes, align them with the loading direction, and torque them to the required level

Since only two instrumented bolts were available (for financial reasons), for each joint configuration, two tests were needed, one with instrumented bolts in positions 1 and 3 (Test 1), the other with instrumented bolts in positions 2 and 3 (Test 2). Fig. 5 illustrates the “Test 1” configuration. Taking the assembly of a C4_C1_C1 joint as an example, for Test 1 an ordinary bolt was first inserted into the central cavity of the assembly jig, and an instrumented bolt was located in one of the peripheral cavities as shown in Fig 4. The laminates were then added and, since the central bolt was in a C1 clearance hole, it fitted Hole 2 tightly, and the instrumented bolt was automatically centred in the C4 hole. The washers and nuts for both bolts were then added and the nuts torqued to 0.5 Nm. The joint was then removed from the jig, and the other instrumented bolt inserted into the remaining C1 (i.e. tight-fit) hole. While keeping the head aligned in the 0° direction with a wrench, the nut was tightened. For Test 2, two ordinary bolts were placed in the jig in positions 1 and 2 (again the tight-fit for Bolt 2 ensured Bolt 1 was centred) and the joint was assembled. After removal from the jig, an instrumented bolt was placed in Hole 3 (a tight C1 fit) and torqued. The ordinary bolt in Hole 2 was then replaced with an instrumented bolt. For the one case involving two clearance holes (C3_C3_C1), the procedure was altered a little but was based on similar principles, with the C1 hole being the locating hole.

This jig relied on precision drilling of the holes, and for this a multi-bolt drilling jig was designed (see Fig. 6). Of critical importance was the location of the two dowel pins, which fixed the position of the edges of the laps of the joint to give a precise overlap length of 120 mm. These pins were located in a stationary back-stop against which the laminates were placed. A moveable slide was then pushed up against the other edges of the laminates to prevent them from moving. The laminates were then clamped in position, with Perspex sheets placed above and below the area in which the holes were to be drilled, to prevent entry and exit damage to the laminate. Drilling and reaming was carried out on a Hurco 3-Axis digitally controlled milling machine. An edge finder in the machine head initially determined the principal locating datum lines consisting of the edge of the backstop and the sides of the dowel pins. These coordinates were entered into the machining program and all holes were subsequently drilled relative to these lines to a high degree of precision.

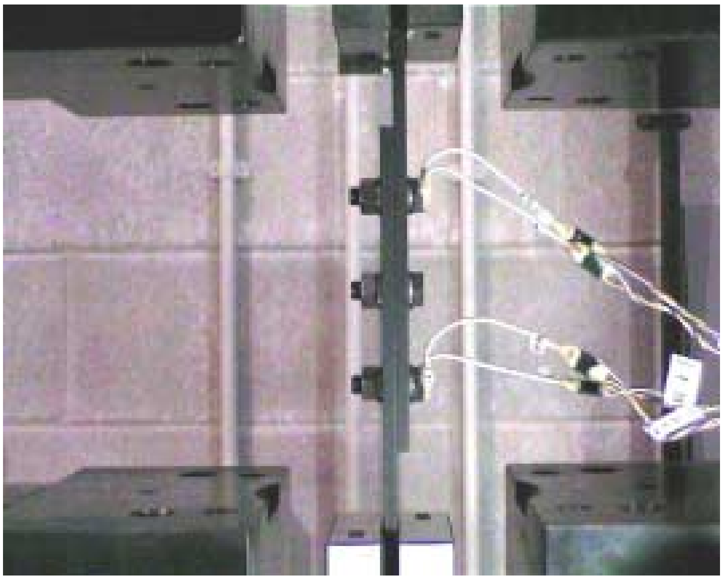

Fig. 5 Test setup with instrumented bolts in positions 1 and 3

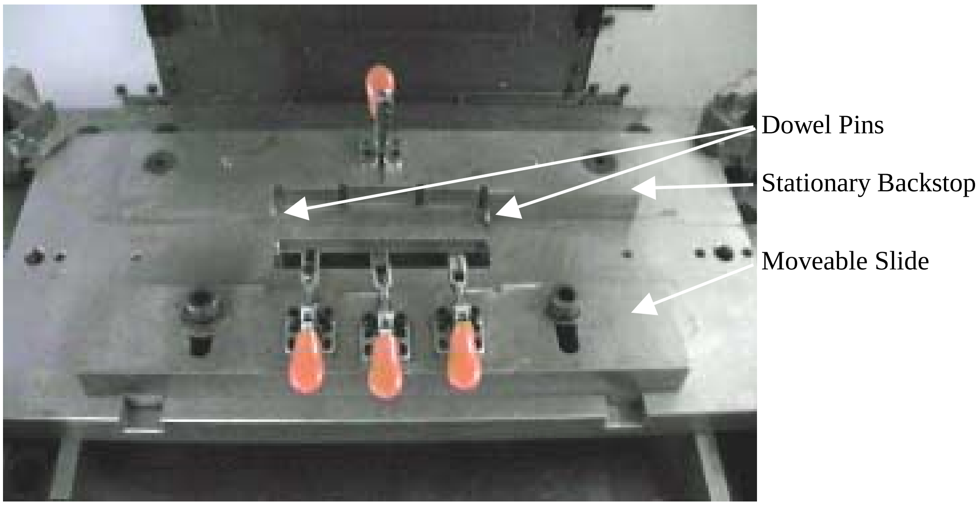

Fig. 6 Multi-bolt joint drilling jig

Care had to be taken when testing, to avoid causing plastic strain in the instrumented bolts. The maximum safe load for the instrumented bolts was conservatively estimated at 8.5 kN. In five of the six configurations, the maximum applied load was set to 15 kN, which meant that if one bolt was not taking any load, and the other bolts shared the load equally, the maximum load in any bolt would be 7.5 kN. The C3_C3_C1 configuration however required extra care, since the C1 bolt was expected to carry most of the load, so in this test the applied load was limited to 10 kN. In tests to failure with ordinary bolts (not shown here) the load deflection curve began to exhibit non-linearity at approximately 20 kN.

3. RESULTS

Fig. 7 shows bolt shear loads versus joint displacement, and bolt shear loads as a percentage of joint load versus joint load, for the C1_C1_C1 and C2_C1_C1 configurations. The joint displacement was obtained from the testing machine stroke, with a correction for machine stiffness, while the joint load was taken from the testing machine load cell. The bolt loads were taken from the shear gauges in the instrumented bolts. The usual assumption in such a three-bolt joint is that the outer two bolts take equal load, while the inner bolt takes less load. This is borne out in the C1_C1_C1 configuration with the outer two bolts taking approximately 35% of the load each at 15 kN applied load. However, in the C2_C1_C1 configuration, it can be seen that even a small change in clearance conditions can change this distribution significantly. In this case, the bolt in the largest clearance hole (Bolt 1) initially does not take any load, since it has been centred in the hole by the positioning jig. Interestingly Bolt 2 (the centre bolt) takes the most load during the period that Bolt 1 takes no load. Once Bolt 1 begins to pick up load though, the percentage load taken by Bolt 2 begins to drop off, and one might speculate that if the joint remained elastic to high enough loads the load distribution might approach that in the C1_C1_C1 case.

It can be seen that the three maximum bolt loads do not add up exactly to 15 kN. Since the effects of friction are expected to be very small due to the finger-tight torque conditions, this anomaly is most likely due to the single calibration equation used for the four different clearances (see previous section). However, for the six joint configurations in Table 1, the sum of the loads estimated in the bolts was *at worst* within 0.7 kN of the total joint load indicated by the testing machine load cell.

Fig. 8 presents the C3_C1_C1 and C4_C1_C1 cases. The main difference in these tests compared to the C2_C1_C1 test is the longer delay in load take-up by Bolt 1, as expected. In the C4_C1_C1 case, Bolt 1 is only just beginning to take up load at the maximum applied load of 15 kN, which as was noted earlier is quite close to the load at which non-linearity begins for this joint. Thus it can be expected that such a clearance situation may have a significant effect on the failure behaviour of this joint.

Finally, Fig. 9 shows the C1_C3_C1 and C3_C3_C1 cases. In the C1_C3_C1 case, it can be seen that the outer two bolts take virtually all the load up to 15 kN applied load. Thus, it can be expected that a clearance in the middle hole may lead to premature failure of the joint compared to the C1_C1_C1 joint. In the C3_C1_C1 joint, Bolt 3 takes approximately 68% of the load at 10 kN applied load, so a premature failure of this bolt is a possibility.

4. CONCLUSIONS

The load distribution in a multi-bolt composite bolted joint in the presence of varying clearances has been measured by the use of instrumented bolts. A special jig designed to centre the bolts in the holes prior to testing, appears to have worked well. The results show that even small variations in clearance can have significant effects on the load distributions in multi-bolt joints. It should be noted that, of the four clearances examined, only the first two (C1, neat-fit and C2, 80 microns) would be commonly found in aircraft joints involving 8 mm bolts. Since the C2_C1_C1 load distribution is approaching that of the C1_C1_C1 distribution at an applied load about 5 kN below the failure load, the effect of clearances up to 80 microns on joint *failure* may be minimal. Only a further test programme to failure loads could confirm this. However, from the results for C3 (160 microns) and C4 (240 microns) clearances, it appears likely that out-of-tolerance clearances could have a significant effect on joint strength.

The experimental method utilising instrumented bolts appears to have worked well, giving results that agree with intuition. Further studies using three-dimensional finite element analysis [7], show good agreement with these experimental results. The main disadvantage of instrumented bolts is their expense, which prohibits using them in tests to failure.

Though the three-bolt configuration tested here is different from the four-bolt configuration used in the analytical study by Fan and Qiu [1], similar results were obtained for the situation where clearance exists in the centre hole(s). Similar also to Fan and Qiu, the load distribution approaches that of the zero clearance (C1_C1_C1) case as the applied load increases (though in reality, failure may happen long before this process completes).

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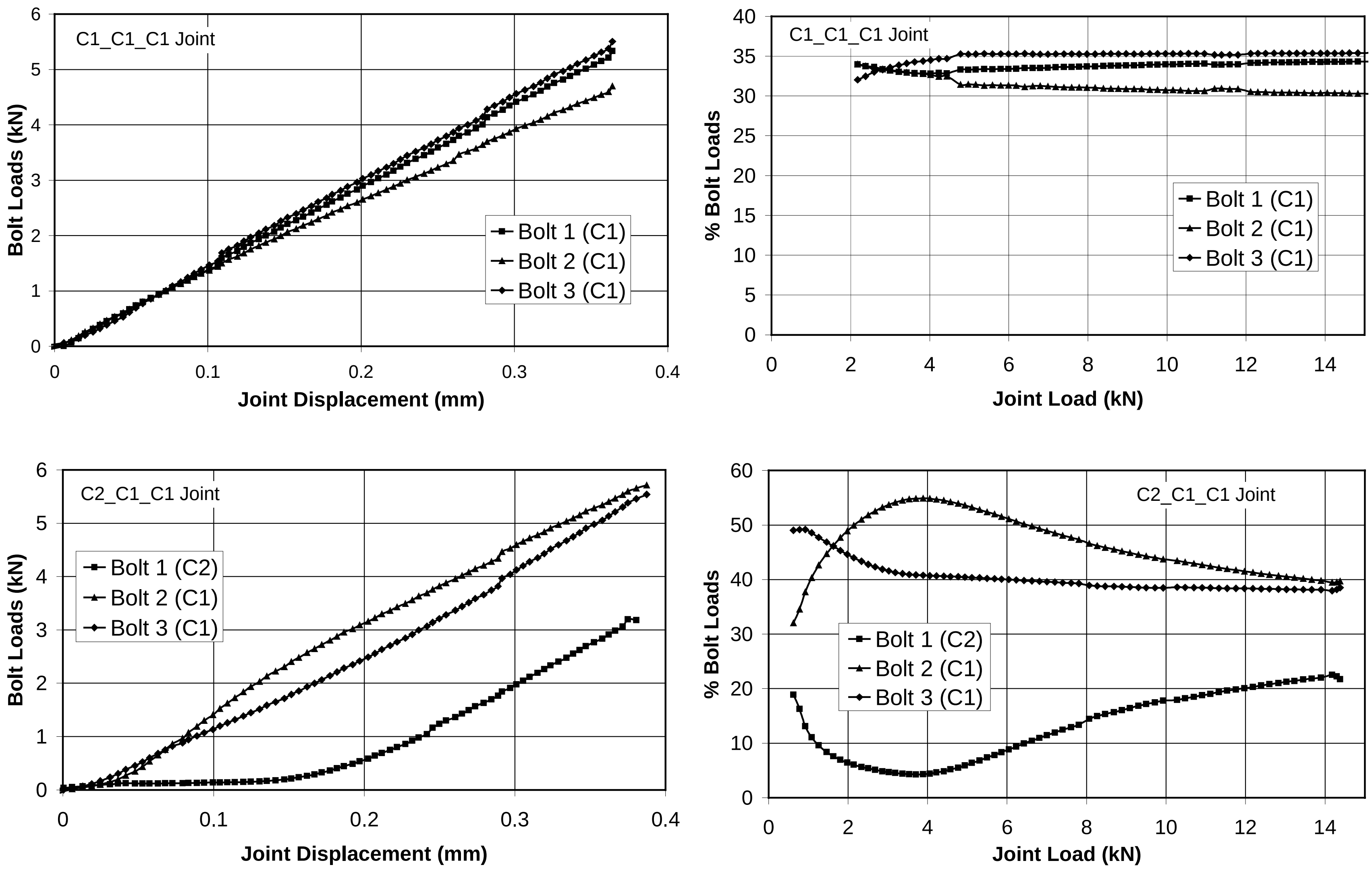

Fig. 7 C1_C1_C1 and C2_C1_C1 cases: bolt shear loads vs. joint displacement, and percentage of joint load taken by each bolt vs. joint load

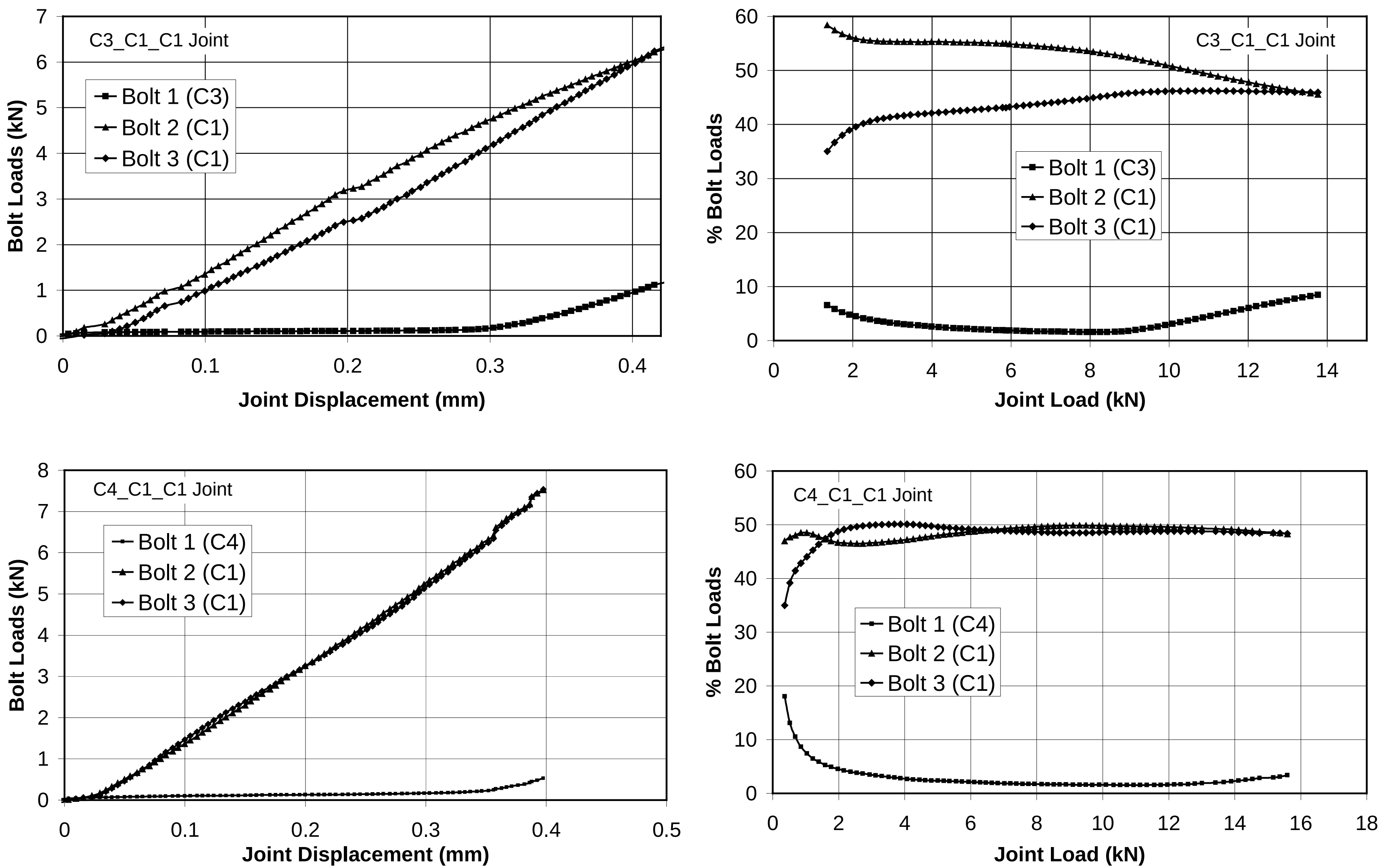

Fig. 8 C3_C1_C1 and C4_C1_C1 cases: bolt shear loads vs. joint displacement, and percentage of joint load taken by each bolt vs. joint load

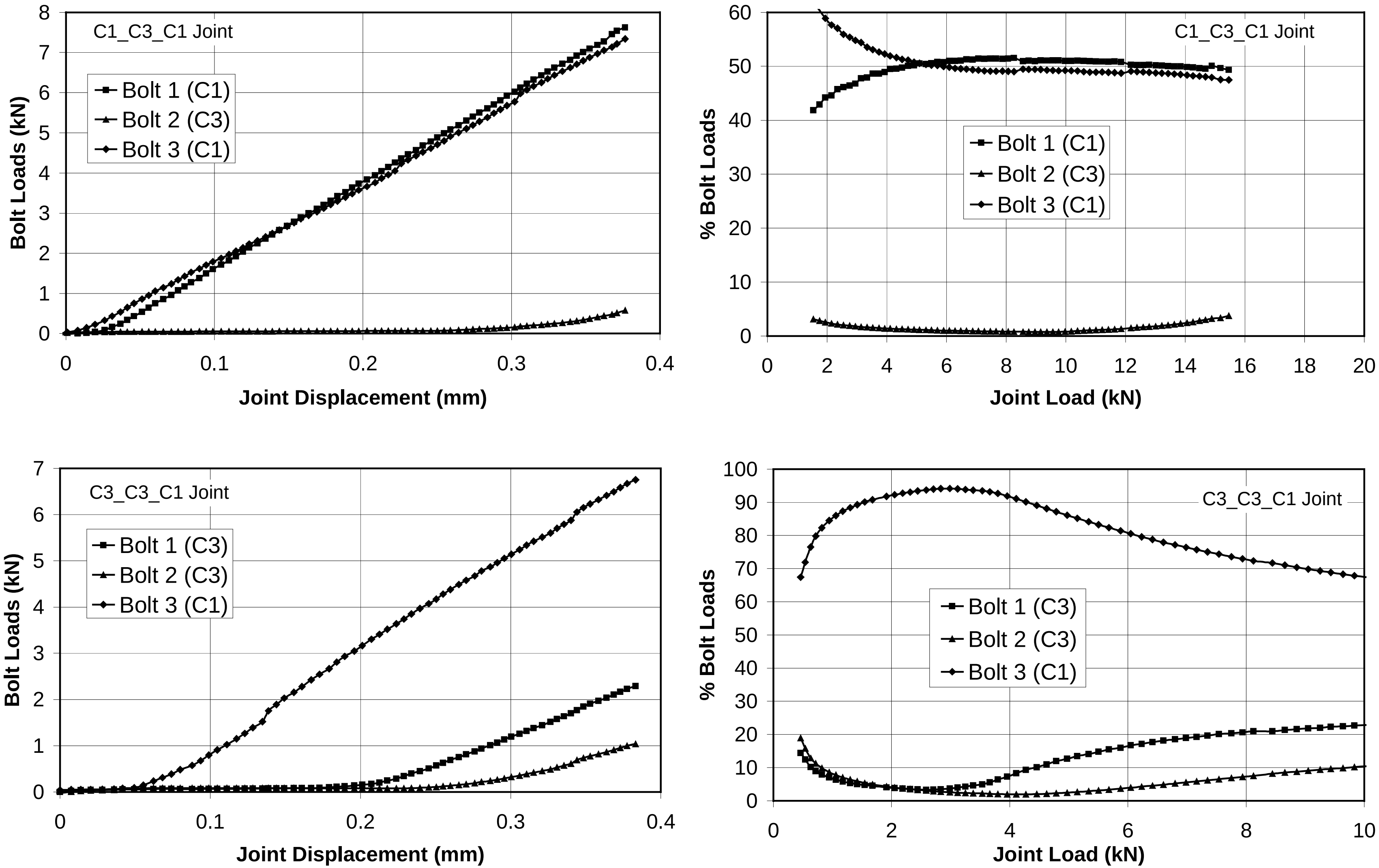

Fig. 9 C1_C3_C1 and C3_C3_C1 cases: bolt shear loads vs. joint displacement, and percentage of joint load taken by each bolt vs. joint load

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