Experimental Mechanics (2013) 53:1073–1104
DOI 10.1007/s11340-013-9736-3

# 2012 William M. Murray Lecture: Some Curious Unresolved Problems, Speculations, and Advances in Mechanical Fastening

## Issues and Opportunities for Research in Joining

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Received: 19 February 2013 / Accepted: 21 February 2013 / Published online: 26 April 2013


**Abstract** Mechanical joining is one of the oldest, most important, and most neglected aspects of engineering design of machines and structures of all types and sizes. Approximately 250 U.S. companies manufacture fasteners worth over $8 billion per year. There are 18,000 fasteners in a common fighter jet airframe, and fasteners account for roughly one-third the cost of a typical airplane. Yet, most failures of structures, including aircraft, originate at fasteners, suggesting that improved understanding of fastener mechanics, better design criteria, and informed applications of fundamental knowledge are required. This issue is exacerbated by increased demands on systems, particularly in the transportation and military sectors, and by the growing use of composites, for which current fastening practice seems to be underdeveloped owing to the complexities of material structure and response. This lecture first traces a brief history of mechanical joining, its importance, and the problems faced by engineers in designing for fastening. Research and development of fasteners through analysis and experiment are complicated by the large array of variables involved, and investigators must have at hand an extensive array of experimental and analytical techniques as well as an appreciation of the practicalities of fastening. Verification and validation of findings are crucial, and extrapolation is fraught with pitfalls. Described subsequently are some examples of experimental results that generate speculation and that might provide points of entry for investigators who are willing to take on difficult challenges where progress would be valuable but is not easily realized. These cases include, among others, odd and perhaps dangerous behaviors resulting from coldworking holes in engines and structures, impact stresses caused by joint slippage in composites, the use of inserts to control stress concentrations, difficulties in applying sufficient clamping force in composites, merits of hole shaping, and unusual configurations of conical washers. Finally, some ideas for hybrid joining and for systems that allow quick field assembly/disassembly are briefly described.

**Keywords** Fastening · Bolting · Composite materials · Coldwork · Inserts · Hybrid joining

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## Introduction and Orientation

### Purpose, Origins, Scope, Limitations

We offer this written version of the SEM 2012 William M. Murray Lecture with abiding humility and devout appreciation upon being selected as the Lecturer. The journey has been long and enlightening. May this paper be accepted with interest as well as the granting of grace for its probable shortfalls.

This presentation is limited to mechanical fastening, already too large a topic for a single paper, but with the addition of some results and thoughts about hybrid fastening. We hope that a similar treatment of adhesive fastening for metals, plastics, and composites will soon appear.

A main purpose of this presentation is to point out several areas in which quality research on mechanical fasteners is needed in order to gain understanding of the basic mechanics of fastening as well as to improve fastening systems as we seek to design structures and vehicles that are more efficient in both fabrication and performance. Works performed by the author and his colleagues over a span of many years for manufacturing concerns, the U.S. Army, and the U.S. Air Force are briefly described to illustrate the origins of the problems still to be faced and the limited progress achieved thus far. The list of cases is neither exclusive nor is it all-inclusive. It is possible, indeed likely, that, in the interim, some of the issues mentioned have already been addressed.

A variety of suggestive experiments and results are offered minus complete descriptions of techniques and with less-than-thorough discussion of the full implications of what was learned. Much of this work has not been published before, partly because it is still not complete. Further, considerable speculation is indulged because it leads to insights as to what needs to be done and it creates an environment for the development of potential advances in fastening technology. A fundamental engineering point of view is adopted whenever the option is offered.

Finally, this exposition is not a proper research paper, as most experimental details are missing. Nor is it a review paper. Given the unique character of the Murray Lecture and space considerations for this print version, little attempt is made to provide a comprehensive list of citations to the works of the author, let alone of the many other investigators, particularly experimentalists, who have labored mightily to gain useful results. Rather, only a variety of the most relevant and most accessible publications are cited whenever possible, perhaps with an unintended bias that might be perceived as ego-centric; and these will lead to the greater body of literature. We hope that none feel slighted by the necessary choices made.

## Does Mechanical Fastening Deserve our Attention?

To begin to answer this question, let us ponder just a few of the available relevant statistics:

- Over 200 billion industrial fasteners are used in the USA every year, and 26 billion of these are absorbed by the automotive industry alone.
- The current annual value of fasteners used in the USA alone is about \$10 billion and is projected to reach \$13 billion in 2013.
- There are about 350 dedicated manufacturers of fasteners in the USA.
- Approximately 18,000 fasteners are used in the F-18 jet fighter, which means, if the average weight of a fastener is only 0.5 oz., then this rather small airplane is carrying over 500 lbs. of bolts.
- Fasteners typically comprise about 1/3 the cost of an airplane, the same as the engines.
- Most machinery and infrastructure failures originate at fasteners. In the late 1970's, an air force study showed that approximately 70 % of aircraft failures involved mechanical fasteners.
- Field assembly, disassembly, repair, modification, and reassembly are important in many applications, particularly in military vehicles and heavy equipment.

Many of us take fasteners for granted both in our design work and in everyday life. After all, we can select from hundreds of types at any home center, the local hardware, or a machine shop supply depot; so it is natural that we limit our design options to what is easily accessible. This trend is unfortunate from both engineering and economic viewpoints. A senior engineer at a large maker of mechanical fasteners told the author many years ago that, "Fastener choices are almost always made too late in a project, resulting in necessary redesign and serious delays, if not outright failures and recalls."

In fact, many unanswered questions and opportunities for research exist in this field, and these questions are of increasing importance with growing demands for structures that are lighter, safer, and stronger. Unfortunately, fastener research tends to be ignored outside a relatively small but dedicated cadre of investigators, and necessity often dictates that fastener design choices be empirically energized. Two factors seem to drive this trend. One is that fastening problems are very difficult to investigate experimentally or analytically because they are inherently nonlinear and because of the large number of parameters involved, so generalized publishable results are not easily acquired. Another factor is that, given value in the competitive marketplace, development is done in-house, is not broadly based, is not sponsored, and publication of results is not rewarded.

## An Example—Airplane Fastener Failure

To illustrate the insidious nature of fastener failure and the huge cost penalties derived from such failures, consider the recently discovered and highly publicized problems with mechanical joinings that grounded the new super jet airliners that are manufactured by one of the world's largest makers of aircraft. In one case, two types of cracks were found in the wing ribs of an airplane that had experienced only 399 flight cycles in an operating time of 2,454 h. These airframes incorporate around 2000 rib brackets with L-shaped feet that carry bolt holes for attaching the wing skins to the ribs. Two types of cracks were found. The ones of most interest were hairline cracks in the area where the rib foot is bolted to the wing skin. The cracks ran from the bolt hole to the edge of the rib foot. The manufacturer claimed that these failures are caused by high stress and the aluminum alloy chosen for the brackets. He further stated that the cracks are not the result of a design flaw!

Another Example—Composite Automobile Wheel

Ponder another unforeseen joining problem that contributed to the eventual termination of a promising and expensive engineering program. Figure 1 is a photograph showing a composite automobile wheel that was developed several years ago by a major manufacturer. This wheel demonstrated many favorable features including an excellent ride, reduced unsprung mass compared with a typical steel wheel, outstanding appearance, freedom from corrosion, and predicted economies in production. It was actually introduced with success on a famous line of "muscle cars." Extensive road testing brought to light two related problems, namely: (1) during mounting of the wheel, it was difficult to obtain consistent and sufficient clamping forces when tightening the lug nuts, and (2) the lug nut clamping force was not dependably retained during use.

We were asked by the manufacturer to identify the causes of these problems and develop, if possible, an acceptable solution. Preliminary examination suggested that a key factor was the nature of the lug nut bearing chamfers, which were machined after fabrication of the wheel. Many sample sections were cut from wheels for optical and electron microscopy of the bearing chamfers in various combinationsof conditions including as-manufactured, after various treatments of the chamfers, and after mounting the wheels following standard practice. Figure 2 shows one sample electron microphotograph of the bearing chamfer in its virgin state. Note the fine, sharp stubble of fiber ends that protrude above the machined surface. These and many similar observations suggested that the fiber stubble was preventing full tightening of the lug nuts, as represented in the sketch of Fig. 3. The short stubble, which is sharp and stiff, evidently resists bending and compaction and so creates high friction, even digging into and scratching the lug nut surface, with the result that the lug nuts cannot be fully tightened onto the chamfer. We reasoned that this effect would occur even if sharp carbide tooling were used in the post-machining process so as to cut the fibers flush, or nearly so, with the matrix material. Most of the bearing forces would still be resisted by the fiber ends.

Several solutions to this problem were suggested. The best, which was discovered by the manufacturer even before

Fig. 1 Composite automobile wheel

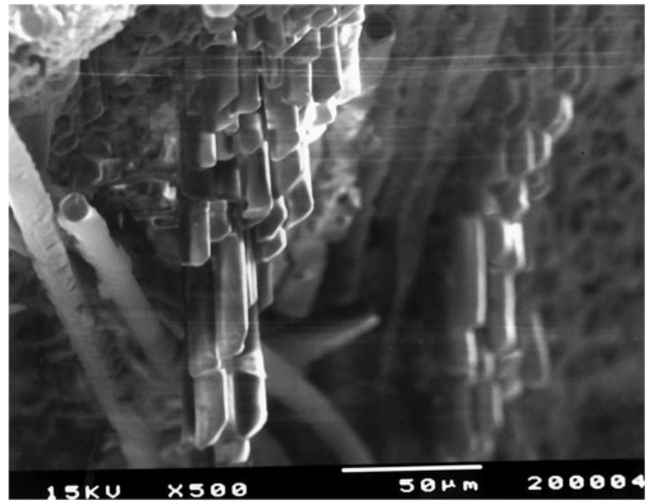


Fig. 2 Sample photomicrograph of machined lug nut bearing chamfer in composite wheel. Photo courtesy of Dr. Kristin B. Zimmerman

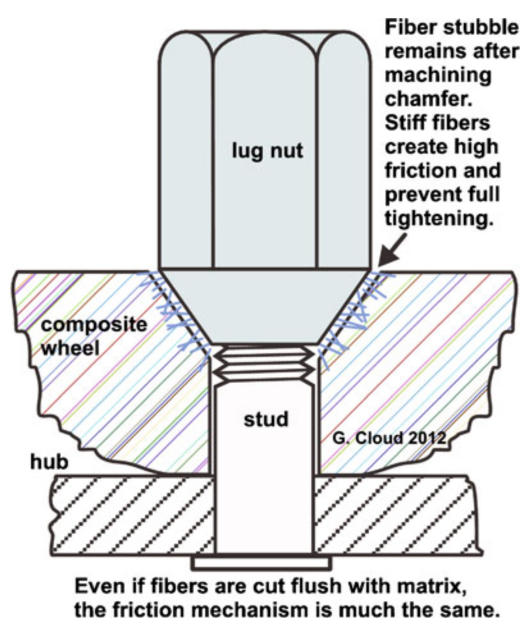


Fig. 3 Probable mechanism preventing full lug nut tightening for the composite wheel

we were asked to look at the problem, was to use a common antiseize product on the chamfers. The particles in the antiseize fill the spaces between the fiber ends so as to smooth the bearing surface, and the lubricant facilitated tightening the lug nut. A single application was found to be effective through numerous mountings of the wheel. Other solutions suggested and tried with various degrees of success included, among others: (1) case hardening the lug nuts to prevent the digging-in of the fiber ends; (2) dry lubricants containing Teflon ®, graphite, and similar materials; (3) simply removing the soft galvanize layer from the nuts; (4) using a thin metal interface that snaps into the chamfer and remains in the wheel; and (5) not post-machining the chamfers so that the lug nuts would bear directly on the resin-rich surface layer, a solution which raised the possibility of creep and wear with subsequent loss of clamping.

The difficulty with all solutions examined is that they are not acceptable in aftermarket sectors because of safety concerns. Tire emporiums have a standard practice for mounting steel and aluminum wheels. They cannot guarantee that, for example, adequate antiseize compound would be in place after, say, the fifth mounting of a given wheel during a tire rotation. As mentioned above, this promising wheel development program was dropped in near-market stage for a number of reasons, one of which was the lack of an appropriate solution to the lug tightening problem.

## Parameters Involved in Fastening

A major issue in fastener investigations at any level is that a great number of parameters must be considered. Some of these are inherent in the nature of mechanical fastening (e.g. clamping force), and others might be introduced as a potential mechanism for improving fastener performance (e.g. changing hole shape). The following is an incomplete, non-exclusive list of variables that immediately come to mind as important:

- material thicknesses,
- fastener type (bolt, cap screw, rivet, etc.),
- material properties and the mix of materials (e.g. composite to metal),
- hybrid joinings (mechanical plus adhesive or insert),
- preload clamping,
- loss of preload (creep and relaxation),
- fastener size and scaling effects,
- bolt-to-hole clearance,
- interference fits,
- tapered threads,
- washers (flat, spring, conical),
- hole shape (non-round holes, tapered holes, etc.),
- fastener shape (non-cylindrical, tapered, etc.),
- hole treatment (coldworked, peened),
- inserts and bushings,
- fastener arrays (shape and spacing),
- edge distance,
- torque (shear) effect upon tightening (especially for laminates),
- blind fasteners (one-side access),
- impact, blast, stress and shock waves,
- manufacturing methods.

We conclude that, when performing theoretical analysis, numerical modeling, or experimental investigation, the test matrix tends to become very large if any generally useful results are to be obtained. A related very important fact is that validation and verification of all models and experiments are necessary, given the complexity of fastener mechanics. Major dedication of resources is required. This prospect can be either daunting or stimulating, depending on viewpoint.

## Concerning Methodologies

Owing to the complexities of fastening research, particularly as experimentation is necessary, critical attention must be given to measurement and test methods. At the fundamental level, the investigator should hold at the forefront of the mind and preach vigorously to all involved the principles that undergird good experimental practice, which may be summarized as follows:

- Choose the experimental method to fit the problem, not modify the problem to fit a favored technique, as is too often the case.
- Know the difference between accuracy and sensitivity.
- Measure exactly what is needed to minimize data manipulation.
- Analyze error propagation, or it might burn you.
- Keep experiments as simple as possible.
- Calibrate often.
- Work in differential mode if you are able.
- Extrapolation can be useful, but it cannot be trusted.
- Validate and verify results by independent methods.
- Avoid circular reasoning when interpreting results.
- Design a double-blind protocol if possible, although this option is often not viable in the physical sciences.

The complexity of fastener research and the principles of good experimentation dictate that the investigator must have at hand and thoroughly understand a diverse array of measurement techniques. Chosen from the many approaches available in our laboratory, the following measuring devices and methods have been used in our fastening research.

- photoelasticity
  - transmission
  - embedded polariscope 3-D

- digital speckle pattern interferometry
  - surface
  - embedded 3-D
- white-light speckle photography
- three-axis, six-beam moiré interferometry
- moiré grating photography with optical Fourier processing
  - surface
  - 3-D with multiple embedded gratings
- resistance strain gages
  - surface and embedded
- fiber optic strain gages
  - Fabry Perot and Bragg types
  - surface and embedded
- instrumented bolts and washers
- optical measuring microscopy
- electron microscopy
- laser extensometry
- assorted testing machines.

## Issues Arising from Coldworking Fastener Holes

Following a brief introduction to the coldworking process and experimental methodology, this part describes in varying degrees of detail six experiments that address certain questions connected with the common practice of cold-expanding fastener holes to improve fatigue performance. No claim is made that these questions have been fully and generally answered. A summary of arising questions that remain to be answered through experimental and analytical research is offered at the end of this section.

### The Cold-Expansion Process and Summary of Potential Problems

For many years, fastener holes have been cold-expanded in order to induce a residual stress regime around the hole boundary that resists the initiation and propagation of cracks, particularly in structures that are subject to cyclic loading [cf. 1–4]. The coldworking is usually implemented by one or another of several available proprietary procedures that involve forcing an oversize mandrel or ball through the hole. Often a sleeve is used, which may or may not be left in the hole depending on the process and application. Coldworking has been extensively used in rework or salvage situations to extend the fatigue life of structures, but it is also implemented in OEM applications. Typically applied to metals, coldworking and interference fits have been explored in recent years for use on certain types of composites and sandwich structures [5, 6].

The efficacy of coldworking fastener holes in joining practice has been well demonstrated. There remain, however, several areas of concern with this procedure that have safety implications, and it seems that these issues have not yet been adequately investigated. The following is a summary list of these issues, which are closely inter-related:

- effects of in-plane compression on the residual stress/strain field,
- interactions between holes,
- hole shape and position after coldworking,
- effect of coldworking order on the residual stress field,
- stress induced in near plate edges,
- fastener arrays: pattern and spacing,
- stress concentration increased in certain locations,
- three-dimensional nature of the stress field.

The following sections outline first the experimental methodology and then present sample results drawn from research on the problems mentioned that highlight behaviors that are unexpected and that could raise questions about the wisdom of prescribing coldworking in the absence of justification through adequate testing. The coldworking equipment and process marketed by J.O King, Inc., which uses a solid sleeve, were utilized as the standard in these studies. One experiment used the so-called "split sleeve" technique. Not all of the issues mentioned above are specifically addressed in this presentation.

### Methodology

Characteristic of problems that involve elasto-plastic material deformations is that the process involves several discrete non-reversible steps and a very broad range of strain. Most of our studies of coldworked fastener holes were accomplished through the use of intermediate-sensitivity moiré, which requires high-resolution grating photography plus Fourier optical filtering. The procedure is described in detail in various reports and books [3, 4, 7–12], so it will not be repeated here. It does seem worth the while to remind ourselves of a few advantages of this technique that make it especially suitable for the problem at hand:

- Data for all stages of the experiment are permanently stored.
- Any two stages of the experiment can be compared in differential mode.
- Sensitivity multiplication is easily attained after data storage.
- Sign and degree of pitch mismatch can be chosen during data processing.

- Signal/noise ratio ranges from acceptable to excellent even after some grating damage.
- Fringes are closely packed so dependable displacement and strain values can be obtained.

Figure 4 shows a much-reduced clip from a moiré fringe pattern that was extracted from these experiments. In this instance, pitch mismatch is imposed, but there was no fringe multiplication. In other cases, fringe multiplications of two and three were utilized to create a fringe density that is too high to be adequately reproduced in this publication.

## Effect of In-Plane Compression Stress on Residual Stress/Strain

### *The Question and its Origins*

The question posed is this, "Do compressive in-plane loads modify the favorable residual stress and so reduce or even negate the benefits of coldworking?" The question apparently arose when coldworking was being adopted to extend the service life of military aircraft. The sketches of Fig. 5 illustrate the origins of the problem. Suppose that the fastener holes in the wing skins of an airplane were coldworked. The finished airframe is then subjected to several landing and take-off cycles, some of which are probably severe, so as to create large alternating in-plane compressive and tensile loads in the wing surfaces. This effect is easily seen if you look out the window of a airliner so as to observe wing deflection during even ordinary take-off or landing. If you happen to look along the wing surface under glancing light, you might notice

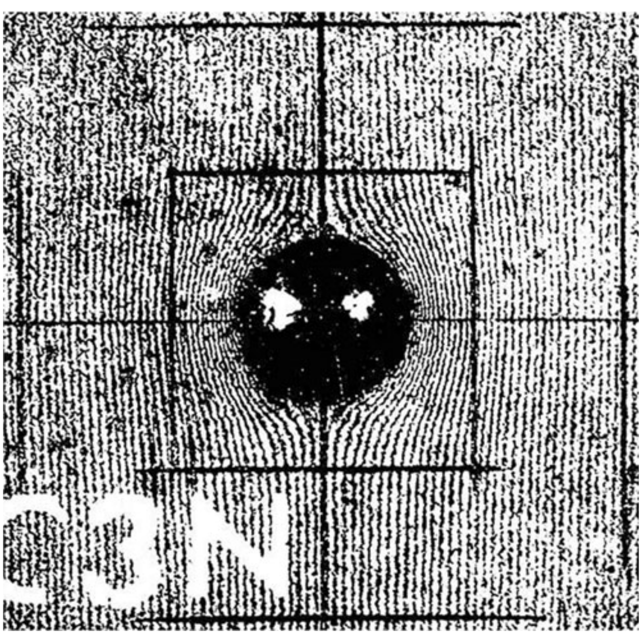

**Fig. 4** Sample clip from intermediate-sensitivity moiré pattern for a coldworked hole

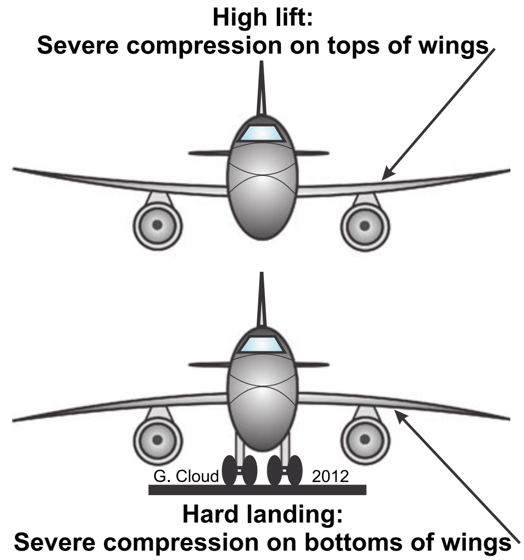


**Fig. 5** Motivation for research into effects of in-plane compression on coldworked fastener holes

bulging of the skin between the rows of fasteners, which indicates that localized elastic plate buckling is occurring. Such behavior is entirely acceptable, but imagine what would happen if one of the rows started to "unzip."

### *Experiments*

Laboratory tests were conducted to determine whether cyclic in-plane compressive loads modified the residual stress/strain state near a single coldworked hole in coupons of 7075-T6 aluminum that were 0.25 in. thick [13, 14]. Figure 6 shows the essentials of the experimental scheme as well as the results for one of the three experiments conducted.

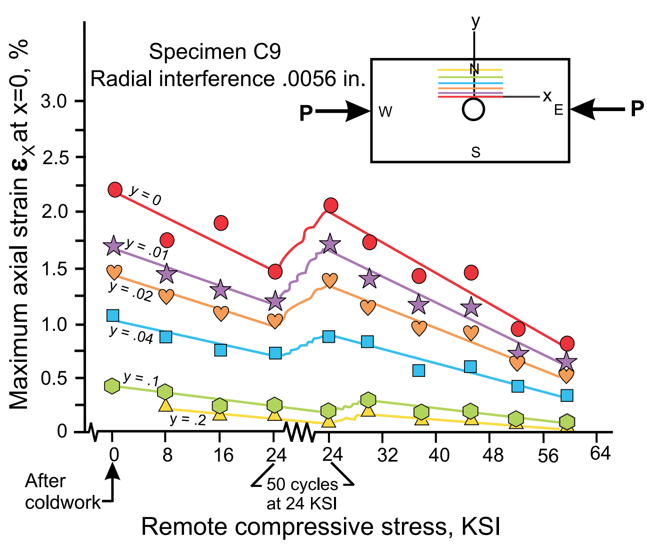


**Fig. 6** Example of measured losses of residual strain near coldworked holes caused by in-plane compression and cyclic loading

After applying the moiré gratings, photographing them, and analyzing the results to establish baseline data, the holes were coldworked and the residual strain profiles along several longitudinal *x*-axes at various *y*-distances from the hole were determined by the technique named above. Of particular interest were the maximum longitudinal strains ($\varepsilon_x$) at points where the chosen *x*-axes intersected a *y*-axis passing through the center of the hole, and these values could be simply picked off the strain profiles. In other words, $\varepsilon_x$ at the critical points is determined as a function both of applied load and of *y*, where *y* is the distance from the hole edge. The first set of points in the figure, marked "after coldwork" show these initial strain values, which agreed with those obtained in comprehensive static experiments [3, 4].

Now, after the coldworking and initial strain measurement, the specimen was placed in slotted sockets in a testing machine so at to apply an increment of in-plane load that created a far-field compressive stress of 8 ksi. Subsequently, the specimen was removed from the loading rig and grating photographs were recorded and processed in order to measure again the strains at the points mentioned above, and these values were plotted as shown in the figure as the second set of data points. This process was repeated for 16 and 24 ksi far-field stresses.

After measuring the strains following application of static 24 ksi stress, the specimen was put back into the testing machine and subjected to 50 cycles of 0–24 ksi far-field stress, after which the strains were again measured. Twenty-four ksi was chosen as the peak value of the cyclic stress because, at the time, this value was considered to be a target design stress level for aircraft wing skins.

Subsequent to the cyclic load, static far-field stresses were repeated at increments up to 60 ksi, at which level plastic buckling of the coupon became evident and the experiment was terminated.

### *Observations*

As expected (or feared), the residual strain produced by coldworking decreased consistently with even small values of static in-plane load and continued decreasing until plate buckling occurred. A most startling observation was that the residual strains at points within .04 in. of the hole edge somehow recovered to approximately the starting (after coldwork) values during the load cycling at 24 ksi. After the first increment of static load following cyclic exposure, the residual strain at points nearest the hole was found to again decrease, but the strains at the two points furthest from the hole recovered to original values. Subsequently, increased static loading caused consistent decline of the residual strains at all points, reaching, in the end, values approximately half the starting values.

The recovery of residual strain during or immediately after cyclic loading caused considerable anxiety. It was contrary to thoughtful expectation. Is this the truth? If so, what is the cause and what is the meaning? Confidence in these findings was gained by re-analyzing the stored grating data several times with the same results. A control specimen that was not coldworked was tested with the identical procedure, and there was no effect of in-plane load. Later, a second coldworked specimen was created and tested by different people with somewhat different equipment and procedures in a different laboratory, and behavior that was similar but not quite identical to that of the first specimen was observed.

### *Summary*

The following general remarks were derived from investigations of the effects of in-plane loads on coldworked fastener holes:

- The moiré technique involving grating photography and optical spatial filtering proved to be well suited for research on coldworked holes.
- Except when cyclic loads were applied, the residual strains decreased linearly with static load.
- The residual strain and, likely, the favorable residual stress that is the goal of coldworking are compromised by application of in-plane compression.
- The residual strain recovered to its original static value upon the application of cyclic loading at the standard design stress.
- The recovery of residual strain with cyclic loading could not be rationalized except in a general way having to do with redistribution of residual stress. This behavior seems to be not widely known, nor, to our knowledge, has it been explained.

## Effect on Residual Strains from Coldworking Adjacent Holes

### *The Question*

Usually, mechanical fasteners are used in an array of one sort or another, and, in many cases, the fasteners are relatively close together. Careful measurements have shown that, in metals, cold expansion of a hole creates elastic stresses well outside the local plastic zone. The following question arises: How are residual strains between holes affected by coldworking adjacent holes in an array? There is no easy answer to this question given the large number of parameters involved, and space limitations prohibit exhaustive discussion. We present here only a few limited results that underline the complexity of the problem and the potential risks involved.

### *Example Experiment*

Figure 7 illustrates a 0.25-in.-thick specimen of 7075-T6 aluminum with a 5-hole in-line array of fastener holes that

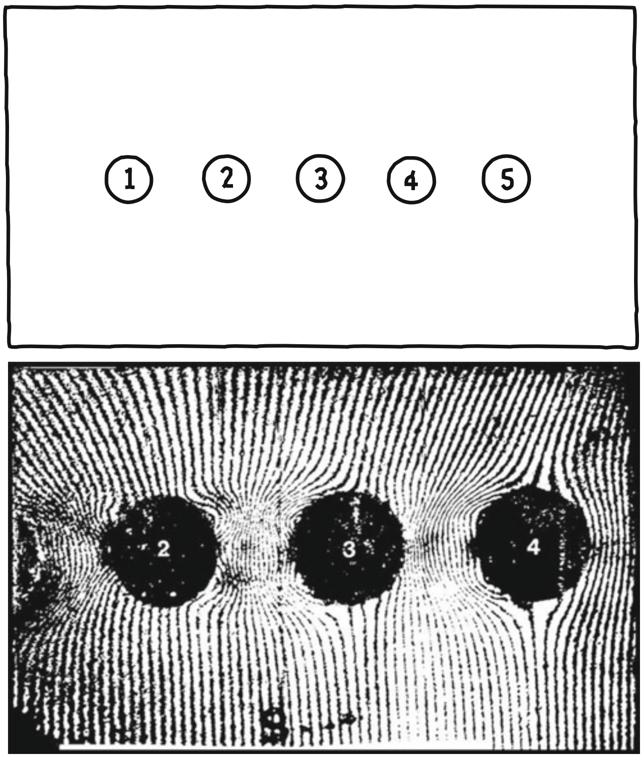


**Fig. 7** Top: A five-hole specimen used to investigate interactions of strain fields caused by coldworking successive holes in a linear array. Bottom: Reduced moiré pattern from part of a 4-hole linear array after all holes were coldworked in the order 1-2-4-3

were coldworked in order left to right, one of a number of array patterns and coldwork sequences that were examined [13–16]. The holes in this particular array are moderately close together (1.75 x hole diameter), so it might represent a worst-case example. This figure also presents a much-reduced moiré pattern for the region surrounding holes 2–4 of a 4-hole array in a different specimen after all holes were expanded in the order 1-2-4-3. Some fringes are lost in this reduction, but one need not be a moiré expert to perceive that the strain profiles between the holes differ widely from one another.

*Observations*

Figure 8 shows the variations of the residual strains created between the first few holes as coldworking progresses along the line of holes for the 5-hole specimen. Unfortunately, the hole sequence was swapped end-for-end in these graphs so the first hole is now on the right. Also, here, compression strain is positive, contrary to usual practice.

Consider first the left-hand graph of Fig. 8, which gives the radial strain profiles between holes 1 and 2. The dotted line shows that, when only the first hole is expanded, a compressive strain of about 4 % is created at that hole edge, but the residual strain falls off and changes sign to 4 % tension at the edge of the adjacent non-coldworked hole. The solid line shows that, when hole #2 is subsequently cold-expanded, the residual strain at the edge of hole #1 remains at about 4 % compression, but the radial strain at the edge of the second hole is now drastically amplified to a value of about 10 % compression.

Now, look at the right-hand graph of Fig. 8, which shows the strain histories between holes 2 and 3 as coldworking progresses. The dotted line is the result when only holes 1 and 2 have been coldworked. The profile is similar to that appearing in the left panel, but the maximum compressive strain near hole #2 is now twice as high at 8 %, and it oscillates and drops to about 4.5 % tension at the edge of non-coldworked hole #3. The solid line shows the strain profile after hole #3 has been coldworked. The residual strain reaches 6 % compression at about .005 in. from this hole, but, for some unknown reason, drops off to only about 1 % at the hole edge.

*Summary*

These results, and many others obtained from these experiments, demonstrate some characteristics that consistently arise when coldworking arrays of holes, namely:

- When coldworking one hole in an array, unfavorable large (e.g. 4 %) tensile residual strain and, likely, a matching tensile residual stress is produced at the edge of the adjacent non-coldworked hole.
- An implication is that if one hole in a “tight” array is coldworked, then all of them should be coldworked.
- Large and unpredictable variations of residual strain will likely arise between adjacent coldworked holes.
- The residual stress state near the holes is not consistent, nor is it as expected from tests on single holes.
- The interactions of the residual strain fields between cold-expanded fastener holes is very complex and depends on many parameters and process variables including hole spacing, type of array, and, especially, the sequence of coldworking.

## Hole Movement and Change of Shape

*The Question*

Fundamental mechanics knowledge suggests that coldworking a hole in an array will cause adjacent holes to permanently move, possibly causing partial interference fits, gaps between fastener and workpiece, and misalignment of holes in workpieces that are being fastened together. Hole motion and shape change have been observed in the field, and one is led to suspect that these effects modify the expected stress states in the workpieces and the fastener.

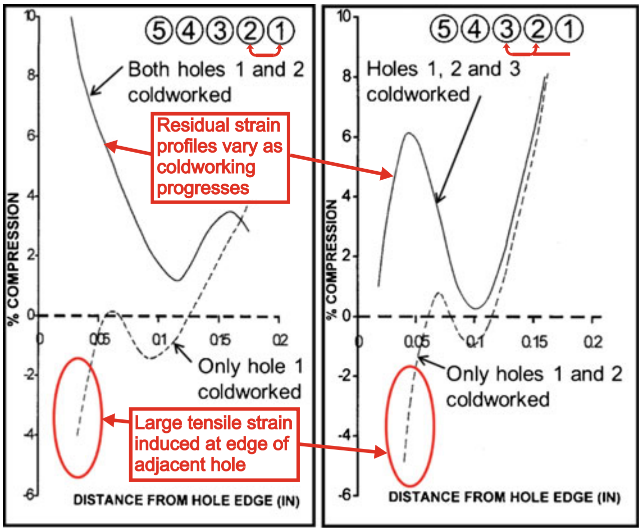


**Fig. 8** Sample of measured interactions between coldworked holes. *Left*: Residual strains between holes 1 and 2 in linear array after coldworking only hole no. 1 (*dashed line*) and after coldworking both holes 1 and 2 (*solid line*). *Right*: Residual strains between holes 2 and 3 after coldworking holes 1 and 2 (*dashed line*) and after coldworking holes 1, 2, and 3 (*solid line*)

*Experiment*

Optical measuring microscopy was utilized to measure the dimensions and shapes of the coldworked holes in various arrays [15, 16]. A sample result for the 5-hole in-line pattern of Fig. 7 is presented in Fig. 9. The shaded circles represent the original holes, and the outlines of the final holes are shown as a solid line. The hole boundary movement is magnified 5X for purposes of visualization.

*Observations*

The findings from these experiments are largely as would be visualized based on experience with material behavior. After hole #1 is coldworked, its boundary is

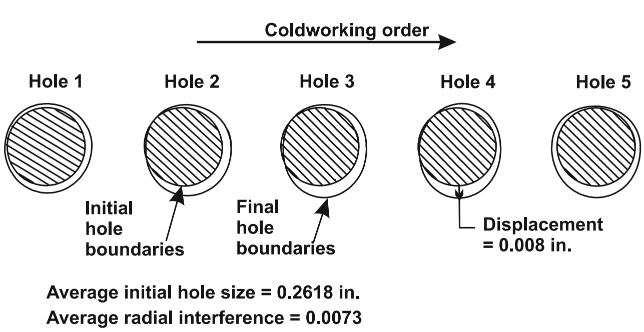


**Fig. 9** Hole movement and change of shape caused by coldworking all holes in a 5-hole linear array

nearly concentric with the original hole; but the act of expanding that hole appears to push the adjacent inter-hole material and hole #2 to the right. This process continues on down the line, and all the holes get pushed over except for the last one, which, like hole #1, ends up being distorted but not moved laterally, probably because the material to the right of that hole is solid and blocks movement. Further, if a hole is between two others, it is both moved and squashed laterally into an oval shape. The maximum boundary displacements are on the order of 0.008 in., which might not seem excessive until one realizes that it exceeds the coldworking radial interference of 0.0073 in. used in this example.

*Summary and implications*

- These experiments suggest that approximately 80 % of the expansion of any fastener hole occurs toward the adjacent unexpanded hole.
- All but the first hole coldworked in a linear array are moved and undergo a change of shape, becoming oval.

A serious implication is that hole movement caused by coldworking might cause misalignment of the holes in the workpieces being fastened together, thereby creating additional stresses if the fastening process forces the holes into alignment. Another implication is that a partial interference fit of the fastener might result and/or gaps could exist between the fastener and the workpieces. If the fastener were a bucked rivet, its shank would be deformed to accommodate the non-round holes that are likely misaligned. All of these effects depend on the spacing of the fasteners and the order in which the holes are coldworked.

Stress/Strain State in Close-by Plate Edge

*The Question*

In many structures, including aircraft, boats, and civil infrastructure, fasteners are necessarily placed near the edges of plates and shear panels. The question arises as to whether coldworking these fastener holes creates a problematic stress/strain state in the close-by edges of the workpieces. Of course, the result depends on many factors including the obvious ones of edge distance, hole spacing, and order of coldworking. Again, we present here some suggestive results from only one set of our experiments.

*Example Experiment*

As in the studies outlined above, moiré grating photography with optical processing was used to explore the residual strain states between a row of holes and a plate edge as coldworking progressed [13, 17]. Various coldworking orders were used. Figure 10 shows one such specimen, the coldworking specifications, and the quantities that were to be measured. The coldworking order for this 4-hole specimen was 1-2-4-3.

*Observations*

Figure 11 is a summary plot of the compressive strain profiles between each hole and the plate edge after all the holes in the specimen of Fig. 10 were expanded. The most striking features of these data are that the strain profiles vary widely and, for three of the holes, the maximum strains are remote from the hole edges where it is expected. Further, the last hole expanded, #3, has very low levels of residual strain in the region, being almost zero at the hole edge where it is needed. Possibly the top edge of the sleeve slipped below the specimen surface during the coldworking of this hole or something else went wrong. But, the strain fields do seem to be interacting in this region between the holes and the plate edge. An encouraging finding is that, even though the edge distance is fairly small, the compressive strain perpendicular to the edge falls to zero at some distance from the edge for all the holes.

The residual tensile strain profiles along the plate edge for three cases are presented in Fig. 12. Concentrate attention on the upper profile, which shows the final residual strain after all the holes are coldworked. While it oscillates somewhat, the strain in the plate edge is seen to be about 1 % tensile, meaning that a large tensile stress exists all along the edge, likely with some bulging of the edge in the regions adjacent to the fasteners. This finding elicits the specters of stress corrosion and/or fatigue crack initiation somewhere in the edge.

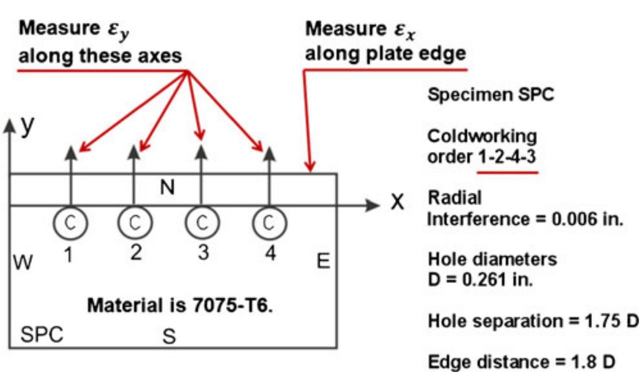


**Fig. 10** Four-hole specimen geometry, specifications, and quantities to be measured to determine effects of coldworking on close-by plate edges

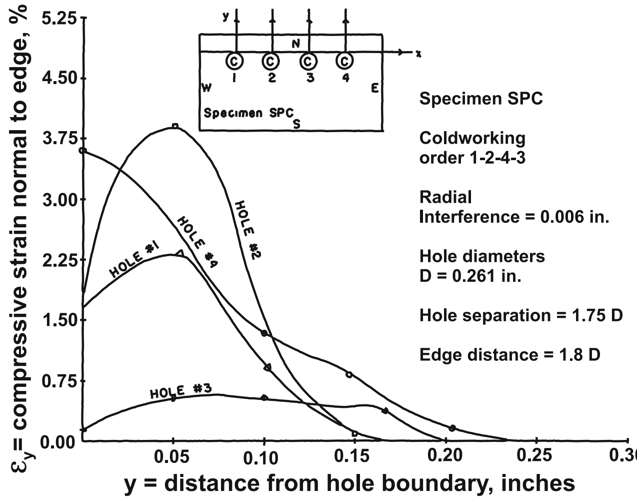


**Fig. 11** Compressive strain profiles between coldworked holes and near plate edge for 4-hole linear array

*Summary*

These experiments on the strain states caused in near plate edges by coldworking holes suggest that:

- Large variations in the residual compressive strain profiles exist between the hole and the plate edge.
- The maximum compressive strain often appears somewhere between the hole and the plate edge rather than at the hole edge, contrary to expectations and desired results.
- This experiment, the preceding one, and others conducted suggest that the edge becomes bulged slightly outward adjacent to the holes.

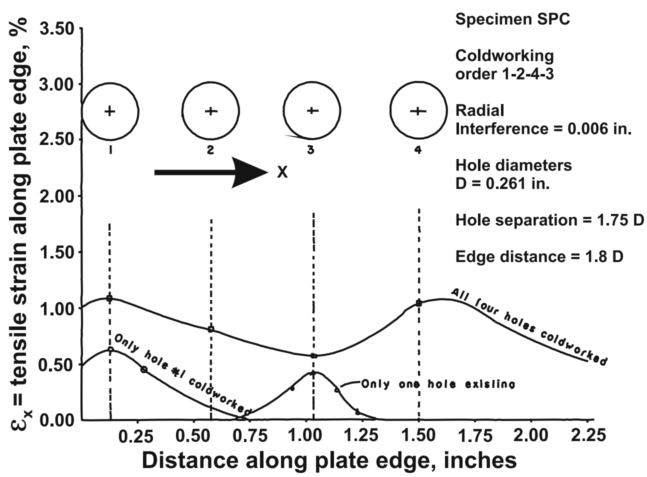


**Fig. 12** Tensile strains along the plate edge caused by coldworking the 4-hole linear array

**Fig. 13** Jet engine rotor used for an investigation of the effects of coldworking the mounting holes. Note the moiré gratings that were created in photoresist (*the yellow coating*) in the regions surrounding three of the holes

- Large residual tensile strain (e.g. 1 %) is induced along the edge of the plate, which creates an environment conducive to stress corrosion or fatigue cracking.

## Stress Concentrations Created by Coldworking

### *The Question*

That cold-expanding fastener holes can actually create significant stress concentrations is best demonstrated by an example drawn from our consulting practice. Figure 13 is a photograph of a jet engine turbine disc that is manufactured of a super alloy. The disc carries near its hub a circumferential array of holes through which pass the bolts or cap screws that bind a stack of discs and spacers together. Standard practice called for coldworking of these holes using the so-called "split-sleeve" process to reduce the probability of fatigue cracks originating at a hole boundary. Given anxieties about stress concentrations in the vicinity of the split in the sleeves, the shop technicians are instructed to align the split so that it points toward the center of the hub. The engine maker remained concerned about the stress concentrations and, if they existed, whether they were aligned according to specifications and whether they were serious enough to consider modifications of procedure.

### *Experiment*

To gain insight into the magnitude and location of the possible stress concentrations, the moiré technique was again used. Photoresist was applied to most of the web of the disc, appearing as the yellow stain in the photograph.

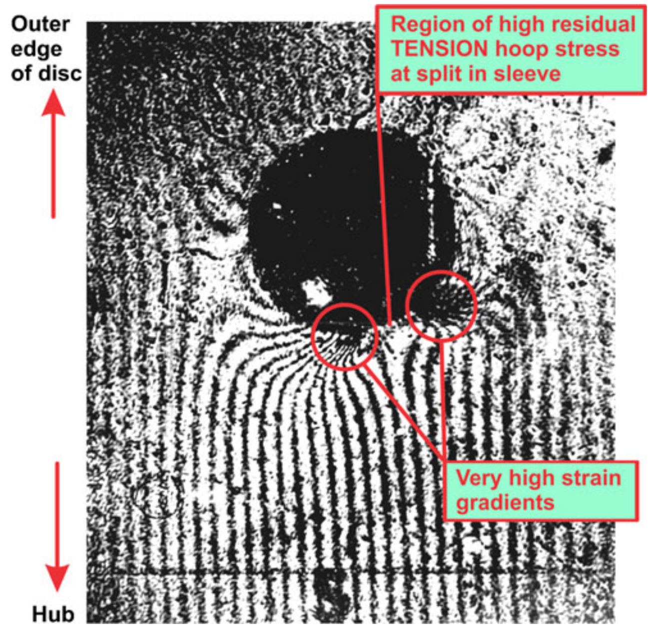


**Fig. 14** Sample moiré pattern from the turbine disc showing the high residual tensile strain and the large stress gradiants in the region of the gap in the sleeve used in the split-sleeve coldworking process

Gratings were created in the photoresist in regions surrounding three of the fastener holes in a non-coldworked disc. The grating orientation was set so as to measure the displacements in the circumferential direction in the regions of each hole boundary nearest and furthest from the hub. These gratings were photographed using the usual high-resolution techniques in order to establish a baseline moiré record for each hole. The disc was returned to the maker for coldworking of the holes using his standard technique. Subsequently, the gratings were photographed again using the same apparatus as was used to record the original gratings. The grating records were then superimposed with master gratings in the Fourier optical filtering system to obtain moiré patterns before and after coldworking for each hole.

### *Observations*

Figure 14 is a much-reduced typical moiré fringe pattern for one of the holes after cold-expansion. Given the difficulty of photographing the gratings on this reflective specimen that had suffered through two-way shipping and standard in-shop coldworking of the series of holes, the fringe pattern is somewhat noisy in spite of the spatial filtering that was done. Still, you need not be a moiré magus to conclude from merely qualitative observation that there exist potential problems arising from the coldworking procedure. The two areas where the fringes are concentrated at the hole boundary bracket the location of the split in the sleeve, and it was not aligned with the center of the hub. Now, the pitch mismatch used was chosen so that, compared with the fringe spacing in the far field, the fringes would come closer together in regions of compressive residual strain. The fringes along the hole boundary furthest from the hub suggest a nicely uniform compressive strain there. But, notice that the fringes are very tightly packed near the edges of the split in the sleeve, so the compressive strains there must be very high. We see also that the fringes in the region between the edges of the split are spaced further apart than they are in the far field, so this segment of the hole boundary must contain residual tensile strain and so possibly be susceptible to fatigue cracking. Finally, the change from large compression to significant tension occurs over a small segment of the hole boundary, so the strain gradient must be very large, a situation not conducive to long life. It is not known whether the manufacturer changed his coldworking process or his procedures.

### *Summary*

Coldworking mounting holes in a turbine disc using the split sleeve process resulted in:

- large residual tensile strain in the hole boundary at the gap of the slit in the sleeve,
- high strain gradients ranging from large compression to severe tension in the regions where the edges of the slit contacted the hole boundary,
- larger than nominal compressive strain concentration in the areas near the slit in the sleeve,
- stress concentrations that are not necessarily aligned with the axis of the hub as specified.

## Three-Dimensional Nature of Residual Strain Field

That the residual stresses/strains created by the usual process of drawing a mandrel through the hole vary through the thickness of the workpiece has long been suspected. We conducted a study to investigate this question using a novel multiple-embedded- grating moiré technique [11] on specimens of polycarbonate resin [18–21]. Space limitations prohibit discussion of these experiments, but we found that the strain state is indeed three-dimensional, with the interior radial strain being smaller than the surface values. Further, the coldworking created a tensile strain perpendicular to the workpiece near its surface where the mandrel enters. This transverse strain changed to compression in the interior, reached a maximum at about mid-plane, decreased toward the bottom, but then increased again near the mandrel exit side. The minimal radial strain in the interior and the presence of tensile transverse strain cause worry, so the three-dimensional aspects of the residual stress state created by coldworking needs considerable elucidation.

## Research Questions Connected With Cold-Expanded Holes

Only a very limited assortment of potentially problematic issues involving coldworked fastener holes have been studied, and the inquiries have not been exhaustive. Many challenging inter-related questions remain for investigators to answer, including:

- What is the change of residual **stress**, if any, in the zone surrounding the fastener hole after static and cyclic in-plane compression?
- Where is the elastic–plastic boundary after in-plane static and cyclic loading? Poolsuk and Sharpe [22] skillfully measured this boundary for coldworked holes and provided a basis for further investigations involving remote loading and interactions.
- What happens to the residual stress if in-plane static tension and/or fully reversed tension-compression cycles are imposed?
- Is coldworking worthwhile or even safe for structural elements that are exposed to in-plane compressive or cyclic loads?
- When hole and edge interactions are involved, what are the final residual stresses in the plastically deformed zone near the hole?
- Where is the elastic–plastic boundary after an array of holes has been coldworked?
- Is fastener performance compromised by hole movement and shape change caused by coldworking adjacent holes?
- What is the optimum array of fasteners?
- What is the optimum order of coldworking holes in an array?
- Do fatigue and stress-corrosion effects arise from the tensile stresses created at plate edges and in the boundary of non-coldworked holes in an array?
- Do the stress concentrations created by the split sleeve technique compromise performance?
- Can the three-dimensional nature of the strain field created by coldworking be adequately defined, particularly as to the effects of tensile transverse strains.
- Can these multi-step elasto-plastic cold-expansion processes be adequately modeled with verification/validation for cases involving in-plane loads, hole arrays, various coldworking sequences, near plate edges, and complex materials?
- What are the answers to these and similar questions for materials other than those studied, particularly composites and sandwich plates.

### Bolting and Washers for Thick Composites

This section offers first some orientation as to the general mission undertaken. Subsequently, the importance of bolted joining of so-called thick composite sections to one another and to sections of other materials is described briefly. The rationale for limiting studies to the single-lap joint is presented along with some discussion of the problems inherent in this type of joining. Then, we examine a limited array of experiments and numerical analyses that were designed to advance the fundamental understanding of mechanical joining of thick sections; to identify, refine and validate suitable measurement techniques; and to develop useful design protocols for these joints. Not all goals were achieved, and, as often happens, more questions arose than were answered. Some research gaps are summarized at the end of this segment.

#### The Mission and its Relevance

The general mission of this research has been to develop validated design protocols for vehicle structures made of thick composites for use in severe service environments. The objective is approached through: (1) expanded fundamental research that advances understanding of the basic mechanics of fastening; (2) function-specific studies for aircraft, ground vehicles, watercraft and subsets of these categories; and (3) the eventual development of design standards, especially for severe service environments.

Most composite structures, e.g. civilian transport and sport vehicles, are made of relatively thin sections. Heavy-duty vehicles, particularly those used in the military sector, must be made of much thicker laminates so as to be resistant to ordnance impact, blast, and fire as well as to tolerate heavy loads resulting from extreme usage in rough terrain or heavy seas. Composites are being introduced as the material of choice for some components of military vehicles, for example the workhorse HMMWV. A goal is to eventually fabricate entire load-carrying structural components and outer hulls of land, air, and water machines of combinations of composite materials. Armored vehicles and ships are included in this initiative, and some prototypes of armored combat platforms have been constructed and tested. Imagine going into armed conflict inside a plastic tank! Of course it is not that simple. Composites have been widely used in military aircraft, and the all-composite plane is supposed to be just around the corner. Beyond the military sector, composites are increasingly used in heavy-haul road and construction equipment. In all these applications, "lighter, safer, stronger" are the watch-words.

Some important hard-learned lessons and criteria must be remembered when introducing composite components to existing platforms or starting from scratch with a new all-composite design. These include:

**Fig. 15** Sketch of the single-lap bolted joint showing the secondary bending caused by axial load

- Simple substitution of a composite part for a metal one is generally not satisfactory.
- Using with composites a fastening scheme that is known to serve for metal components is not a good idea, and early failures will result if this lesson is ignored.
- For combat and remote site applications, quick field assembly and disassembly are important.
- Inspection, repair, and validation of repair are necessary for operation in severe environments, no matter the material used.
- Composites can be fabricated into large complex shapes, e.g. an entire monocoque or monolithic chassis, which reduces the number of joints but which imposes a new set of fastening criteria.
- The failure mechanisms of composites, particularly in blast and fatigue, are not yet well understood, and safety factors that include fastener design must be incorporated until we gain sufficient knowledge to take full advantage of the possibilities offered by these remarkable materials.

#### The Single-Lap Bolted Joint

The single-lap bolted joint, shown in the simple illustration of Fig. 15, is the first choice for perhaps the majority of joinings in structures and vehicles. On the one hand, this mechanical joining is the simplest possible. On the other hand, it is the most complex because, especially for joining thick workpieces, three-dimensional effects derived from secondary bending and tilting of the fastener create large stresses and amplify stress concentrations. In general, these three-dimensional problems are difficult of solution either analytically or experimentally. Inevitable joint slippage arising from bolt-to-hole clearance is a complicating factor. Further difficulties arise when complex materials, such as composites, are to be joined in this way. Curiously, relatively few published works about this seemingly simple joint are extant, possibly because of the forbidding difficulty of obtaining fundamental, validated, publishable results that would lead to design guidelines. We wryly observe, based on our own experience, that it is much easier to analyze or perform experiments upon symmetrical double-lap pin joints; and there exist many reputable papers on that subject.

Even though, from a mechanics viewpoint, the single-lap joint is complex, its mechanical simplicity makes it attractive for severe applications. It facilitates field assembly, disassembly, inspection, part substitution, and repair; and

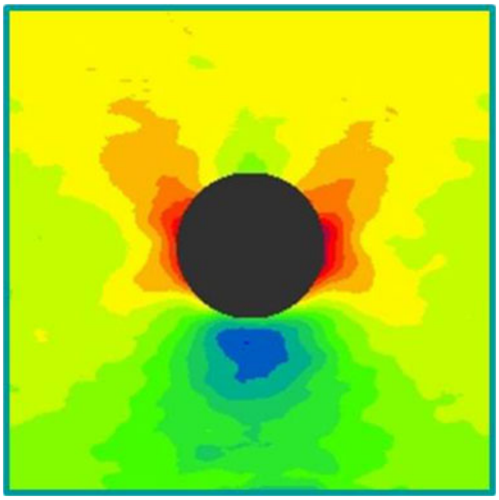

Fig. 16 Surface strain map obtained by digital speckle pattern interferometry for a pin-loaded double-lap joining of "thin" glass-epoxy composite

these characteristics alone might be sufficient to choose this type of fastening.

Given its practical importance and the relative dearth of useful relevant design information, the experiments described below focus solely on the single-lap bolted joint.

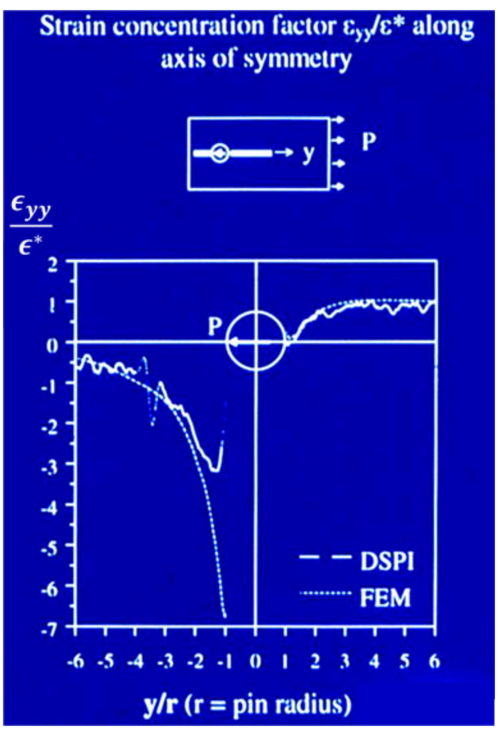


Fig. 17 Comparison of strain concentration factors obtained by digital speckle pattern inteferometry and finite element analysis for a pin-loaded double-lap joining of glass-epoxy composite

Validation/Verification Example

Before examining the single-lap joint, it seems worthwhile to emphasize through an example the importance of validation and verification of numerical and experimental analyses [23, 24]. Figure 16 shows a false-color whole-field surface strain map for a pin-loaded FGRP laminate as obtained via digital speckle pattern interferometry (DSPI). The strain concentration factor along the axis of symmetry was extracted from the measured strain values (not the fringe map), and these are plotted as the solid lines in Fig. 17. The values obtained from finite element analysis (FEA) are plotted as dotted lines in this figure. The agreement is excellent except right at the loaded hole boundary. The DSPI values near this boundary are likely contaminated by error resulting from filtering and/or taking the derivative of the displacement map. The important point is that, overall, the agreement is fine. Probably these independent studies are not both wrong, so confidence that both are correct seems appropriate.

## Single-Lap Bolting of Thick Isotropic Materials

### *Purpose*

Preliminary to the experimental and numerical analyses of single-lap bolted joints of composite and aluminum, we explored, developed, and refined suitable techniques on similar joinings of isotropic materials, in this case a birefringent plastic and aluminum [25–27]. The objective was to determine, with validation, the strain fields near the fastener, particularly in the load-bearing region, on the surface and in the interior of a single-lap bolted joint of "thick" plates of these materials for various clamping forces and applied loads. Although seen as a pilot study, the results were expected to be useful in themselves because of the previously mentioned shortfall of information about the behavior of such a joining.

### *Methodologies*

The techniques chosen for stress analysis of the bolted joining of thick isotropic materials were:

- embedded polariscope three-dimensional photoelasticity,
- surface and embedded resistance strain gages,
- finite element analysis using HyperMesh 5.0, ABAQUS 6.2 and ABAQUS/Viewer 6.3-1.

Embedded polariscope photoelasticity is one of the very few methods that can be used to examine the stress distribution inside an object, and it is quite easy to implement. The technique is well known [cf. 28], so it will not be described in any detail here even though it is, perhaps, on the verge of being forgotten. Its main disadvantages are that it yields the stresses in only one thin slice of the object; it gives directly only the


**Fig. 18** Early step in the fabrication of an embedded polariscope specimen for 3-D photoelasticity analysis of the stresses in the bearing region of a bolted single-lap joint of "thick" isotropic plates

maximum shear stress distribution; goniometric compensation is not possible because the polarizers are fixed; and, unless several specimens are made with linear polarizers, isoclinic data indicating principal directions are not to be obtained.

Figures 18 and 19 show how the specimen was made. It was machined from a plate of PSM-9 (CR-39) photoelastic resin having a thickness of 0.5 in.. The bolt hole was then rough-drilled. Next, a thin slice was carefully cut from the bearing section below the hole. Resistance strain gages were applied to the cut interior face of the slice. Then, the slice with its strain gages, along with properly oriented circular polarizing material (combined linear polarizer and quarter-wave plate), were glued into the slot from which it was originally cut using the matching cement. The saw kerf provides just enough space for the polarizers, glue, and strain gages. Finally, the hole was finish reamed to provide a smooth sliding fit for ground shoulder bolts having a shank diameter of 0.5 in. Strain gages were installed inside the bolts, converting them into transducers that were calibrated to provide accurate measurement of joint clamping force.

After appropriate curing and hooking-up of the strain gages, the specimen was attached to a similar aluminum plate using the shoulder bolt mentioned above. The bolted specimen was then placed into a dead-weight loading apparatus with a light source behind it and a lens and digital camera on the near side for viewing the isochromatic fringes for various levels of bolting torque and applied load. To maintain similitude to real-life applications, the load path was not arranged so that the load was artificially aligned with the plane of contact. Remember that, in this case, the observer is looking through the specimen from edge to edge.

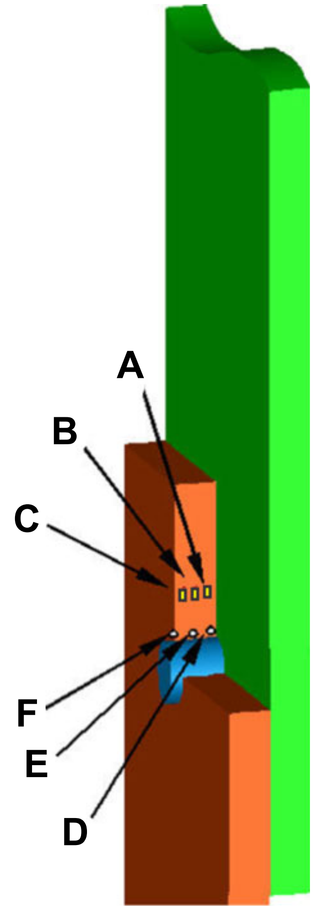


**Fig. 19** Cut-away schematic of the single-lap bolted joint specimen for combined 3-D photoelasticity, strain gage, and numerical investigation of stresses in the bearing plane of a bolted joining of "thick" plates. Points A, B, and C are the locations of the strain gages. Points A-F are where experimental results are compared with stresses from finite element analysis

*Sample Results*

Overall, good quantitative agreement between the results from the strain gages, photoelasticity, and finite element analysis was attained. Figure 20 shows a sample comparison of the photoelasticity fringe pattern and the maximum shear stress from FEA for one case of bolting torque and applied load. The small strain gages on the interior slice and their lead wires can be seen in the photoelasticity fringe photograph.

A serious problem arose during the testing, however. At higher loads, the cement binding the polarizers and the slice into the specimen failed, probably because of the high shear stresses at those surfaces in the bearing region.

Quantitative observations from this program are too extensive for inclusion here, but they may be found in the papers and the thesis cited above.

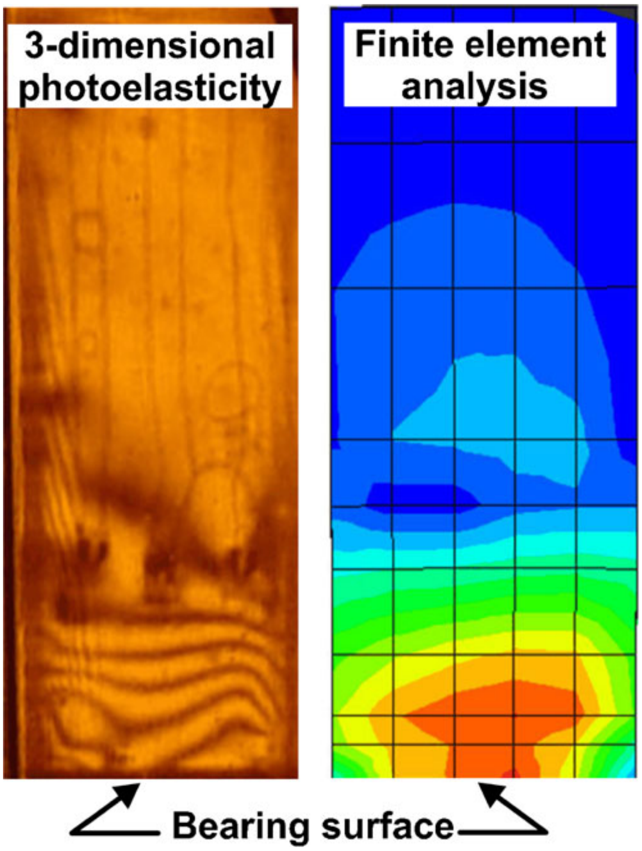


**Fig. 20** *Left panel*: Sample photograph of a photoelasticity pattern showing maximum shear stresses in the central bearing plane for the bolted single-lap joint. *Right panel*: False-color plot of the maximum shear stresses obtained from finite element analysis of the same joint under the same conditions as were used in the experiment

*Summary*

These studies suggested that embedded polariscope photoelasticity and embedded strain gages coupled with carefully implemented finite element analyses are dependable for investigating the three-dimensional stress distribution at critical areas in the region of a mechanical fastener. Subsequently, the FEA analysis was extended to explore the effects of joint modification, such as chamfering the corners of the holes in the inside faces of the workpieces [27]. This simple but uncommon procedure was found to improve the stress distributions by a significant factor, and it should be considered for critical applications pending experimental verification.

## Bolted Joining of Thick Composite to Aluminum

*Objectives*

The major goals of this project were to determine by independent experimental and analytical techniques the stress/strain distributions in the interior and on the surface of thick composite plates that were bolted by means of a single-lap joint to aluminum plates with varying degrees of bolt-to-hole clearance and subjected to various clamping forces and applied loads. If the FEA and Boundary Element Analysis (BEA) results could be validated, then these numerical models could be used to analyze an extensive matrix of cases with only a limited number of experimental checks being necessary. The eventual objective was to establish dependable design protocols for this type of joining [29–35].

*Methodologies*

As mentioned above, the experimental techniques and numerical analysis methods developed and proven for joinings of isotropic materials were intended to be used to study bolted joints of composites to metals. Reliance was to be placed on the transparent birefringent composite materials that were developed many years ago by Dally and Prabhakaran [36] and by Daniel, et. al. [37]. It turned out that the resins for these materials are no longer available. Thus, considerable effort was expended in developing a new recipe for a composite that could be use in photoelasticity experiments [38]. The main problem is matching the indexes of refraction of the glass fibers and a transparent resin while at the same time achieving appropriate mechanical and stress-birefringence properties. This initiative was eventually quite successful. However, the results obtained from photoelastic calibration tests and experiments on a simple model were disappointing because of creep and relaxation effects in the matrix resin. Clearly, more research is needed in order to develop a successful recipe, so photoelasticity was abandoned for the duration of the study.

Given the necessary adjournment of three-dimensional photoelasticity as the method of choice, several surface

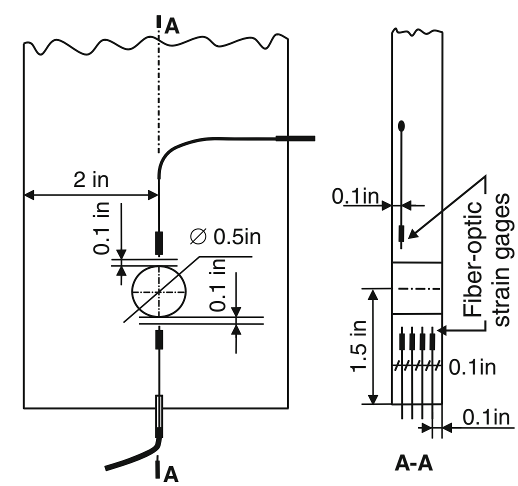


**Fig. 21** Drawing of specimens used for experimental analysis of single-lap bolted joining of thick composite to aluminum showing the locations of the embedded fiber optic strain gages

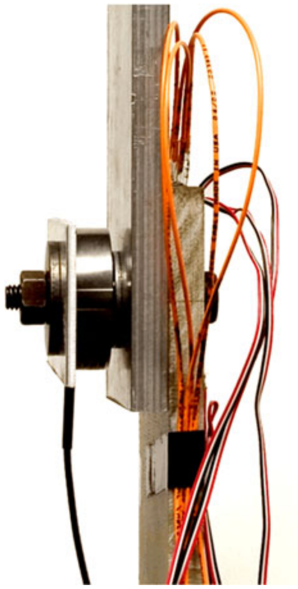

**Fig. 22** Thick composite-aluminum bolted joint specimen with embedded fiber optic strain gages, resistance strain gages, and bolt clamping load transducer ready for testing

resistance strain gages and embedded fiber optic strain gages were adopted in order to measure the strains at several critical locations. Adding only a few (e.g. 6) glass fibers to the FGRP composite should not modify the material properties enough to cause discernible error. To eliminate the scatter in the well known torque-clamping force relationship for bolted joints, an annular bolt transducer made specifically for the purpose was used to measure directly the joint clamping force.

Two specimens were fabricated by hand layup of 60 layers of plain weave E-glass plus epoxy resin to achieve a thickness of 0.5 in. with a fiber volume fraction of 0.35 [34]. For the second specimen, which is the only one that will be discussed here, five Bragg-grating-type fiber optic strain gages (FOSG's) were embedded during layup by attaching them to the appropriate glass cloth layers with care taken to assure that they did not move while compacting the assembly for curing. Still, keeping track of the locations of the sensing elements was expected to be a significant issue. They were tagged using tiny droplets of a resin mixed with metal powder at specific locations from each of the sensing elements so that their locations could be determined by x-ray if necessary. As it turned out, the tag and gage locations could be observed visually by backlighting the translucent specimen. Figure 21 shows a schematic of the specimen and its final FOSG locations. Following machining of the specimen to dimensions and drilling and reaming the hole, resistance strain gages were applied to the surface of one of the specimens. Figure 22 is a photograph of a finished specimen with the clamping-load transducer in place as it is ready to be loaded.

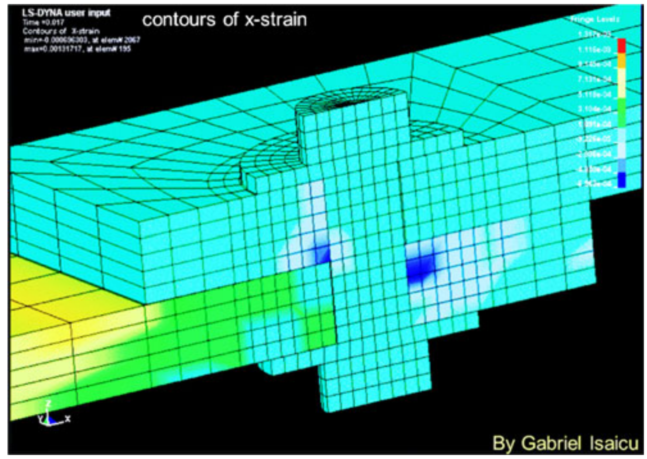


**Fig. 23** Map of longitudinal normal strain from finite element analysis of a bolted joint of thick composite to aluminum at load for one value of clamping force

Finite element analysis was accomplished using LS-DYNA, the method of choice for these problems that involve many contact surfaces, point contact, clearance, slippage, impact, and rigid body motion. The number of mesh nodes typically ran to about 5000 for analysis of these problems. Details of the modeling scheme are omitted from this presentation, which focuses on experimentation. But, understand that the details are important and care must be taken; this problem cannot be simply meshed and plugged into a solver. Some of the FEA results are in the form of "movies" that show the progression of selected stress or strain components in various views as the load is increased.

*Sample Observations*

Owing to the number of parameters involved, the results for even this one specimen are voluminous. We offer here only a nibble.

Figure 23 shows the color-coded FEA map of the longitudinal component of strain at a rather small maximum load for one level of clamping load, this being the last frame of a

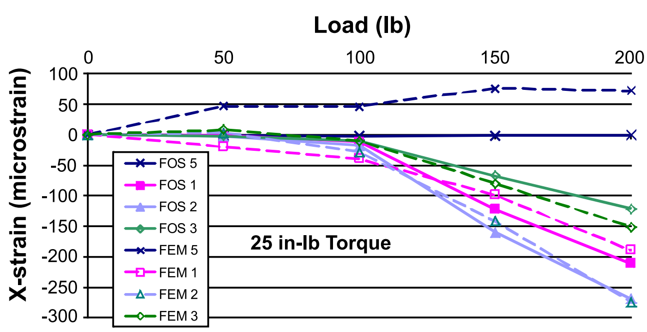


**Fig. 24** Comparison of the measured strains using embedded fiber optic strain gages (FOS) and predicted strains using finite element analysis (FEM) as a function of load for one value of bolting torque. Note that fiber optic gage no. 5 failed in this specimen

time-sequence. The composite plate is on the bottom in this view. Notice first the large (ca. 0.07 %) tensile strains that are created at the top surface of the composite remote from the fastener by the secondary bending inherent in this joint. Second, observe that, even though the bolt-to-hole clearance is small (smooth sliding fit), the tilting of the bolt causes very large (ca. -0.05 %) compressive strain near to but not coincident with the corner where the bolt contacts the composite. These effects are consistently found in analyses of the single-lap joint. The strains observed in the aluminum are, of course, much smaller than those found in the composite.

Figure 24 is a comparison of the measured strains and the FEA results from one specimen at one level of clamping torque and over a load range from 0 to 200 lb. The solid lines are from the FOSG's and the dashed lines show the values at the FOSG locations as obtained from the FEA. Gage locations correlate with the diagram of Fig. 21. The results from these two independent means agree satisfactorily except for gage #5, which was damaged during specimen handling, as was gage #4. Improved protection of the fiber leads where they exit the specimen has been implemented for subsequent studies.

An interesting feature of these results is that the strains remained very small up to an applied load of about 100 lb., at which point the joint began to slip and take up the clearance. At this point, the strains started to increase rapidly owing to contact between the bolt and the workpieces. An unexpected result is that both the experimental and numerical analyses showed that the highest strain occurred at the location of gage #2 rather than at gage #1, which is closer to the contacting corner. Figure 23 shows this happening in the composite but not in the aluminum, where the contact stresses are overall much smaller than they are in the composite. Unless the hole diameter was not uniform, a possible reason for this behavior, which has been consistently seen, is that the composite might slightly bulge as the clamping load is applied.

### *Torsional Strains in Composite*

Although the experiments are not described here, worth mentioning is that measurement of surface strain by moiré interferometry showed that high torsional strains are created when tightening a nut onto a laminated composite even when a washer is present [29, 39–41]. These strains are caused by the friction between the nut and the washer or laminate; and they would, evidently, cause very high inter-lamina shear stresses near the surface. The only way found to totally eliminate these strains was to use a thrust ball bearing between the nut and the washer, a practice certainly unacceptable in manufacturing.

### *Summary*

The results of both the numerical and experimental investigations are far-reaching, quantitative, and provocative. We must be content here with limited qualitative observations.

- A suitable recipe using currently available components for a birefringent transparent composite would be extremely useful for research on mechanical or adhesive fastening of composites using photoelasticity.
- Fiber-optic strain gages serve well for measuring internal strains at specific points in composites, and installation is reasonably easy during layup of the material.
- Instrumented bolts or a bolt load transducer must be used to accurately measure the clamping load; calculating the clamping force from the bolting torque is not acceptable for fundamental research.
- Validation and verification are especially important for these problems that are characterized by stress states and materials that are complex.
- Numerical analysis of problems of this type must be implemented with great care.
- Secondary bending in the single lap joint, when loaded as it would be in practical applications, creates large strains adjacent to the lap.
- Even with small clearance, tilting of the bolt causes large contact strains near the inside hole corner.
- The contact strain is maximum at a point on the hole boundary that is removed from the contact surface by roughly 20 % of the plate thickness.
- The strains in the region of the bolt remain small up to the point at which the joint slips to take up the clearance, after which the strains increase quickly at a rate that is roughly proportional to the additional load. This result underlines the importance of adequate clamping load to the performance of a bolted joint.
- Tightening a nut, with or without a washer, can create high torsional strains near the surface of the composite, with concomitant large inter-lamina shear stresses. This effect is minimized with difficulty.
- The fundamental mechanics of bolted joints involving composites, and particularly single-lap bolted joints, have not yet been sufficiently investigated to develop anything approaching adequate general design criteria.

## Impact Strain Caused by Joint Slip

### *The Question*

The studies summarized above suggested strongly that additional investigation should be undertaken in order to understand the limiting effects of clearance, clamping force, and slip.

### *Methodology*

A limited FEA study was undertaken using LS-Dyna. Experimental investigation of this problem has not yet been implemented.

### *Some Observations*

That clamping force is important and that joint slip causes high impact strains and stresses is suggested by the map of longitudinal strain shown in Fig. 25, which is the final frame of a sequence showing axial strains obtained through FEA for increasing load obtained by Isaicu [Isaicu GA, Cloud GL (manuscript not published) Clearance effects in single-lap bolted joints of thick composites: a finite element investigation]. The bottom workpiece is a composite and the upper one is aluminum. The hole clearance in this example is on the large side for purposes of illustration. The clamping force is, however, quite large. Up to the load at which slip occurred, the time-sequence shows that the typical bending strains remote from the joint increase to a large value with increased load, but the strains downstream of the bolt remain at low levels. At a certain limit load, the friction between the workpieces created by the clamping force is overcome, and the sudden slippage of the joint causes impact between the bolt and the hole boundaries at, apparently, substantial velocity so as to cause a very sudden increase of strain.

The time sequence was not extended enough to show it, but the short rise-time of the strain and the nature of the strain distribution downstream suggest quite strongly that the slippage and impact possibly create a stress wave that would travel to the end of the composite bar and be reflected.

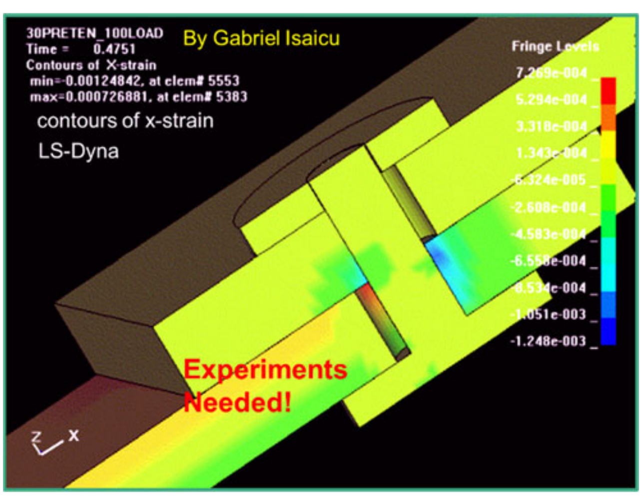


**Fig. 25** Map of longitudinal normal strain from finite element analysis for thick composite to aluminum bolted joint showing the effects of slip and impact resulting from large bolt-to-hole clearance

One can imagine what would happen to the workpieces if the load were rapidly oscillating so that this stick–slip behavior and the resultant impacts affect both sides of the hole. Further FEA and careful experimental investigations are badly needed to assess the worth of these speculations.

Notice that the highest dynamic stress is not at the corner of the hole, the same as was found for the case mentioned above. Further, examination of the strain in the hole boundary in the rear quadrant where it not in contact with the bolt shows that there are very large tensile axial strains in that region, especially near the interface between the composite and the aluminum. Whether this concentration is abetted by secondary bending migrating inside the region under the washer, by dynamic effects, or by the high stress concentration in the ligament region close to the hole is not known.

### *Summary*

This pilot investigation suggests, without independent confirmation, that a lap joint with significant bolt-to-hole clearance might likely exhibit the following behaviors.

- If the bolt fit is not uniformly tight, and especially if the clearance is large, joint slip and bolt tilting cause large impact strains near the inside corner of the bolt hole in the composite.
- This destructive effect can be overcome only by applying clamping load sufficient to prevent or at least drastically slow the slippage so that dynamic impact does not occur. Otherwise the joint must be designed so that the impact stresses can be tolerated.
- The impact seems to cause a stress wave that travels down the plate and back, but this suggestion must be confirmed.
- The destructive effects of impact would probably be amplified if the applied loads were to oscillate rapidly.
- Large axial tensile stress concentrations are predicted in the region surrounding part of the unloaded portion of the hole boundary, but these predictions must be verified.
- We speculate that the performance of a joint that is subject to impact can be improved by reducing the stiffness of the composite in the region of the bolt. This approach might require that material be removed as by drilling additional holes or by using an insert.
- Experimental research on the effects of clamping load, hole clearance, and the impact resulting from dynamic slip is lacking. This insufficiency is retarding the development of adequate design criteria.

## Fastener Arrays in Composites

We have investigated at some length the use of fastener arrays in FGRP materials [29, 42–44] Space limits do not permit discussion of these rather complex studies in this survey. Suffice it to say that simple transferal of the practice used with metals is not wise. Better to use a few carefully designed fasteners than a closely packed row or array. Additionally, the use of unloaded extra holes to reduce stiffness in the region affected by the bolt is suggested, an old practice that seems to have been largely forgotten.

## Washers for Composite Bolted Joints

Standard practice is to incorporate washers of various types into bolted joinings in order to distribute the clamping load over an area larger than that contacted by the bolt head or the nut, to prevent galling, and to limit tilting of the bolt. The fundamental mechanics of washers seem, however, to be poorly understood. They work, so why worry about them? But, do they always provide the improvements assumed, and, of the many types available, which type is best for a given application? Available information, which is actually extensive and usually dependable, is mostly derived from simple mechanical testing or from field trials and long experience. There is a tendency to think that what has worked for many years for joining of metals will suffice for composites.

Thorough investigation of the mechanics of washers in composite joints is a forbidding and, perhaps, a thankless undertaking. We have, however, investigated in a limited way two special problems of interest. The first involves the merits of conical washers relative to flat washers. The second examines the stress/strain field in the region directly underneath a flat washer.

## Conical Washers for Composite Joints

Conical washers (a.k.a. spring washers, Belleville® washers) are often used when it seems worthwhile to add compliance to a joint or to maintain clamping force and friction in situations where the joint geometry might change owing to creep or wear. One common example of such an application is when very stiff conicals are employed in mounting clutches or flywheels to the crankshafts of small engines. These washers have been proposed as a means to compensate for anticipated creep and relaxation in joints involving plastic or composite components.

### *The Question*

The fundamental concern is whether or not conical washers provide sufficient advantages over flat washers in composite joining to justify the extra cost imposed during fabrication of structures. A potential manufacturing problem is to assure that the washers are installed correctly on an assembly line, that is, the right way up.

### *Methodologies*

We investigated, using a highly sensitive 6-beam 3-axis moiré interferometry technique [11], the degrees of strain relief in the region of a fastener hole resulting from the use of flat and conical washers on FGRP composite [39, 40]. Rather on a whim, but guided by the attractive prospect of reducing delamination around the hole boundary, the conical washer in an upside-down orientation was also tested. Figure 26 shows the three configurations compared. In order to create the "conical washer up" setup, two conical washers were used, one in the normal position under the bolt head and the second in reversed orientation, as shown. The peak and average bearing strains, ligament strains, and shear strains in the region surrounding the hole were measured, as were the normal strains at 45°, at low and high loads for each fastener configuration after installation with both "low" and "high" bolting torque.

Some years after the experiments were conducted, FEA using LS-DYNA was implemented and used to study the strains and stresses of both the composite and the conical washer in its usual downward configuration.

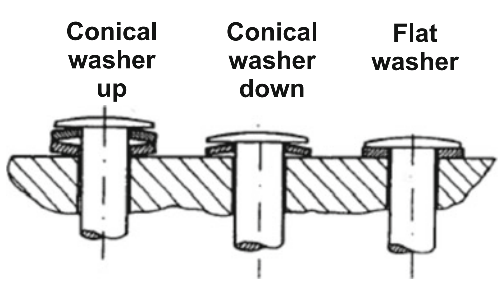


**Fig. 26** Flat and conical washer configurations investigated

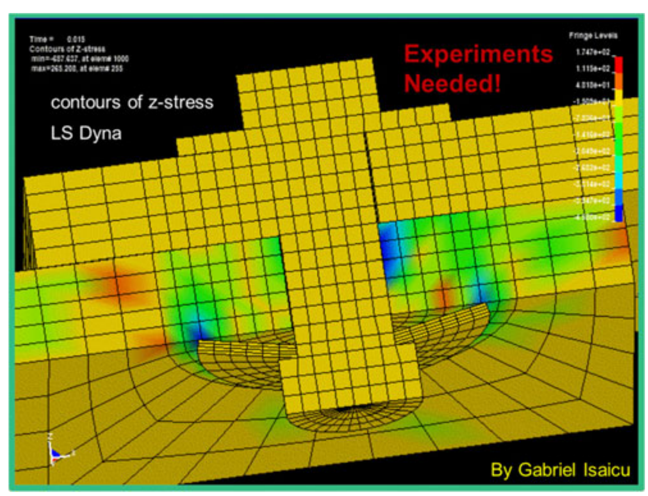


**Fig. 27** Map of transverse normal stress in a loaded thick composite to aluminum bolted joint that incorporated a conical washer in normal configuration

### *Some Observations*

Even though limited in scope, the lessons taught by these experiments and the FEA were extensive. We focus here only on the major findings that bear directly on the wisdom of using conical washers on composites.

Somewhat surprisingly, the experiments showed that the degree of strain relief provided by the various cases studied fell, with rare exceptions, into the following order of effectiveness:

- Best—Conical washer UP (reverse of the usual practice) with high bolting torque.
- Intermediate—Flat washer with high bolting torque.
- Worst—Conical washer DOWN (the usual practice) with high bolting torque.

The strain relief for each case with low bolting torque was inferior to that obtained with high clamping force.

Figure 27 is the last frame of a time-sequence generated by FEA for increasing load on a joint containing a conical washer in its normal downward configuration [Isaicu G (2007) unpublished communication]. This frame shows the contours of the z-component (perpendicular to the plates) of stress in the composite plate at maximum load. The stresses in the conical washer were also calculated, but they are not shown in this view. We see, again, high contact stress near the inside corner of the hole resulting from tilting of the bolt. Further, there appears, as expected, high compressive stress around the periphery of the washer where it contacts the outer surface of the composite. What is most surprising is that a region of large tensile z-stress appears in the region between the washer contact zone and the hole. This stress acts directly transverse to the matrix between the plies and could contribute to or even cause premature delamination. The experimental finding that the normal conical washer down configuration gave poor strain relief seems to be justified by the FEA results. These observations suggest that using conical washers on laminated composites should be approached with considerable care and maybe even some trepidation.

**Fig. 28** Composite crossmember designed for implementation in a line of large automobiles. Also shown are two of the end pieces that were used to test various schemes for attaching the crossmember

### *Summary*

- The use of conical washers on laminates might be counterproductive and should be implemented only after testing and analysis.
- If conical washers are to be used, it seems that they should be oriented in the reverse of the usual configuration, that is, with the base of the cone facing away from the laminate.
- An ordinary flat washer is better than a conical washer installed in its conventional downward orientation.
- In all cases, high bolting torque gives better strain relief than does low bolting torque.
- In its normal downward configuration, the conical washer creates high transverse tensile stress in the bearing region between the edge of the washer and the hole.
- The results suggest that a reversed conical washer with high bolting torque does, indeed, compress the region near the hole and reduces the transverse stress that might cause the lamina to separate, hence its apparent superiority to even a flat washer. Certainly, additional research on this problem is advised.

## An Example of Washers for Mounting a Composite Automobile Component

Difficulties that might arise when substituting composite components for existing metal parts are highlighted by this example. Here we have, possibly, a classic example where fastening issues were considered too late in a program. Figure 28 is a photograph of a composite cross member that was designed for installation in an automaker's "large" cars. It was made using a woven preform over a foam core, but the attachment points at its ends were solid. This item was to replace a very heavy and ugly component that was fabricated by bending and welding steel tubing. The composite replacement was strong, stiff, low in weight, quite easy to fabricate, and tidy, perhaps even pretty.

### *The Question*

How could the cross member be attached to the automobile substructure so that it is safely retained for many years at extremes of temperature, vibrations, dirt, moisture, and road chemicals. Of particular concern was maintenance of bolt clamping force under high ambient temperatures that might cause creep and relaxation.

### *Experiments*

Instrumented bolts were used to measure the clamping load over a broad range of temperature and time with various combinations of washers and bolting torques.

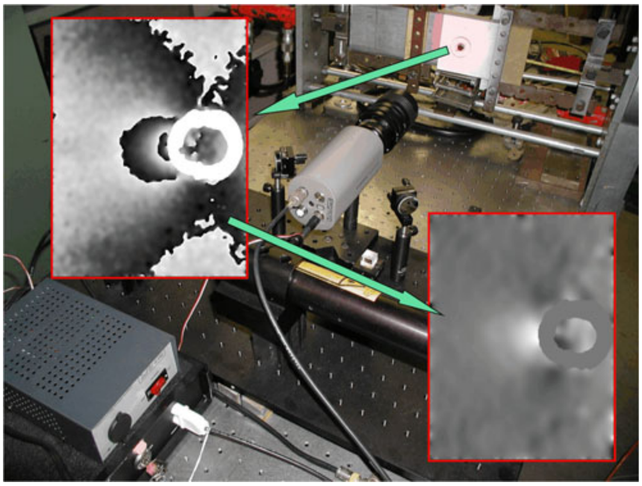

**Fig. 29** *Background*: Digital speckle pattern interferometer used with a transparent "washer-nut" to measure strains beneath washers at load for various bolting torque levels. *Top left inset*: A sample of the change of phase difference maps obtained in these experiments. *Bottom right inset*: A sample monochrome map of the longitudinal normal strain

*Recommendation*

It was learned and recommended to the automaker that conical washers should be used, with the best performance shown by the upside-down configuration of the washer.

*Result*

The recommended assembly solution was not acceptable in the shop for fear that the conical washers would not consistently be properly installed. For this and other reasons, this entire development program was terminated. A few years later, the manufacturer stopped production of the lines of cars in which this cross member was to be installed.

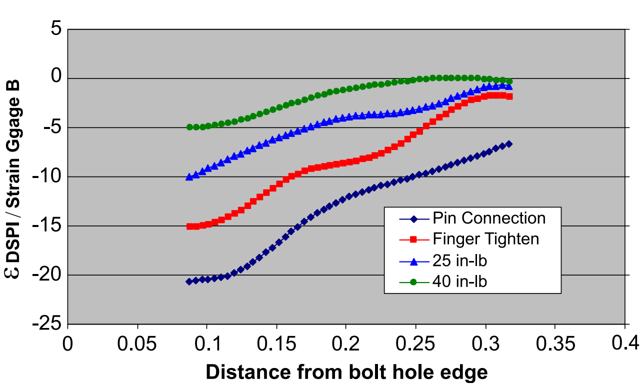


**Fig. 30** Normal strain profiles beneath a washer in a bolted joint at load for various bolting torques. The strains are normalized with respect to the far-field strain as obtained with resistance strain gages

## Strain Measurement Beneath Washers

*The Question*

We next discuss some progress on a problem that has troubled fastener researchers for many years. What is the surface strain distribution in the region where a washer contacts a workpiece in the loaded and unloaded states? Bolt clamping loads, friction, and joint slip preclude the use of most experimental techniques, including strain gages and pressure-sensitive films. Typical optical techniques cannot be used because the washer is opaque. Some creative thinking was needed to devise an approach.

*Methodologies*

The techniques that were adapted to this difficult measurement problem included the following [38, 45]:

- Use a transparent "washer-nut" made of polycarbonate resin. This device consists of a thick washer that incorporates a threaded hole so it acts both as a washer and a nut. The thickness of the washer-nut is determined, using plate theory, so that its stiffness is the same as that of a conventional steel washer.
- Apply the same clamping loads as would be used with a steel washer.
- Implement embedded digital speckle pattern interferometry (EDSPI) to measure the displacement and strain fields at the interface. The embedded speckle technique has been proven to work in such applications, and possible errors caused by refraction and birefringence in the materials have been analyzed and can be corrected [46–48].
- Utilize resistance strain gages immediately adjacent to the washer and in the far field to obtain data for checking and for normalization of the measured strains beneath the washer.

*Observations*

For this limited feasibility study a quite ordinary phase-stepping DSPI setup, shown in the background of Fig. 29, was assembled and used with home-grown software [47, 49], to measure the displacements in the direction of the load applied to the single-lap joint of aluminum-to-composite by means of a hydraulic load frame. Observations were taken for pin-connection (no clamping load) and finger-tightened cases, then for clamping torques of 25 in-lb. and for 40 in-lb., all of these being considerably smaller than the clamping loads that would likely be used in practice.

Presented in the upper-left inset of Fig. 29 is one example of the EDSPI phase change maps that were obtained. From these were extracted the strain maps, of which one is shown in the lower-right inset. These maps look rather messy and useless to the uninitiated, but, of course, they are used only for visualization. The strain data are extracted from the numerical arrays stored in the processing computer.

The graph of Fig. 30 presents the normalized strain profiles beneath the washer-nut for the various bolting torques as a function of distance from the edge of the bolt hole. Normalization is with respect to the far-field strain as measured with the strain gages. The data appear to make sense. The pin connection allows the highest strain concentration, as expected, and the other cases fall into line, with the highest clamping load offering the best strain regime.

*Summary*

- The washer-nut concept worked. It could probably be improved by modifying its cross section so as to account for the stiffness variation of the combination of a nut and a washer.
- Embedded digital speckle pattern interferometry is useful for obtaining strains at a contact surface.
- Quantitative results were obtained and they correlated well with strain gage measurements located beyond the washer.
- So far, only one configuration of flat washer and bolt has been investigated.
- Further exploration of the stress/strain fields beneath all kinds of washers is needed to advance understanding of clamping, slippage, and failure mechanics.

## Research Gaps Connected with Mechanical Joining of Thick Composites

As is usually the case, research on mechanical fastening of thick composites has generated more questions than have been answered. Appearing below is a lengthy but by no means complete list of research topics and questions that await attention by skillful and determined experimentalists and analysts.

- Development of thoroughly adequate design criteria for bolted joints of all types in both composites and ordinary materials awaits advancements in understanding of the fundamental mechanics of such joinings through experiment.
- A suitable recipe for a birefringent transparent composite that can be used for fastening research using plane and three-dimensional photoelasticity is badly needed to augment resistance and fiber-optic strain gages as well as conventional mechanical testing.
- Extensive improvement and validation of numerical models is in order so that parametric studies can be performed with confidence.
- Deep exploration of the stress/strain fields beneath all kinds of washers is needed to advance understanding of clamping, slippage, and failure mechanics so that washer design options may be optimized.
- Does a conical washer installed in its normal configuration actually generate a transverse tensile stress in the bearing region as predicted by numerical analysis?
- Additional research is required to decide if a reversed (upside-down) conical washer with high bolting torque is, indeed, preferable to other washer configurations.
- What are the dangers resulting from the interlamina torsional shear stresses caused by tightening a nut onto a composite? How can this effect be eliminated?
- Experimental research on the effects of clamping load, hole clearance, and the impact resulting from dynamic slip is lacking but is badly needed for design purposes.
- Experiments are required to measure the stresses in the ligament region of the hole boundary that is not contacted by the bolt in order to support or disprove existing analytical models.
- For thick composites, is the region of highest contact stress actually displaced from the hole corner as predicted by models?
- Does simple hole shaping, as by chamfering the inside corner of the hole, actually enhance strength, as models predict?
- Limited analytical research has shown that non-round fastener holes improve stress distributions in composites that are stiff and somewhat brittle, and enlightened exploitation of this idea should yield significant benefits.
- Could non-cylindrical fasteners be designed to improve joint performance, as some research has suggested?
- Interference-fit fasteners are known to improve joint performance in ductile materials, and they also have been examined in a limited way for composites. This approach should be more fully explored.
- Cold expansion of certain composites and sandwich materials has also been explored to a limited extent, and this technology merits further investigation.
- What are the optimum fastener arrays for composites?
- Can the performance of a mechanical fastening be improved by "softening" stiff composites in the region of a fastener as by the well known practice of removing material?
- What is the response of a bolted joint in composite to external stress waves caused by blast or impact, and can the response be improved by creating a mechanical impedance mismatch in the region of the fastener?
- Given the ability to create complex composite structures, what is the potential for reducing the number of fasteners in a given application through careful design?

- What degradations of joint performance are caused by creep and relaxation of the composite matrix, and how can these effects be minimized?
- What factors support crack initiation and propagation at composite joints under both static and cyclic loading, and how can these problems be minimized?
- What are the best failure criteria for mechanical joinings involving composites?

## Fastener Inserts and Hybrid Joining for Composites and Other Materials

This segment introduces the concept of using a bushing, insert, or sleeve, and possibly introducing some adhesive to advantageously modify the stress state in the region of a mechanical fastener. Subsequently, we describe some experiments suggesting that drastic reductions of stress concentrations or, at least, worthy trade-offs can be achieved through the introduction of an insert between a pin or bolt and the surrounding material. Finally, a novel approach to creating an insert that fills the bolt hole clearance gap and, if desired, creates an adhesive patch is presented along with some test results that demonstrate the performance improvements that can be attained even with the most basic form of this concept.

### Inserts, Bushings, and Hybrid Joining

Provision of a bushing or sleeve between a bolt or pin and the hole boundary to reduce wear and redistribute stresses is a very old practice. These devices have been used, for example, in harnesses and hitching gear of draft animals (e.g. the whiffle tree), water-powered mills, and the riggings of sailing ships. Likely, the function of such bushings at the mechanics level was not understood, but they were known to work, especially when attaching together load-carrying members made of the composite that we call wood. Bushings and inserts have also been used to good advantage in fastening brittle materials such as cast iron and concrete.

A question that always arises when discussing the use of inserts and bushings is, “Would you not achieve the same performance if you just forget the insert and use a larger bolt?” The answer is, in short, “No.” The reason is that the bushing or insert deforms slightly so as to redistribute loads and, so, reduces stress concentrations. At least one major manufacturer of airliners has used tight bushings in major structural connections. The company’s engineers had learned, and I quote, “A bolt with a bushing is always better than a large bolt alone.”

Given known stress channeling effects, anisotropy, tendency to delaminate, and ofttimes brittle behavior observed in composites, including wood, one is led to the idea that inserts might offer a good way to enhance the performance of mechanical joinings that involve these materials.

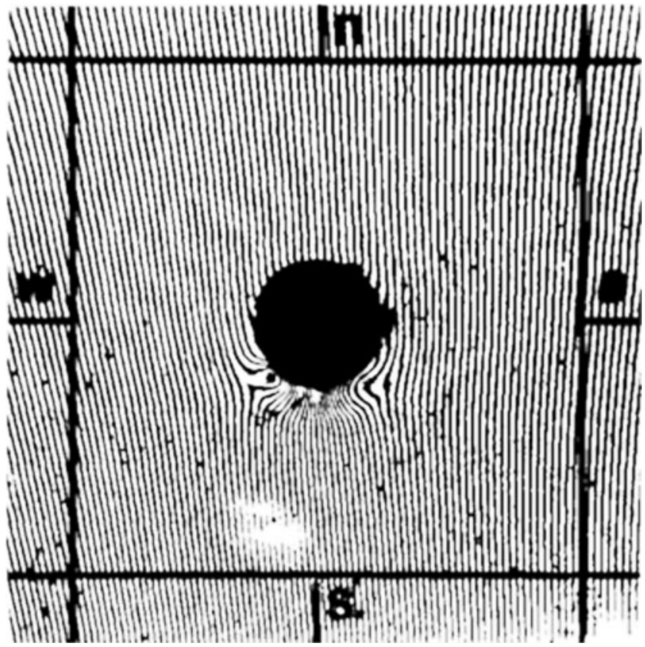

Fig. 31 Sample moiré interferometry fringe pattern showing the transverse displacements in a pin-loaded double-lap joint of orthotropic composite with a plastic insert

Another technique for improving joint function is to combine mechanical fasteners and adhesives. If you have worked with wood, you have probably used glue with either nails or screws to good effect when joining cabinetry. While often described as the “chicken rivet” or the “belt and braces” idea, this hybrid approach offers possibly the best combination of

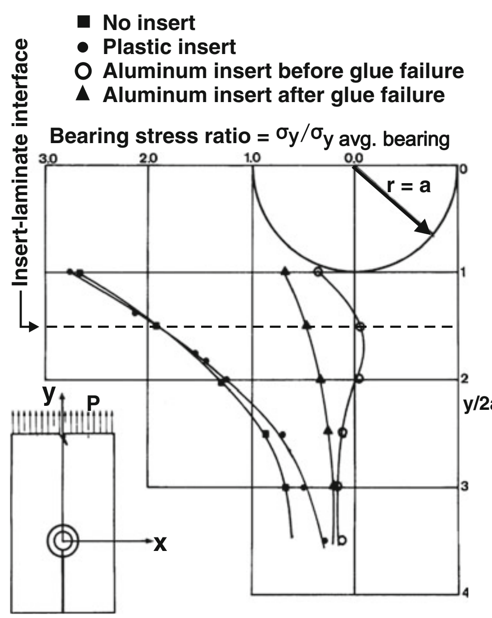


Fig. 32 Stress concentration factors in the bearing plane for double-lap pin-loaded joints of orthotropic composite having no insert, a plastic insert, a bonded aluminum insert, and an unbonded (after glue failure) aluminum insert. Normalization is relative to average bearing stress

optimum stress distribution, best fatigue life, ease of installation, reduced weight, and ultimate safety, particularly for composites. Hybrid mechanical and adhesive fastening certainly deserves considerable attention for critical applications.

The following section summarizes some experiments that were conducted to assess the potential of inserts for alleviating stress concentrations in joints of composite material.

Inserts for Reducing Stress Concentrations in Composites

*The Questions*

Two questions were asked, namely:

- Can the stress concentrations created by a loaded fastener be reduced by the use of inserts that surround the fastener shank?
- What should be the stiffness of the insert material?

*Methodologies*

In this study, double-lap pin joints of relatively thin isotropic epoxy material with a "soft" insert and of an orthotropic glass-epoxy with both "hard" and "soft" inserts were tested in a basic hydraulically actuated loading frame, with the load measured by means of a strain gage force transducer [29, 40, 41, 50]. Specimens of both materials with no inserts were tested to provide baseline data. Two types of inserts that accidently evolved into three types were implemented. The first was made by filling the bolt-to-hole clearance gap between the pin and the hole with PC-10C photoelastic epoxy resin. The second type was fabricated by machining an aluminum bushing so that it was a snug fit to both the hole boundary and the pin. This bushing insert was glued into the hole using the epoxy mentioned above. During testing at high load, the glue failed, so that it could subsequently be tested as a specimen with an "unbonded hard insert."

Measurements were implemented using high-sensitivity 6-beam 3-axis moiré interferometry with a grating frequency of 60,000 lines/in [11, 51]. Data acquisition and reduction were always done in differential mode so as to eliminate the effects of optical aberrations and irregularities in the gratings. Pitch mismatch was used to enhance data processing. This technique yields full-field displacements and subsequently strains at 0°, 90°, and 45°, so the strain rosette equations can be employed to determine principal strains and maximum shear strains in the entire region of view. The moiré fringe data were digitized and processed to obtain displacement and strain maps using a well tested computer program [3, 51].

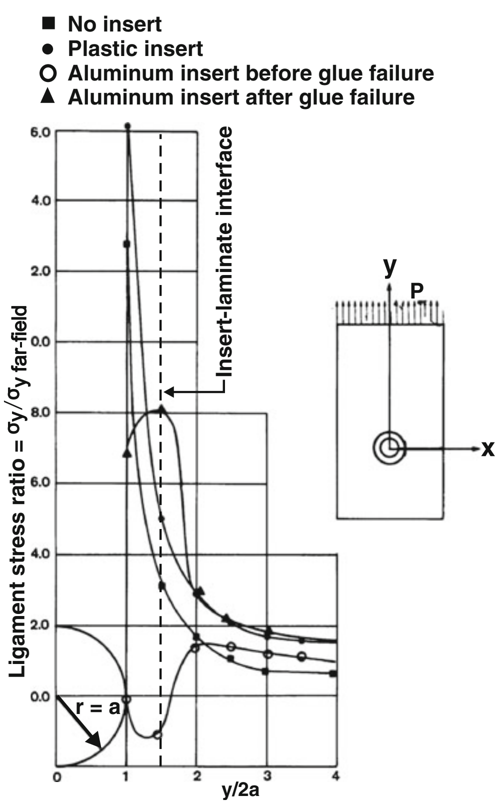


**Fig. 33** Stress concentration factors in the ligament region for double-lap pin-loaded joints of orthotropic composite having no insert, a plastic insert, a bonded aluminum insert, and an unbonded (after glue failure) aluminum insert. Normalization is relative to far-field normal stress

*Sample Observations and Results*

Figure 31 is a sample of the moiré interferometry fringe patterns obtained, this one showing the displacements perpendicular to the load axis for the orthotropic glass-epoxy specimen with the "soft" plastic insert. Some of the densely packed fringe detail in the load bearing area is lost in this reduction. The boundary of the insert can barely be discerned by the ripples appearing in the fringes in some parts of its circumference. The fastener diameter was held constant throughout these experiments. For the cases where no insert was used, the hole was finish reamed to provide a smooth sliding fit with the pin. The hole was made larger to accommodate the insert when it was used, with, again, a smooth sliding fit provided for the pin.

The bearing stress concentration factors in the orthotropic composite for all four insert cases are plotted in Fig. 32. The measured bearing stress is normalized with respect to the average value, calculated as the load divided by the projected bearing area (hole diameter times plate thickness). Distance from the hole is normalized with respect to hole radius. We see that the plastic insert does not have much effect; its stress concentration approaches 3 at the hole edge, roughly the same as when no insert is used. Very striking is the bearing stress concentration for the bonded aluminum insert; at the hole edge it is reduced to about 0.3. The unbonded aluminum insert factor is twice as large, but still small at only 0.6. Clearly, the use of a stiff insert reduces significantly the bearing stress concentration factor in this material.

Figure 33 presents the ligament stress concentration factors in the orthotropic material for the four insert situations. As is usual, it is normalized with respect to the far-field average tensile stress. Notice that this factor is very large at the hole edge when no insert is used, reaching a value of 16. This large stress concentration is characteristic for materials of this type, a fact that is often not realized in design. Also, it settles to a factor of 2 rather far from the hole, apparently not reaching unity. With the soft plastic insert in place, the stress concentration at the hole edge is reduced to about 13, and it falls quickly to about 1 further afield. Again, very worthy of attention is the reduction of the stress concentration and the shape of its variation seen with the bonded aluminum insert. The ligament stress concentration factor is zero (nil stress) at the hole edge, and it surprisingly reaches a negative value of −1.0 a short distance away within the boundary of the insert. Further away, it increases to roughly +1.5. Finally, the stress concentration for the unbonded aluminum insert is 7 at the hole edge, rises to 8 at the insert-composite interface, then settles to about 1.25 remote from the hole.

*Summary and implications*

- These results demonstrate that bonded and unbonded inserts can drastically reduce stress concentrations created by loaded mechanical fasteners and/or reshape their stress profiles in ways that can be used to advantage.
- Using inserts with mechanical fastenings in composites and other materials seems to be highly desirable for tailoring stress concentrations and improving performance.
- A properly designed insert is not merely a gap-filler; it is an important structural component.
- These gains might be especially valuable in applications involving blast, impact, or fatigue, or where sealing is crucial.
- We speculate that the dynamic performance of a joint can be adjusted to advantage, possibly by creating a mechanical impedance mismatch with a soft insert or bushing.
- Best practice might call for the use of fewer fasteners, carefully engineered, that incorporate washers and inserts.

*The Acceptance Problem*

Although inserts have long been known to improve fastener performance, that they are not widely accepted in engineering practice is likely because of perceived cost and manufacturing issues. The machining and fitting of the typical insert is expensive in time and labor. Each fastening situation requires a specific insert design. If the insert is to be bonded into place, then the surfaces must be cleaned, the insert must be coated with adhesive, it must be carefully installed, and the adhesive must be cured before component assembly and installation of the bolt or pin. Successful implementation of fastener inserts in the manufacturing arena requires simplification of installation procedures and reduction of costs.

## Hybrid Mechanical Joining System

*The Question and Objectives*

How can we reap and further cultivate the benefits of inserts and bushings while minimizing the factors that block their adoption? Creative thinking about this question defines the goals and a suggests a systematic solution. The general objectives are to develop a unified fastening system that can be adapted to a broad range of applications, that offers many mix-and-match options, and that provides distinct performance improvements over conventional fastening practice. The desired characteristics are summarized in the following non-exclusive list:

- Insert, sleeve, and bushing materials that can be tailored to specific applications, with many mix-match options, including:
  - "soft" insert,
  - particulates or fibers added to insert to tailor properties,
  - partial adhesive bonding (hybrid joining),
  - composite sleeve or bushing,
  - bonded or unbonded "hard" sleeve,
  - fixed length or self-adjusting "hard" bushing,
- Simple to install,
- Tolerance and repair of misalignments and crude drilling practice,
- Can repair damaged connections in field or shop,
- Improvement of static and dynamic (impact, blast) strength,
- Improvement of fatigue performance,
- Potential for reducing the number of fasteners in a given application,
- Sealing of hole edges against moisture and contaminants,
- Compatibility with ordinary fabrication processes,
- Field assembly and disassembly possible if needed,
- Simple application tools for use in shop and field,
- Not costly in material or labor,
- Adaptable to many different types of fasteners (e.g. bolt, self-drilling/tapping cap screw, blind rivet, bucked rivet, backed or unbacked, spectrum of materials).

*A Systems Solution*

The goals summarized above led to the development of a novel hybrid mechanical joining system [52]. The basic concept is to install a fastener (pin, rivet, bolt, or screw) so that it is roughly centered inside its hole, which is made oversize. A loose sleeve or bushing may or may not be installed with the fastener depending on the application and the results sought. Then, the insert material, typically a polymer augmented with additives to give it desirable mechanical, chemical, and electrical properties, is pumped into the cavity between the fastener and the hole boundary to bind the components together, eliminate bolt-to-hole clearance, seal the hole, and create a monolithic fastening structure. The system includes simple tools and methods for installation. If disassembly is expected to be necessary, then release agents can be applied to the hole boundary and/or the fastener shank and/or a special extraction tool can be used to remove the insert.

These fasteners are especially advantageous for composite materials, including wood and other natural materials that tend to split and delaminate, but they should also improve the performance of joinings involving conventional construction materials.

That the insert so created is a structural component must be emphasized. It is not merely an adhesive, although adhesive properties might be engineered into the system. Nor is it merely a gap filler. Material choices are key to the tailorability and effectiveness of this system.

Approximately 30 mechanical variants of the fastening system have been described, and many others are obvious extensions of the art. If the named material options are considered, then the number of variants increases to several hundred. In the following sections, only the basic concepts and a few of the variations are described, with many details omitted. Then, some validation experiments are described and the results are presented.

*The Basic Concept*

Figure 34 shows the fundamental ideas of the injected insert in its simplest form and using a hex-head bolt as the exemplar. Here, we have two workpieces that are to be bolted together. The fastener holes might be quite roughly made, of different sizes, and are shown to be misaligned. The head of the bolt and the nut each carry a boss whose function is to loosely center the assembly in the hole. Alternatively, this task might be performed by properly sized washers that are attached to the bolt head and the nut or held in place by other means. A small passageway is provided inside the bolt, originating at either its head or the bottom of the shank and ending at one or more points on the periphery of the shank. The origin of the passage is shaped to mate with a tool that injects the insert material through the internal passage so as to fill the clearance cavity. At least one small bleed groove is provided on the contacting faces of the bolt head and the nut to allow the escape of air and to assure complete filling of the cavity.

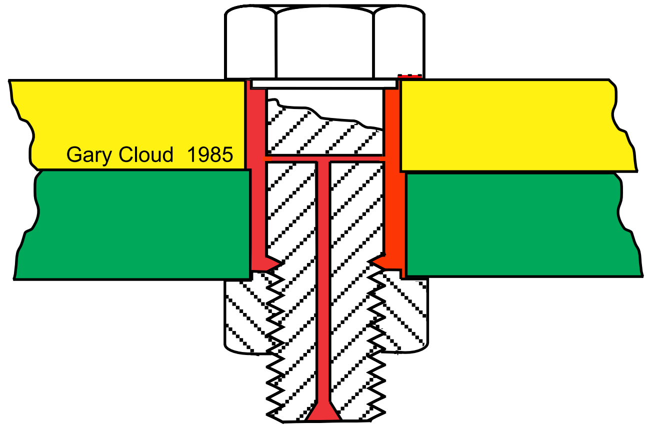


**Fig. 34** The basic concept of a mechanical joint with an injected insert in its simplest form

Although the provision of a small hole through the bolt does not significantly weaken it, instances where it would not be allowed can be envisioned. In some cases, the insert material can be injected through an off-center hole in the bolt head or the nut or through a small passage drilled beside the bolt through one of the workpieces.

*The Encapsulated Insert*

If injection of the insert through a channel in either the bolt or the workpieces is not acceptable, then the insert material can be contained in a capsule that initially embraces the shank of the bolt. Such an arrangement in its preassembly form is illustrated in Fig. 35. The capsule is merely slipped over the shank of the bolt, and it splits open as the nut is tightened down. It might be necessary to provide a sharp projection on the bolt head or the nut to help break the capsule and assist in mixing its contents. The capsule envelope collapses and remains within the insert, possibly adding strength if the envelope material is properly specified. As with the injected insert, the encapsulated one is compatible with various forms of hard and soft sleeves or bushings.

*A Hybrid Joining*

Figure 36 teaches one example of how the injected insert concept can be modified to create a bonded-in mechanical fastener with an adhesive patch surrounding it. In this case, the workpieces are either scarfed or counterbored to create a gap bordering the hole. As the insert material, in this case probably an adhesive resin with additives, is pumped in, it

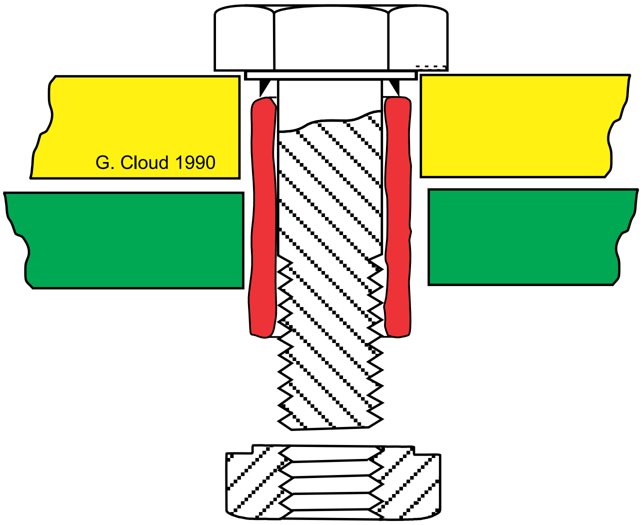


**Fig. 35** Mechanical fastener with encapsulated insert: expanded view before installation

fills the gap between the workpieces as well as the clearance gap and binds all together.

*An Insert with a Composite Sleeve*

As mentioned, the injected insert is easily enhanced with many types of fixed-length or self-adjusting sleeves or bushings that can be made of various metals, plastics, or composites. A particularly interesting example is demonstrated in Fig. 37. In this instance, a woven tube of, say, glass or graphite fiber is slipped over the bolt shank prior to assembly. The nut is tightened as usual. Upon injection of the insert material, the tube becomes saturated and also is forced to expand so as to contact the hole boundary. As the insert hardens, the entire assembly becomes a strong three-dimensional monolithic structural component that binds the workpieces together and also prevents their separation or delamination.

*A Self-Tapping Screw with an Injected Insert*

Figure 38 shows one way to create an insert with a self-drilling self-tapping screw with or without a sleeve (not shown). In this case, the shank of the screw is made smaller than the minor diameter of the screw threads to create a cavity sufficient to take advantage of the insert material. The insert resin with or without additives is injected through the screw head. Again, a bleed slot is provided beneath the screw head to assure complete filling of the cavity. The version shown incorporates a backing plate for added strength, but that is not a requirement. A significant advantage of using the injected insert with self-tapping screws is that the insert material completely fills the screw threads that have been carved out during installation, thereby greatly reducing stress concentrations as well as eliminating clearance. As before, if provision of a passage through the screw or one of the workpieces is not acceptable, then an encapsulated insert can be implemented or else an alternated injection path can be provided.

*Experiments*

Specimens were fabricated and tested to failure to compare the performance of ordinary bolted joints with those carrying the most basic form of the injected insert. Twelve pairs of composite plates were cut from 12 mm thick plates made of SC-15 toughened epoxy plus woven glass mat. The plates were joined using ordinary grade 8 nominal 1/2 in. bolts and matching flat washers. Six of the specimens had no insert, and the bolts were not drilled. Bolts with drilled 5 mm diameter passageways for resin injection were used for the remaining six specimens. All the bolts were torqued to the

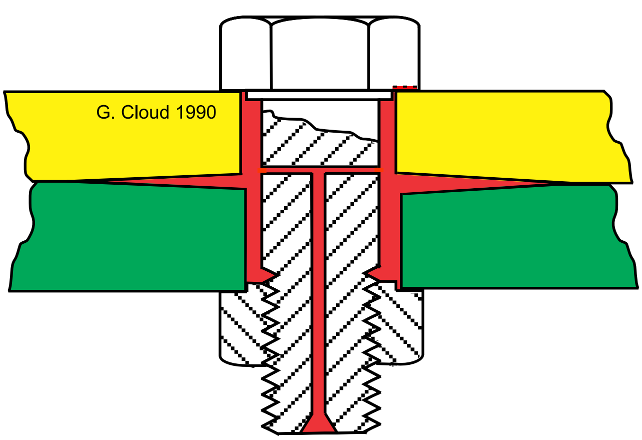


**Fig. 36** Hybrid mechanical fastening system with injected insert and adhesive patch implemented via scarfing, spacing, or counterboring the workpieces

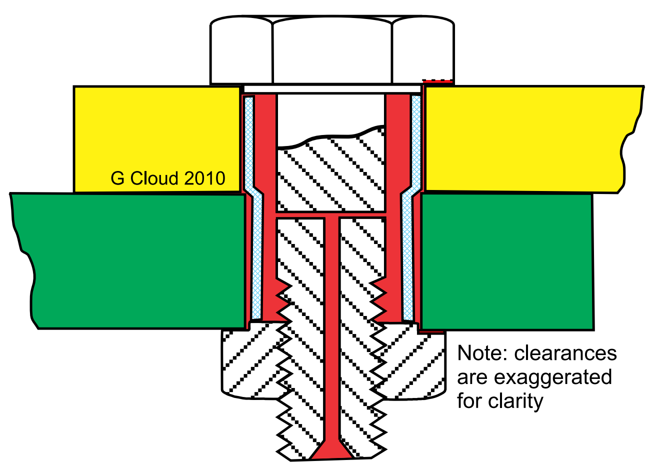


**Fig. 37** Mechanical fastener system with a tubular composite sleeve that is impregnated and expanded as the insert material is injected

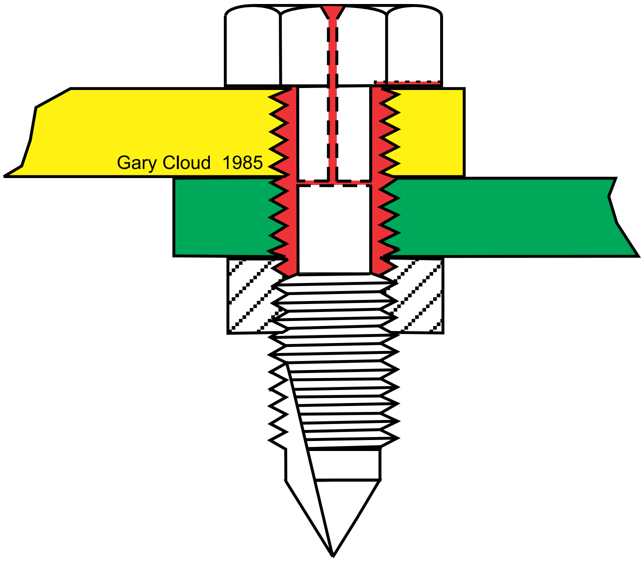


Fig. 38 Self-drilling self-tapping screw joining with injected insert

same level. Following assembly, type SC-15 resin was injected through the six drilled bolts to create the inserts, with care taken to insure that the clearance gap was filled. No bushings were used, and no particulates or fibers were added to the insert resin, so these inserts qualify as “soft.”

The specimens were pulled to failure in tension in a standard hydraulic testing machine. A laser extensometer was used to measure the relative displacement between the two joined plates, and photographs were recorded during the tests.

### Test Observations

Figure 39 shows plots of load versus relative displacement for one of the joints lacking an insert and for one of the specimens incorporating the simple injected insert. Some critical points in these graphs are highlighted to correlate with the key provided.

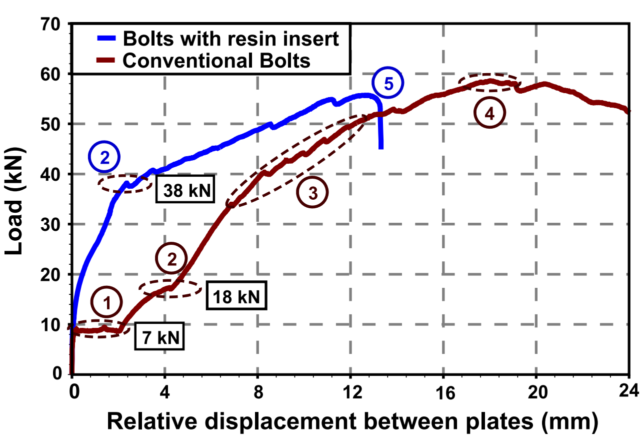


Fig. 39 Comparison of load–displacement behaviors of a conventional bolted joint and a bolted joint incorporating a basic injected insert of neat resin. Tests conducted by Dr. Mahmoodul Haq

Consider first the graph for the conventional bolted joint, shown as a brown line. We find that significant joint slip occurs at about 7 kN load, where the applied load is sufficient to overcome the friction caused by the clamping force. At 18 kN load, delamination begins. The delamination grows, the washer deforms, and the bolt bends as the load increases up to a maximum load of about 58 kN, after which the joint can be considered to be destroyed.

Now, examine the load–displacement plot, shown in blue, for the joint having only the basic injected resin insert. This joint shows zero slip, and it remains stiff up to the point where delamination begins at 38 kN. Beyond that load, the joint loses stiffness, but the load-deflection relationship remains essentially linear up to a load of 56 kN, at which point the bolt breaks.

Comparison of these data suggest that joint performance is positively modified if even the simplest “soft” injected insert is used. The degree of improvement depends on what one is looking for. If slip is critical, then the joint with the insert is infinitely better than the conventional one. If failure is defined as the onset of delamination, then the joint with the insert is 210 % better. If joint stiffness is the criterion, then the insert should be used. If slow catastrophic failure is preferred over sudden breakage, then the conventional joint might be seen as the better option.

Subsequent testing with other resins and/or a composite sleeve shows significant improvement in all categories, including failure mode, over the simple case reported here. These results will be reported soon.

That the failure modes of the two types of joints differ drastically is illustrated by comparison of the photographs appearing in Fig. 40. We see that, for the conventional joint, there is readily apparent relative displacement that implies

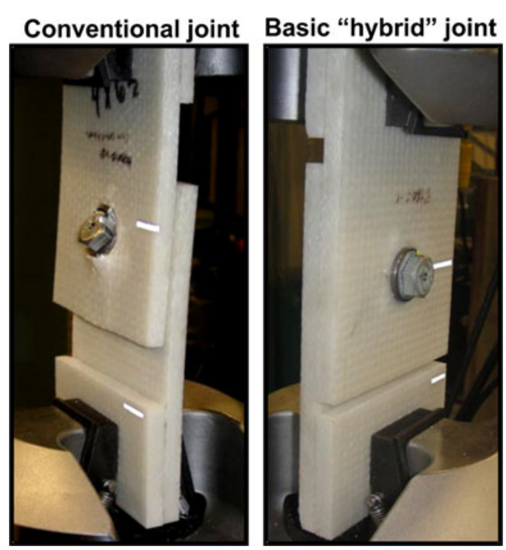


Fig. 40 Comparison of failure modes of a conventional bolted joint and a bolted joint incorporating a basic injected insert of neat resin. Tests and photograph courtesy of Dr. Mahmoodul Haq

low stiffness, the bolt tilts and bends, the area of delamination grows large, the washer deforms and begins to pull through the composite plate, and the plates separate as clamping force is completely lost. For the joint with the basic insert, the relative displacement is too small to be assessed by eye, the washer remains intact, the bolt does not tilt or bend, the delamination patch is small, and clamping force is evidently retained until the bolt breaks.

*Summary*

The following tentative conclusions can be drawn from the research conducted so far on hybrid joinings using injected inserts:

- A functioning insert that is a structural component is easily created by injection after installation of a bolt.
- The insert fills the clearance gap between the bolt shank and the workpieces, so precision reaming and alignment of the holes are not required.
- Because clearance is reduced to zero, joint slip under load is eliminated, a feature that cannot be realized in conventional manufacturing.
- If failure is defined by the onset of delamination, joints with even "soft" injected inserts outperform conventional bolting by a factor of about 2.
- A joint with an insert is much stiffer overall than one without an insert.
- A joining of composite plates with a bolt plus a soft insert remains essentially intact until the bolt breaks, whereas a bolted joining lacking an insert ultimately fails by bolt and washer deformation plus pull-through of the fastener.

Research and Development Gaps Concerning Inserts and Hybrid Joining

Although we have demonstrated that the novel approach described above to incorporate inserts and bushings into joinings offers significant benefits while minimizing penalties, the technology is still in the early stages of development. Some of the questions that remain to be answered are offered below with the understanding that they are related to the research questions listed at the close of the section on bolting and washers for thick composites:

- What are the fundamental mechanics that govern the behavior of fastener inserts and hybrid joining?
- What combinations of sleeve, bushing, and insert materials are best suited for specific applications?
- What is the optimum size of the insert and/or sleeve?
- What is the optimum adhesive patch for the hybrid system?
- What are the correct failure criteria and what are the optimum failure modes?
- Can suitable experimentally validated numerical models of joinings with inserts be developed as an aid to the designer?
- What are the responses of various forms of insert-protected joints to external stress waves as caused by blast or impact?
- What are the best manufacturing parameters, including cost?

## Concluding Remarks

This written version of the 2012 Murray Lecture, although condensed from the oral presentation, is of apparent necessity rather lengthy. An extensive field had to be plowed and sown in order to reap any sense at all of a complex topic. We hope that the product is interesting and useful, particularly to those who, perhaps in the early stages of a career in the rewarding but often under-appreciated field of experimental mechanics, might be on the hunt for research topics that are both challenging and potentially rewarding at many levels.

The paper has looked into some broad categories of subjects related to mechanical fastening; examined many experiments as well as some areas where experiments are needed to support analyses; summarized methodologies, observations, and lessons gained from those experiments; and presented lengthy lists of research gaps and problems that require attention to bring to the art of fastening design a proper scientific foundation. Certainly, additional topics can be extracted from this paper and the extant literature on fastening; but we have gained, one hopes, some momentum.

There is no need to re-summarize here the results and specific research ideas that have already been offered. Let us settle for only the broad lessons taught us, namely:

- The demands for lighter, safer, stronger structures and machines require advances in fastening at many levels.
- Solutions to a broad array of relevant problems remain incomplete or nonexistent.
- Answers do not come easily owing to the inherent complexities of fastening and the materials involved.
- Thoughtful and well-informed experimental work with a broad range of techniques is necessary for exploration and validation.
- Clever and well-grounded modeling and analyses are necessary.
- When working with composites, experience suggests that we remember cast iron.

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