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LOAD TRANSFER MECHANISM IN BOLTED DOUBLE LAP JOINTS

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ABSTRACT

In mechanically fastened single and double lap joints there are two distinct mechanisms of load transfer, through friction between the component sheets and through bearing at the fasteners. When a joint with clearance fitted fasteners is subjected to increasing loads, the load transfer initially occurs through pure friction, but subsequently shifts to a combination of friction and bearing at the fasteners. The ratio between the component of load transferred by friction and that transferred by bearing has a significant effect on the fatigue life of the joint, since the fasteners are stressed only by the load transferred in the bearing mode. This paper describes the results of an experimental study on aluminium double lap joints with a single bolt attachment investigating the effects of friction and clamping force on the mechanisms of load transfer. Friction between the faying surfaces in the joint is altered by the application of a Water Displacing Corrosion Preventative. Plots of axial load versus axial displacement are obtained for different values of clamping forces on the bolt, for joints with and without the WDCP treatment. The load transfer behaviour observed in the tests agree well with that predicted by a finite element model.

INTRODUCTION

In the aircraft industry mechanically fastened joints are still extensively employed to maintain load transmission from one member to another, such as in the construction of the fuselage skin lap and butt splice joints and joints between stiffener runouts under the skin.

The two major concerns regarding the durability and structural integrity of such joints are fatigue and corrosion. The structural members of the airframe being subjected to cyclic operational loads, the cyclic variation of stress concentration at the periphery of the fastener hole gives rise to initiation of fatigue cracks. The propagation of cracks from adjacent closely spaced fastener holes leads to the phenomenon of widespread fatigue damage, wherein the cracks act in concert with one another to accelerate the deterioration of the structure[1,2]. The situation is made worse by the fact that often by the time the fatigue begins to take its toll, the structure would also have accumulated appreciable amounts of corrosion damage. The combined action of fatigue and corrosion can lead to disastrous effects on the structural integrity of the airframe as evidenced by the well known Aloha incident of 1988.

In the last two decades there have been a number of major research programs aimed at devising ways and means to ameliorate the effects of fatigue and corrosion in order to extend the life of the airframe. In the past these attempts at life extension have mainly concentrated at combating one or the other of the two effects. For instance the Aeronautical and Maritime Research Laboratory of the Defence Science and Technology of Australia has conducted extensive research on the fatigue life enhancement of mechanically fastened joints by the introduction of controlled residual stress fields around fastener holes, either by installation of interference fit fasteners or by cold expansion of the holes[3]. Experimental testing of bolted joint specimens at the DSTO under flight by flight fatigue loading sequence have established that compared with joints with close-fit bolts in reamed holes, the fatigue lives of specimens incorporating interference bolts, interference bushes and cold expanded holes were respectively about 9, 5 and 3 times higher[3]. In an even more recent investigation it was found that using thick walled low modulus interference fit bushes can increase fatigue lives by as much as 29 times in comparison to the lives of plates with holes with no bushes[4]. On the other hand both manufacturers and aircraft operators have increasingly relied upon the use of corrosion inhibiting compounds to control corrosion damage in the structure. Of particular interest is the use of water displacing corrosion Preventative (WDCP) compounds which generally consist of a volatile, low surface tension solvent, a non-volatile hydrophobic additive, various corrosion inhibitors and a film former such as oil, grease or resin[5]. WDCPs have been found to very effective in penetrating into the interfaces between faying surfaces of airframe structural joints and arresting corrosion development. Thus the battle is being fought at both fronts separately, the underlying assumption being that keeping either of the enemies at bay will lead to an increase in the durability and reliability of the structure. However, this assumption may not hold good if the means of controlling one damage leads to the enhancement of the other. Specifically, in the last few years concerns have been raised regarding the possible detrimental effect of the use of corrosion prevention compounds on the fatigue life of joints[6]. These concerns revolve around the fact that friction between the faying surfaces plays a major role in determining the extend of load transfer through the fasteners and hence the initiation and propagation of cracks at the fastener holes. The application of oily film water displacing corrosion prevention compounds, while efficiently inhibiting corrosion, can also lead to a reduction in the friction coefficient between the faying surfaces and hence an increase in the fastener bearing load. The potentially degrading effect of reduction in friction on joint durability has to date mostly been overlooked, perhaps due to the conservative practice of ignoring the beneficial effect of friction in the design of mechanical fastener joints on the airframe. However in assessing the benefits to life extension by arresting corrosion damage with the use of CPCs, one should also take into consideration the possible reduction in life due to the higher load transmitted through the fasteners. It is far from clear at this stage that the net effect is a significant improvement in the residual life of the joint.

Previous experimental work on fatigue cracking of mechanically fastened joints have conclusively shown that in joints in which a large fraction of the load is transmitted through friction, the initiation of cracks occur at the faying surfaces away from the edge of the hole; whereas in joints which predominantly transfer load through bearing at the fasteners, fatigue cracks usually initiate at the edge of the hole[7]. Thus a change in friction characteristics not only affects the magnitude of the load on the fasteners but also the location of crack initiation. It is obvious that the crack propagation and the fatigue behaviour of the joint will significantly be influenced by both these factors. The Australian Defence Force Academy and the Aeronautical and Maritime research Laboratory of Australia is currently involved in a collaborative experimental program investigating the effect of CPCs on the fatigue behaviour of medium load transfer joints representative of stiffener runouts in the airframe with and without corrosion. The experimental work is being complemented by finite element modelling aimed at simulating the effect of CPCs on the fatigue behaviour of the joints. For the estimation of residual life of these joints based on the fracture mechanics approach of predicting crack growth from stress intensity factors, it is imperative that both the location of the crack initiation site and the stress distribution in its vicinity are accurately determined. It is therefore important to understand clearly the role played by friction in the load transfer through the joint and the effect CPC has on it. The work reported herein is a preliminary investigation conducted on double lap joints aimed at obtaining this basic understanding. The double lap joint, although it is not representative of any joint in the aircraft, is chosen for its simplicity, in order to reduce the number of variables affecting the load transfer behaviour to a minimum. The symmetric geometry of the joint eliminates secondary bending characteristics to a large extend. Further, complexities due to fastener geometry and interaction between fasteners is minimised by the use of a single straight shanked bolt for the specimens. The study focuses o the two main parameters affecting friction, the fastener clamping force and the coefficient of friction between the faying surfaces. The tests also served as a means of measuring and establishing friction coefficients between surfaces treated with WDCPs.

TEST SPECIMENS

The dimensions of the specimens employed in the experiments are shown in Figure 1. The specimens consisted of three rectangular panels of aluminium alloy sheet 7075-T6 of width 25 mm and length 160 mm. The thickness of the middle plate as well as the two side plates is equal to 3mm. Each plate was drilled with a 5.3 mm diameter at a distance of 25 mm from one edge. Before joining, the faying surfaces of all the plates were polished with 3M polishing paper P320 under running water to ensure that they were flat, clean and scratch free. The plates were joined together with a 5mm diameter hexagonal steel bolt and a hexagonal nut, with a 3mm thick aluminium alloy (7075-T6) washer on either side of the plate. Additional 1mm aluminium washers were also employed between the thick washers and the bolt-head and the nut respectively as indicated in Figure 1. The use of the 5mm diameter bolt in the 5.3 mm diameter holes provided a radial clearance of about 1.5 mm between the fastener and the holes in each of the three component plates. The faying edges of the 3 mm thick washers were also polished to the same degree as the component plates using the P320 polishing paper. It is to be noted that after each loading and unloading, the joint was disassembled and all the faying surfaces repolished to remove all the scratch marks and achieve the same degree of flatness and smoothness as before. Two groups of specimens were prepared for the experimental work. The first group contained polished specimens without any other surface treatment, while the specimens in the second group were treated with LPS2, a commercial Water Displacing Corrosion Preventative (WDCP), commonly employed in the aerospace industry. The WDCP was applied in controlled amounts of 0.1 ml, to each of the specimens after it was polished and assembled (with only a nominal amount of clamping force) through the edges, using a graduated pipette. A time duration of about 5 minutes was allowed after the treatment to permit seepage of the WDCP into the interfaces by capillary action, before the full clamping force was applied and the specimen mounted in the test rig for loading.

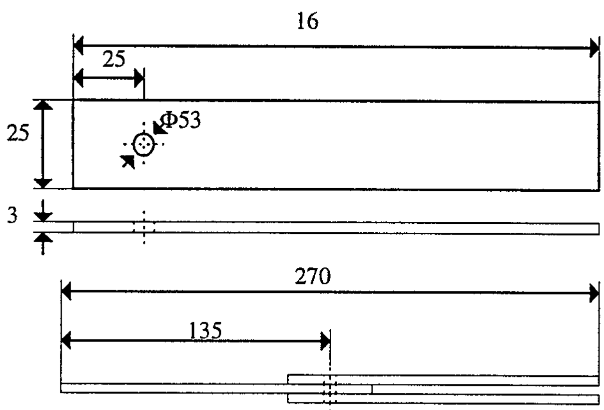

Figure 1. Double Lap Joint Specimen Geometry (All dimensions in mm).

The clamping force in the bolt was measured using a specially designed transducer, a schematic of which is shown in the assembly in Figure 2. The transducer consisted of a cylindrical tube insert with two flats machined on diametrically opposite sides and instrumented with strain gauges as indicated. The clamping force transducer was calibrated by loading it axially with weights and recording the strains. To correlate the clamping force with the torque applied to tighten the bolt, a number of preliminary tests were conducted by applying known amounts of torque to the bolt using a calibrated torque wrench and measuring the axial forces from the strains recorded. This was intended to produce a consistent correlation between the torque applied and the clamping force in the bolt, so that the clamping force transducer could be eliminated and the reading from the torque wrench employed directly to determine the magnitude of the clamping force. However it was found that due to variations of friction coefficient in the screw threads of the bolt and the nuts, there was considerable variation (about 20 to 30%) in the slopes between the torque applied and the measured clamping force. This inconsistency in the relation between torque applied and clamping force, due to its sensitivity to the tangential friction in the bolts, has been observed and reported before[8]. It was therefore decided to employ the clamping force transducer and use its strain gauge readings directly to determine the clamping force in all the tensile tests performed.

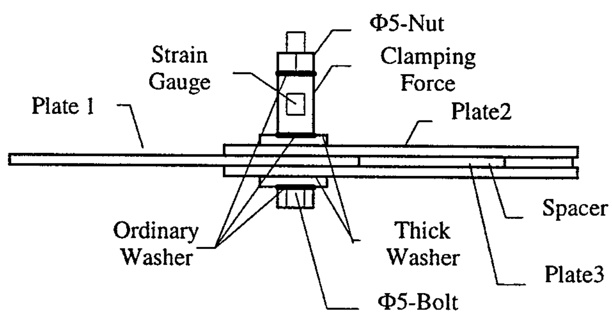

Figure 2. Attachment of Fastener Clamping Force Transducer.

EXPERIMENTAL SET-UP

The tensile tests on the double lap joint specimens were performed on a J.J. Lloyd mechanical testing machine with a capacity of 20 kN. The specimen was held using friction grips within the jaws of the machine, employing a spacer at the end with the two outer plates (Plates 1 and 3 in Figure 2). The axial load applied to the specimen was read directly from the load cell on the J.J. Lloyd. It was however found that the LVDT displacement data from the J.J. Lloyd was unsuitable for accurate measurement of the axial elongation of the specimen, since it was affected by the relatively low stiffness and the play in the grips and connections in the machine. Therefore a displacement transducer consisting of a capacitance based proximity probe was designed for measuring the elongation between two known points on the specimen - one on the middle plate and another on one of the outer panels, as indicated in the experimental set-up in Figure 3. The attachment of the proximity probe to the specimen proved very tricky, as initial experimentation with cantilever beams bonded on to the specimen showed that the transducer was sensitive enough to pick up the variations in the relative displacement between the cantilevers caused by the bending induced in the component panels when the joint was axially loaded. To eliminate the displacements caused by the bending rotation of the plates, it was necessary to simply support the measuring beams. This was achieved by resting the measuring beams on thin brackets bonded to the desired points on the specimen on one side and similar brackets attached on a rigid support at the other side.

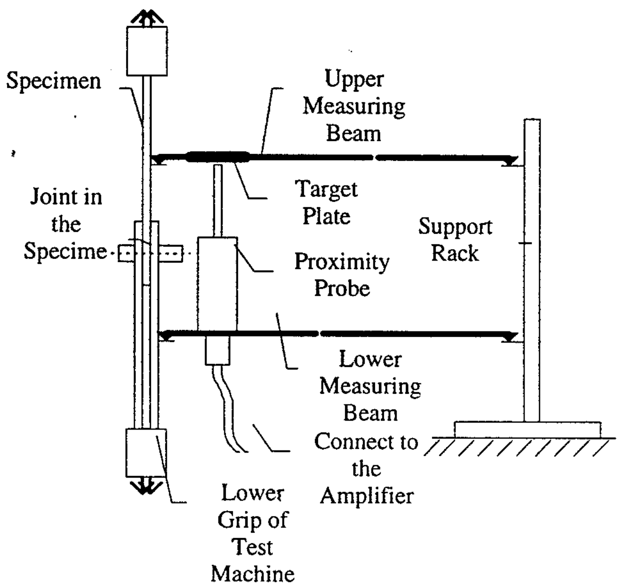

Figure 3. Experimental Set-up for Displacement Measurement.

The brackets on the rigid support were mounted at approximately the same heights as the corresponding ones on the specimen so that the beams are initially held horizontal. This provided simple support conditions on either end of the beams, eliminating rotations caused by the bending of the joint's outer panels under axial loading as well as bending of the measuring beams themselves. The proximity probe was sensitive enough to record displacements of the order of 1 micron. The reading from the proximity probe, the load cell in the J.J. Lloyd and the clamping force transducer were directly recorded on a computer for plotting, data reduction and analysis.

FINITE ELEMENT MODEL

The load displacement behaviour of the double lap joint with a clearance fitted bolt fastener was initially studied with a static elastic 3D finite element model. The modelling was performed using the commercial finite element package ANSYS 5.3. Figure 4 shows the finite element meshing of the full model. In the model, the bolt head and the nut at the other end was assumed to be of the same form and the presence of the washers and the clamping force transducer was not considered. A value of 5.4 mm was employed for the fastener hole diameter and the bolt diameter was 5mm. Taking advantage of symmetry only the model of one quarter of the joint, as shown in Figure 5, was employed in the analysis. The details of the fine mesh in the vicinity of the fastener hole and the model of the straight shank bolt are shown in Figures 6 and 7. Eight noded 3D structural solid elements (Solid 45) with 3 degrees of freedom at each node were employed for modelling all the components. The outer plate is modelled with 3 elements across the thickness and the middle half plate has two rows of elements across the thickness. The contact between the fastener head and the plates, the bolt shank and the plates as well as those between the plates are all modelled using 3D point to surface contact elements (Contact 49). Due to the presence of the contact elements the analysis requires a non-linear procedure, with the boundaries between the different surfaces checked at each load step for determining contact and establishing the contact forces. Since no friction measurements had been performed, a friction coefficient of 0.25 was assumed between all interfaces for this preliminary finite element study.

The clamping force was applied by introducing a thermal strain in the fastener, by subjecting it to a thermal load, after assigning a finite value to the coefficient of thermal expansion (CTE) in the axial direction of the bolt elements, and setting the CTE in the radial direction to zero. The thermal load was varied until the desired clamping force was achieved in the fastener. The analysis was performed in a displacement controlled mode, with load step increments consisting of axial displacements at the end of the middle plate of about 0.003 mm. Figure 8 shows the plot of the far field axial stress in the middle plate versus axial displacement obtained from the analysis of the model with a clamping force of 4 kN. It can be seen that the load displacement behaviour of the model agrees well with the behaviour expected from basic structural analysis.

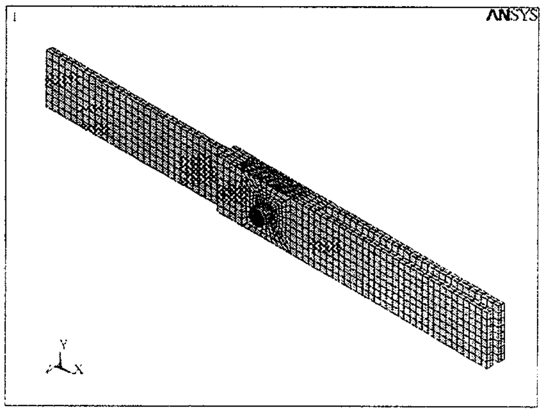

Figure 4. Finite Element Model of Full Double Lap Joint.

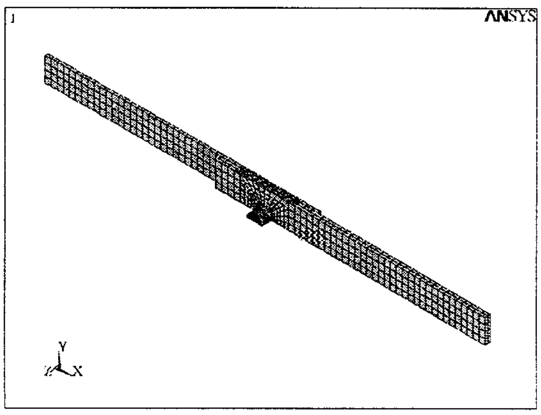

Figure 5. Finite Element Model of One-Quarter of Double Lap Joint.

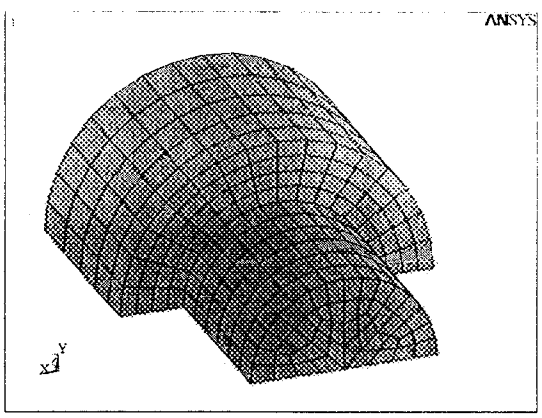

Figure 6. Detail of Mesh around Fastener Hole.

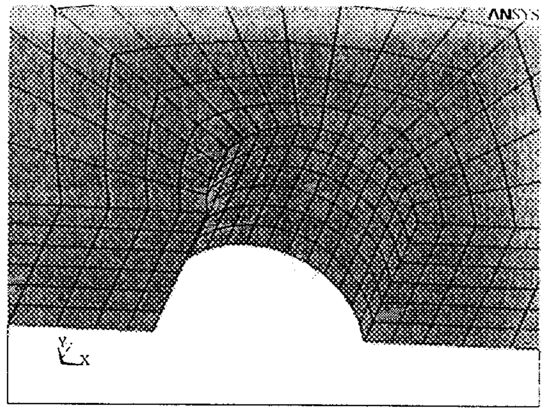

Figure 7. Quarter Model of Straight Shank Bolt.

At the commencement of the loading, the bolt is placed concentrically along the axis of the holes in the three plates, with a radial clearance of 0.2 mm. In the initial part of the load history (region OA in the plot in Figure 8), when the axial load is resisted by frictional forces between the horizontal surfaces in contact, the load transfer occurs purely through friction, the elongation in the joint is purely due to the axial strain in the three plates and the slope of the load-displacement diagram is governed by the in-plane stiffness of the plates. Slip occurs in two stages. In the first stage, A to B, the middle plate slips in the direction of the applied load on it, by overcoming friction at its two interfaces with the outer plates, and comes in contact with the bolt shank. In the region B to C the load transfer is partly through friction and partly by bearing of the middle plate on the bolt. At this stage the clearance between the bolt and the outer plates is still maintained. The second stage of slip, C to D, is the slipping of the outer plates in the direction of the load acting in them, which occurs when the applied load exceeds the friction at all four interfaces - the two between the outer plates and the middle plate and those between the outer plates and the bolt head and the nut. At D, the end of the second slip, the outer plates also come in contact with the fastener shank. The load transfer from D to E occurs through friction and bearing of the bolt on the plates, hence the slope at this stage is governed by the in-plane stiffness of the plates as well as the bending stiffness of the bolt. Since the slip in the first stage involves friction at two interfaces, and the second stage involves friction at four interfaces, the ratios of the axial loads at which these occur to the clamping force are respectively equal to twice and four times the friction coefficient between the interfaces in the joint.

Figures 9 and 10 respectively indicate the contour plots of two principal stresses around the fastener hole at the end of the loading sequence, at point E.

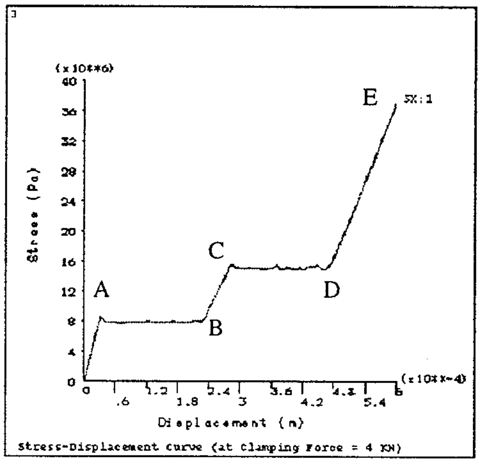

Figure 8. Load Displacement Diagram of the Double Lap Joint in Tension.

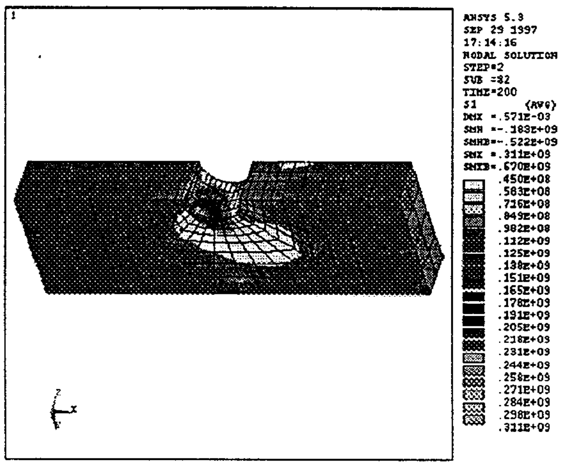

Figure 9. Contour Plots of Principal Stress S1 at end of Loading Sequence.

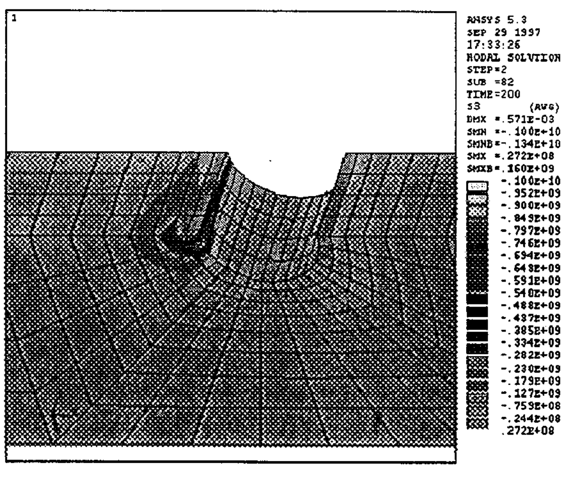

Figure 10. Contour Plots of Principal Stress S3 at end of Loading Sequence.

TESTS ON SPECIMENS WITHOUT WDCP

The specimens were loaded in the J.J. Lloyd under displacement control at a constant loading rate up to a maximum value of about 7 kN of the axial force. Figures 11, 12 and 13 depict the load histories of the test specimens without any WDCP, for clamping forces of the order 1 kN, 2.3 kN and 3.2 kN respectively. The actual clamping force in each joint as measured by the clamping force transducer is also plotted in the figures. As can be seen from the figures, the load-deflection diagrams closely follow that predicted by the finite element modelling (Figure 8). It is to be noted when the specimen is unloaded no slip occurs indicating that that the bolt remains in contact with the plates even when the joint is completely unloaded. Thus a compressive load will need to applied to the joints before the original clearance between the bolt and the plates can be reinstated. This suggests that if a clearance fitted joint is subjected to cyclic loading with a positive R value, ie under purely tensile loads, all the slip will occur in the very first loading cycle and the load transfer mechanism will involve both friction and bearing at the fasteners in the subsequent cycles. The coefficients of friction computed from the ratios of the average values of the axial load at which slips occurred in the first and second stages to the applied clamping force is indicated in the left hand corner of the figures. It can be seen that the values of the friction coefficients are reasonably consistent, although there is an appreciable reduction in the friction coefficient computed from the second stage of slipping in Figure 13 when the clamping force was about 3.2 kN. This may have been caused by the noticeable reduction in clamping force as evident in the sharp drop in the clamping force plot at the beginning of the second slip in Figure 13. It may also be noted that the load path becomes increasingly non-linear and steers away from the horizontal as the value of the clamping force increases. This is perhaps due to the high degree of fretting contact and degradation of the specimen surface when slip occurs under high values of frictional load. In tests conducted on specimens with very high values of the clamping force, it was observed that the load history became highly non-linear and no clear stages of slip and non-slip could be distinguished (see Figure 14), indicating that other mechanisms such as localised cold welding may come into play.

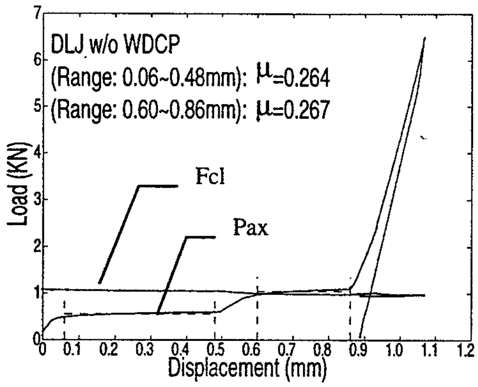

Figure 11. Load history of Specimen without WDCP (Clamping force = 1 kN).

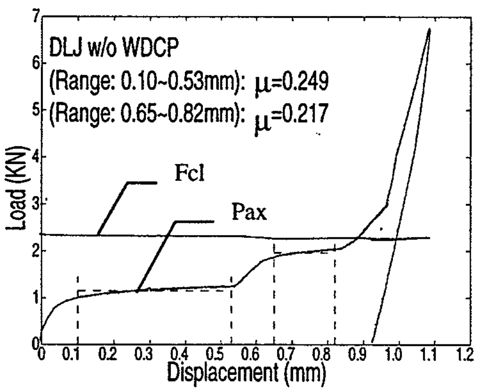

Figure 12. Load history of Specimen without WDCP (Clamping force = 2.3 kN).

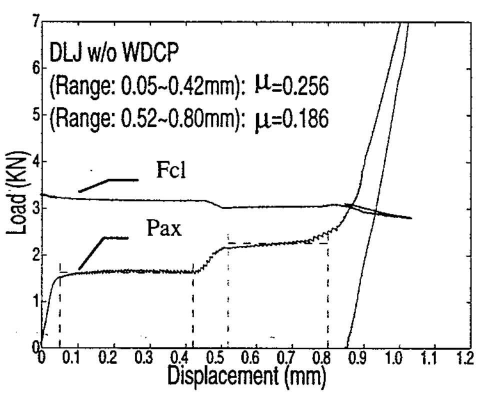

Figure 13. Load history of Specimen without WDCP (Clamping force = 3.2 kN).

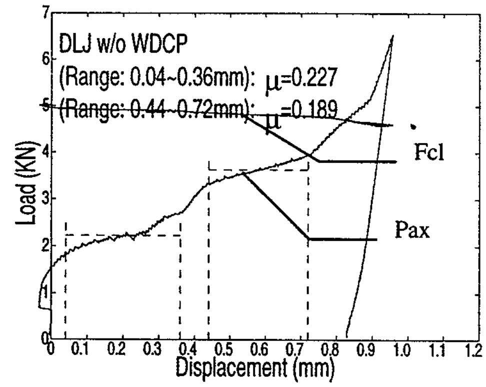

Figure 14. Load history of Specimen without WDCP (Clamping force = 5 kN).

Figure 15 shows the plots of the average values of the axial loads measured for the first and second stages of slip against the corresponding clamping forces in about 20 tests performed on specimens without any WDCP. The values of the friction coefficient as computed from the slopes of the linear regression analyses of the two groups of data respectively yield values of the friction coefficient $\mu = 0.246$ for the first stage and $\mu = 0.215$ for the second stage. Noting that in the second stage the axial force has to overcome about double the frictional force as in the first stage for friction to occur, this seems to indicate that as the friction force increases there may be a reduction in the effective coefficient of friction between the faying surfaces.

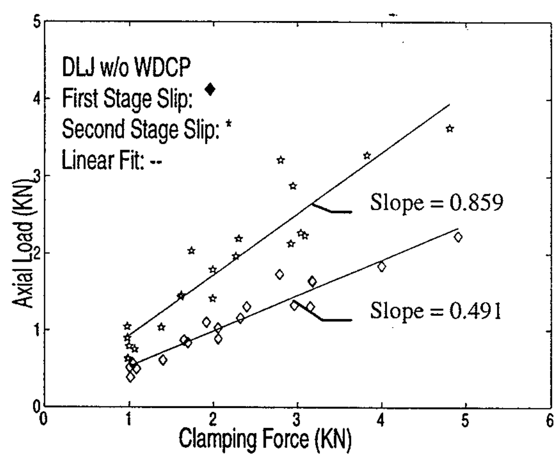

Figure 15. Plot of Axial Force against Clamping Force for Specimens without WDCP.

TESTS ON SPECIMENS WITH WDCP

Figures 16, 17 and 18 depict the load histories of the tests on specimens without any WDCP for clamping forces of the order 2.1 kN, 3.1 kN and 4.2 kN respectively. Once again, the load-history of these specimens closely follow that predicted by the results of the finite element modelling (Figure 8). As before the figures contain graphs of the actual clamping force measured in each case, as well as the coefficients of friction computed from the ratios of the average values of the axial load at which slips occurred in the first and second stages to the applied clamping force. It can be seen that the values of the computed friction coefficients are approximately equal, with no consistent increase or decrease with increasing values of the applied clamping force. Once again there is a sharp change in the measured clamping force at the beginning of the second slip in the case of the highest clamping force as seen from its plot in Figure 18.

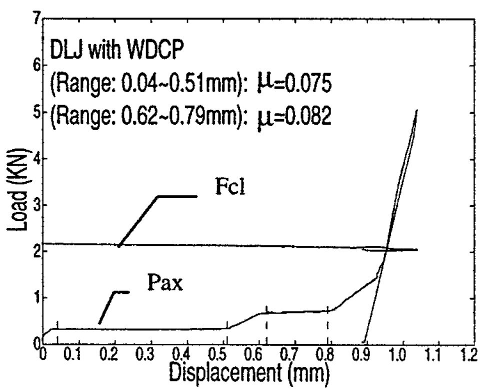

Figure 16. Load history of Specimen with WDCP (Clamping force = 2.1 kN).

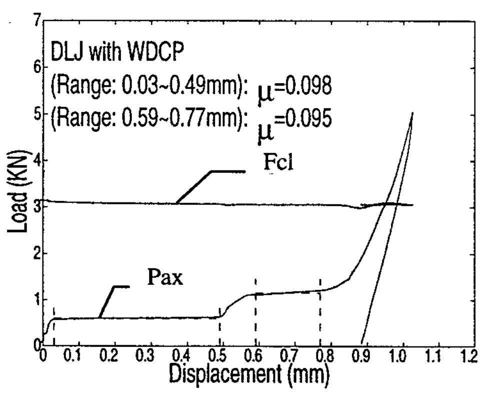

Figure 17. Load history of Specimen with WDCP (Clamping force = 3.1 kN).

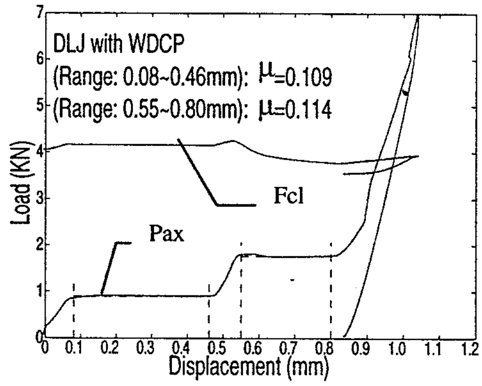

Figure 18. Load history of Specimen with WDCP (Clamping force = 4.2 kN).

The average values of the axial loads measured for the first and second stages of slip against the corresponding clamping forces in the 20 tests performed on specimens with WDCP are shown in Figure 19. The values of the friction coefficient as computed from the slopes of the linear regression analyses of the results of specimens with WDCP yield values of $\mu = 0.098$ and $\mu = 0.097$ for the first and second stages respectively. Thus a major effect of the application of the Water Displacing Corrosion Compound seems to be reduction in friction coefficient from about 0.23 to 0.97, ie by an order of about 60%. The total force transmitted through friction in the joint is reduced to about 40% of its original value when the faying surfaces of a double lap joint is treated with corrosion prevention fluids. It follows that there is a corresponding increase in the load carried by the fasteners, which obviously has major implications on the life to crack initiation and crack growth behaviour at the fastener holes in the joint.

CONCLUSION

The results of an experimental program on the load transfer behaviour of double lap joints fastened with a single straight shank bolt with a clearance fit is described. The tests were conducted on specimens with carefully controlled surface finishes at the interfaces with and without treatment with Water Displacing Corrosion Prevention compounds. The load displacement plots of the test specimens with low to moderate clamping forces indicate two distinct stages of slip in the joints, one when the middle plate translates

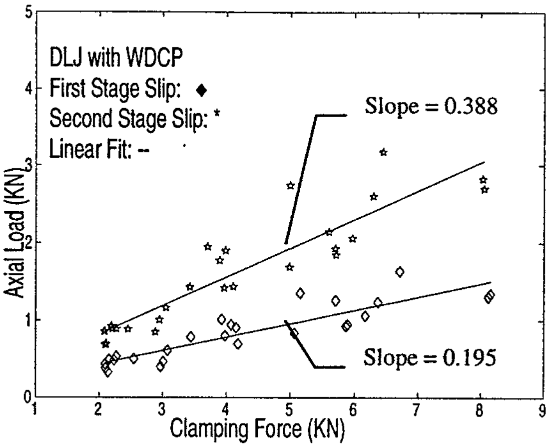

Figure 19. Plot of Axial Force against Clamping Force for Specimens with WDCP.

in the direction of the applied load on it and the second when the outer plates translate in the direction of the loads applied on them. The axial loads at which the slips occur is linearly proportional to the applied clamping force indicating a fairly constant value of the friction coefficient at low to moderate values of the clamping force. At high values of the clamping force, the correlation between clamping force and axial force becomes markedly non-linear which can be attributed to degradation of the surface quality by the relative movement of the faying surfaces against each other and point to other mechanisms such as localised cold welding coming into play. The application of the WDCP reduces the friction coefficient between the faying surfaces to about 40% of that of the untreated surfaces, indicating an increase of about 60% on the load transmitted through bearing at the fasteners, which has major implications on the fatigue life of the joint. Further work in terms of finite element modelling of the joints with the experimentally determined values of friction coefficients with and without the WDCP treatment is currently under progress for establishing locations of critical stress distributions and identifying possible crack initiation sites.

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