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Structures Technical Memorandum 409

STRESSES AND STRAINS IN A PLATE RESULTING FROM TWO
CLOSELY-SPACED INTERFERENCE-FIT FASTENERS

by

R.P. CAREY

and

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SUMMARY

Electric resistance gauges have been used to investigate the strains induced around a pair of closely-spaced interference-fit fasteners (nominal centre distance equalling two hole diameters) in a D6ac steel plate. For comparison, a finite element analysis was performed for the same geometry using the penalty element method.

Measured strains showed broad agreement with the results of the finite element analysis and the latter indicated that the peak stress level in the plate at the hole circumference would be 1.14 times the level for a single similar interference-fit fastener in a infinite plate.



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NOTATION

E	Modulus of elasticity
G	Shear modulus
q	Stress non-dimensionalising factor = $\frac{4G\lambda}{\kappa+1}$
K_{pe}	Penalty element stiffness matrix
f_{pe}	Penalty element force vector
R	Hole radius
x,y	Cartesian co-ordinate system relative to origin at fastener centre
α	Penalty number
$\epsilon_{p\theta}, \epsilon_{pr}$	Circumferential, radial strains in plate
ν	Poisson's ratio
δ	Fastener radial interference
λ	Fastener interference ratio = δ/R
κ	= $(3-\nu)/(1+\nu)$ for plane stress = $3-4\nu$ for plane strain

.../contd

NOTATION (CONT'D)

$\sigma_{p\theta}, \sigma_{pr}$ Circumferential, radial stresses in plate

$\sigma_{f\theta}, \sigma_{fr}$ Circumferential, radial stresses in fastener

θ Angular position (c.f. Fig. 1)

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1. INTRODUCTION

The widespread use of various fatigue enhancement processes, such as the cold-working of bolt holes and the use of interference-fit fasteners has led to many studies on the benefits available and the mechanisms by which the gains are achieved. Even so, a recent literature survey¹ has found that current capabilities in respect to stress analysis around life enhanced holes is insufficient for the reliable prediction of the effects of such processes on fatigue life.

In order to further the knowledge of stress/strain fields associated with one such life-enhancement system, studies are being pursued at the Aeronautical Research Laboratories (ARL) on tapered interference-fit fasteners. These studies particularly relate to various fastener/parent plate configurations. Topics already reported include the elastic stress fields induced by a fastener located centrally in a large plate^{2,3}, and a fastener near an edge of a plate^{3,4,5}. In this paper we consider the elastic stress fields induced by a pair of fasteners situated close to one another; by experimental methods (using strain gauges) and numerically by the penalty finite element method which has previously been found effective for similar problems³. The general notation adopted in this paper is covered by the Notation section and Fig. 1.

2. EXPERIMENTAL PROGRAM

2.1 Test Specimens

Two specimens were used. Both were made from D6ac steel heat-treated to an ultimate tensile stress in the range 1510 to 1650 MPa. The stress-strain characteristics of the material of specimen F are shown in Fig. 2. The form of the specimens is illustrated in Figs 3 and 4 - basically Specimen F was a simple square plate, while Specimen 8/9 was of 'dogbone' shape. Both specimens contained a pair of tapered holes centred nominally two diameters apart. The holes were finish-reamed with an 18 flute tungsten carbide reamer and fitted with tapered bolts of the Taper-Lok type, Code 2TLHC2-6, having a nominal diameter of 3/8 inch (9.525mm). The bolts were manufactured from H-11 steel, and finished by nickel-cadmium plating and coating with cetyl alcohol. The taper of one in 48 allowed accurate increments of interference to be calculated from measured increments in insertion distance.

Short gauge length electric resistance strain gauges were attached near the holes on one face of the specimens and also on the sides of the Specimen 8/9. The gauge types are given in Table 1. Their locations are shown in Fig. 3 (Specimen F) and Fig. 4 (Specimen 8/9). With the exception of those gauges located on the specimen sides, the gauges were bonded to the faces from which the bolts were inserted in the holes.

Although there were differences in geometry the specimens were considered to be essentially equivalent with respect to elastic behaviour close to the fasteners. The key ratio of hole centre distance to hole diameter was virtually the same in both specimens (1.99 for Specimen F, and 1.97 for Specimen 8/9, measured at the fastener insertion face). The lug ends of Specimen 8/9 were sufficiently distant to have little influence, whilst the differences in the hole-centre-to-specimen-edge distance were considered to be of secondary importance and that any effects would not be significant. The ratio of hole-centre-to-edge distance to hole diameter was 2.9 in Specimen F and 2.6 in Specimen 8/9.

2.2 Test Procedures

Each tapered bolt was inserted with firm thumb pressure while one side of the specimen was clamped in a vice. Interference was increased in steps of approximately 0.01mm on diameter by tightening the fastener nuts. Corresponding strains and insertion distances were measured - the latter by a depth micrometer from displacements of the fastener head. Tables 2 and 3 give the values recorded for each specimen. The procedure was repeated until an interference on diameter of 0.137mm was attained, but data collected only within the elastic regime are presented here.

2.3 Experimental Results

The measured (elastic) strains were plotted against fastener insertion distance (Figs 5 to 8), and straight lines fitted using regression analysis. Some of the lowest strain readings were excluded from the analysis because of inconsistencies arising, possibly, from hole imperfections.

The slopes of the fitted lines represent strains per unit insertion distance. These are readily converted to strain per unit (dimensionless) interference, which are listed in Table 4. They are also plotted, in Figs 9 and 10, against non-dimensional distance from a fastener centre for two radial traverses - perpendicular to and parallel with the axis joining the fastener centres.

2.4 Estimation of Yield Incipience

As the initial stage of pin insertion often induces a non-linear response in the specimen, strain level rather than insertion distance was used to determine the upper limit of the elastic regime, i.e. yield incipience. For this purpose the strain gauge nearest to each hole was adopted as the control gauge. The strain corresponding to yield incipience was computed from a finite element analysis at the

relevant gauge positions, and used to define the upper limits of insertion corresponding to fully elastic behaviour at each hole.

In more detail, the peak stress levels for a known interference level were obtained from finite element analyses for both plane stress and plane strain conditions. When these stress levels were considered in relation to a yield stress of 1100 MPa (estimated from Fig. 2) it was found that yield onset (according to the Tresca criterion) could be expected at a threshold non-dimensional interference level of 0.00474. Strain distributions from the finite element analyses (see Figs 9 and 10) were then utilized to estimate the critical strain at the control gauges.

2.5 Error Levels

Strain measurement in similar tests indicated⁴ that the largest error was less than 2.8%. An approximate comparison with the present case indicates the maximum error levels should be no greater.

The measurement of fastener insertion by depth micrometer would usually be accurate to within approximately ± 0.02 mm. However, the pattern of data from Specimen 8/9 (Figs 6,7, and 8) suggests larger errors - possibly resulting from irregularity in the thickness of the strain gauge waterproofing material covering the surface of the specimen which was used as a reference face for the depth micrometer measurements.

3. NUMERICAL INVESTIGATION

Finite element analyses were carried out to verify the experimental results for Specimen 8/9 detailed in Section 2. For this work the specimen was modelled exactly; except for the regions around the lug ends, which are remote from the fastener holes. The specimen material properties were taken as $E = 209$ GPa, and $\nu = 0.3$.

3.1 Modelling Details

In performing the analysis it was necessary to deal with a problem common to interference cases, i.e. the values of displacement and forces around the fastener/hole interface are unknown. A method for overcoming this difficulty has been formulated and implemented in a previous investigation³ by applying the penalty finite element method^{6,7}. The analysis given in Reference 3 allows the required constraint condition around the fastener/hole interface to be enforced in a finite element solution - the constraint condition being that the sum of the radial displacements of the fastener and hole corresponds to the interference level; this condition is enforced for

all nodal points around the interface. To implement the method very large external penalty forces are applied at interface nodes and correspondingly large stiffness values for the penalty elements added to the standard finite element formulation. The penalty forces f_{pe} and stiffness matrices K_{pe} were determined using equations (3.6) and (3.7) respectively of Reference 3 and weighted over the circumferential length of the standard iso-parametric elements using a three-point Simpson's rule. The value of the penalty number α used in the equations was 7500E, which was chosen since it was previously found to be effective in Reference 3. The suitability of this value was verified for the present analysis, since convergence to the required constraint condition was achieved to within 0.001%.

Because of symmetry of Specimen 8/9 it was sufficient to model one quarter of the structure. The resultant finite element mesh (Figs 11 and 12) consisted of 129 eight noded iso-parametric quadrilateral elements, 36 six noded iso-parametric triangular elements and 25 two noded penalty elements.

It was assumed that there was no slip along the fastener/hole interface and hence adjacent nodes on the fastener and hole were restrained to have no displacement relative to one another in the direction tangential to the hole.

Two separate solutions were obtained based on the following assumptions:

- (i) that an elastic plain stress state existed for both the fastener and plate, and
- (ii) that an elastic plain strain state existed for both the fastener and plate.

The stiffness matrices of the standard elements were obtained using double precision and reduced integration and double precision was also used in the solution.

3.2 Results

Stresses derived from the finite element analyses for points on the fastener/hole interface are presented in Figs 13 to 16.

In the above, stresses have been presented in the non-dimensional form stress/q,

where
$$q = \frac{4G\lambda}{K+1},$$

λ = fastener interference ratio,

and $K = (3 - \nu) (1 + \nu)$ for plane stress

or $K = 3 - 4\nu$ for plane strain.

Curves have been fitted by eye to the data to smooth out small local irregularities introduced by the finite element analyses. In each case one curve was adequate to represent both the plane stress and plane strain finite element results, and for radial stress plots a single curve was applicable to both plate and fastener stresses.

In addition strains were derived, using the information from finite element analyses, at convenient positions along the axes on which strains were measured. Comparisons are given in Figs 9 and 10.

4. DISCUSSION

Various tests can be applied to estimate the accuracy of the finite element analysis. For example:

- (a) the variation between stress levels determined from different elements at the same node;
- (b) the differences between the radial stresses in the fastener and plate at the interface (Figs 14 and 16); and,
- (c) the variation, about a smooth curve, in the circumferential stresses in the plate and pin at the interface (Figs 13 and 15).

Examination of these aspects suggests, for the finite element analysis, an accuracy of about $\pm 2\%$ at the hole perimeter.

Fig. 13 depicts non-dimensional circumferential stresses at the hole boundary in the plate. The maximum value is 1.14, located nearest to the other hole on the common centreline. For comparison, the non-dimensional stress ratio for a similar interference-fit fastener in an infinite plate would be exactly 1.0.

A good correlation was found between the experimental strain results and the finite element analysis for the limited data from Specimen F. The strains measured on the other specimen (8/9) exhibit similar trends to the numerical results and there is broad agreement. However it is felt that the accuracy of fastener insertion measurements may have been reduced by unevenness in the thickness of gauge waterproofing (c.f. § 2.5) with some effect on the overall accuracy of the results for the latter specimen.

The experimental evidence is not precise enough to indicate whether the plane stress or the plane strain solution is the more accurate.

Hole irregularity is reflected in delayed strain response (see Fig 5 to 8) - corresponding to approximately 0.5 mm ineffective insertion distance, or 0.001 non-dimensional interference.

5. CONCLUSIONS

- 1. Stress and strain levels in the vicinity of a pair of closely-spaced interference-fit fasteners in a steel plate (nominal pitch of two hole diameters) have been evaluated by the finite element method and appear to be accurate to within $\pm 2\%$ in the vicinity of the hole boundary. The analysis assumed perfectly accurate hole and pin geometry and a no-slip interface.
- 2. Strain measurements in one specimen with close-spaced fasteners agree well with the finite element results at four gauge positions. Results from the second specimen are less accurate but follow the trends of the finite element results.
- 3. The peak stress level around the holes (as derived by finite element method for a hole centre distance to diameter ratio of 1.97) is 1.14 times the peak stress for a single similar fastener in an infinite plate.

ACKNOWLEDGEMENTS

The support and interest shown by Dr. R. Jones and Dr. G.S. Jost is gratefully acknowledged. The advice of Mr. S.W. Gee regarding strain measurement is also most appreciated.

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TABLE 1. ELECTRIC RESISTANCE STRAIN GAUGES

Gauge Type	Specimen	Gauge			
		Number*	Orientation relative to hole	Length (mm)	Width (mm)
Metalfilm C6-1X1-M15E	F	All	circumferential	0.38	0.51
Micromeasurements WA-06-125-BT-120	8/9	1,11	specimen side		
Micromeasurements EA-06-015-DJ-120	8/9	2,3,5,6,7	circumferential	0.38	0.51
Kyowa KFC-03-C1-11	8/9	4	circumferential	0.3	1.8
		8,9,10	radial	0.3	1.8

Adhesive used: M Bond 200 with catalyst.

* For locations see Figs 3 and 4.

TABLE 2: STRAINS RESULTING FROM INSERTION OF TAPERED BOLTS

SPECIMEN F

Gauge No.	Strain ($\times 10^{-3}$)					
	Fastener Insertion Distance (mm)					
	Thumbtight	0.254	0.667	0.954	1.248	1.899
1	0.000	0.092	0.311	0.506	0.821	1.561
2	0.000	0.070	0.387	0.604	1.064	1.824
	Thumbtight	0.320	0.670	0.938	1.301	2.001
3	0.000	0.058	0.191	0.563	0.731	1.731
4	0.000	0.113	0.229	0.598	1.132	1.792

TABLE 3: STRAINS RESULTING FROM INSERTION OF TAPERED BOLTS

SPECIMEN 8/9

Gauge No.	Strain ($\times 10^{-3}$)				
	Fastener Insertion Distance (mm)				
	Thumbtight	0.735	1.724	2.156	2.409
1	0.000	0.101	0.302	0.362	0.449
2	0.000	0.302	0.955	1.010	1.304
3	0.000	0.446	1.445	1.532	1.986
4	0.000	0.391	1.256	1.575	2.050
5	0.000	0.499	1.611	1.822	2.368
6	0.000	0.308	1.117	1.271	1.692
7	0.000	0.229	0.965	1.099	1.533
	Thumbtight	0.558	1.801	2.156	2.587
8	0.000	-0.020	-1.288	-1.773	-2.733
9	0.000	-0.022	-0.628	-1.097	-1.983
10	0.000	-0.068	-1.213	-1.692	-1.900
11	0.000	0.033	0.243	0.316	0.405

TABLE 4: MEASURED STRAIN PER UNIT INTERFERENCE RATIO

Specimen	Gauge			Strain/ λ
	No.	Location Axis	Position x/R or y/R	
F	1	Oy	1.16	0.482
	2	Ox	1.19	0.558
	3	Oy	1.20	0.523
	4	Ox	1.21	0.548
8/9	1	Ox	5.19 (spec. side)	0.093
	2	Ox	1.49	0.263
	3	Ox	1.20	0.404
	4	Oy	1.23	0.439
	5	Ox	1.25	0.492
	6	Ox	1.54	0.363
	7	Ox	1.79	0.338
	8	Ox	1.14	-0.593
	9	Oy	1.14	-0.412
	10	Ox	1.12	-0.436
	11	Ox	5.20 (spec. side)	0.085

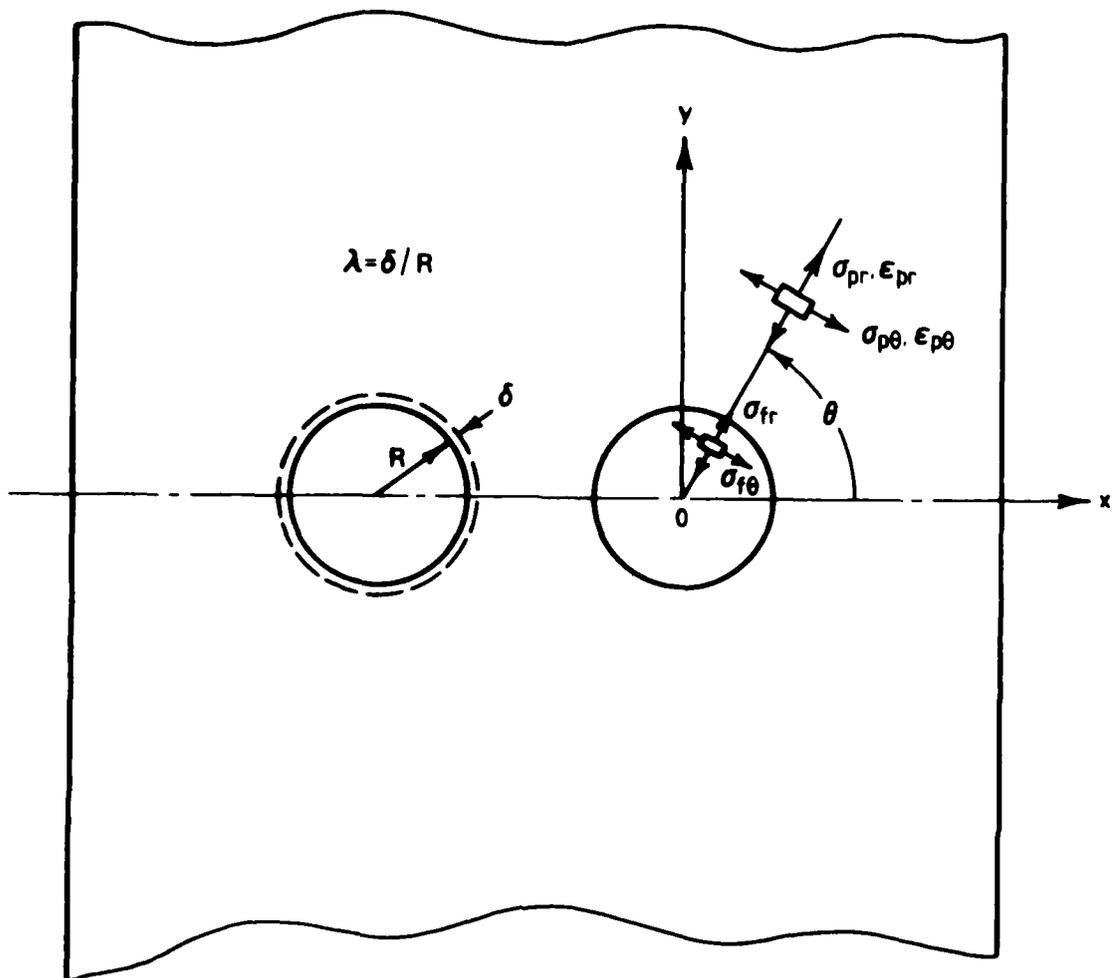


FIG. 1 NOTATION DIAGRAM

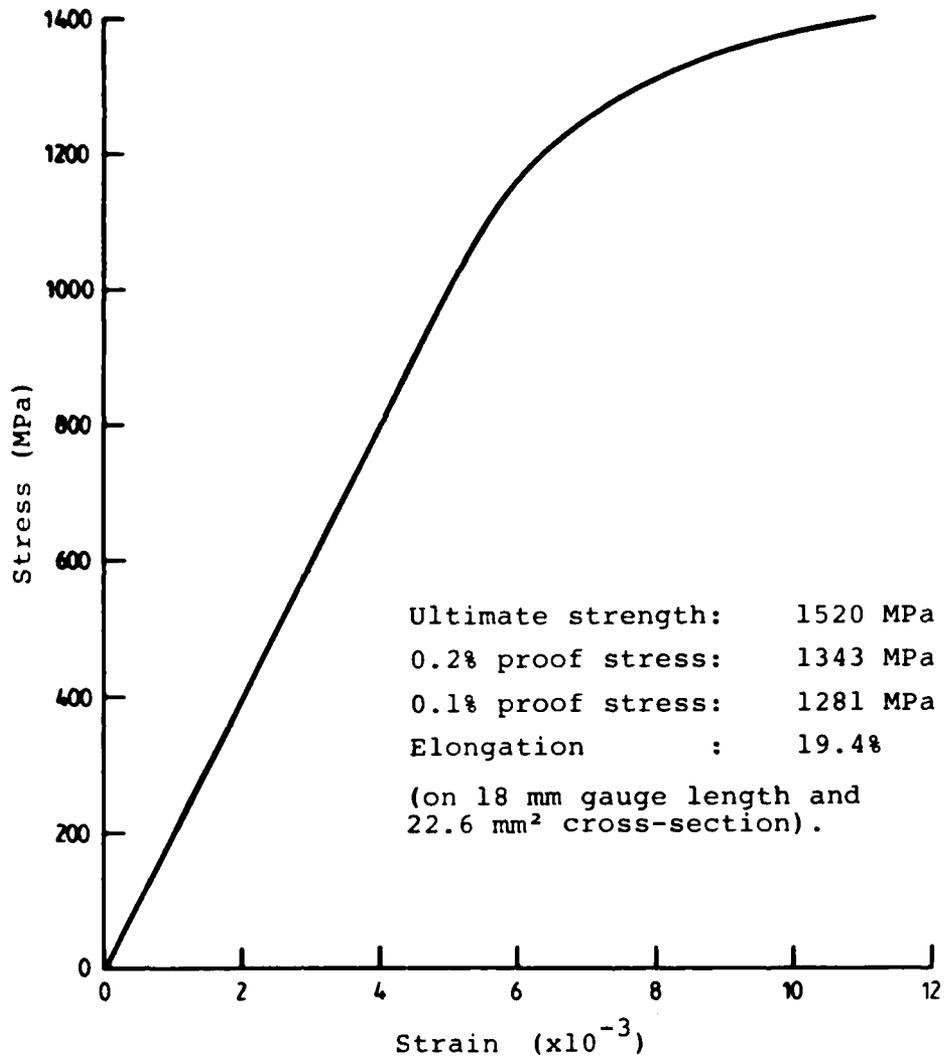
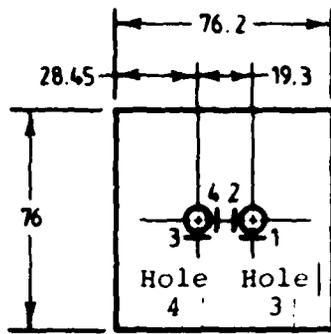


FIG. 2 STRESS-STRAIN CHARACTERISTICS FOR D6AC STEEL (MATERIAL OF SPECIMEN F).



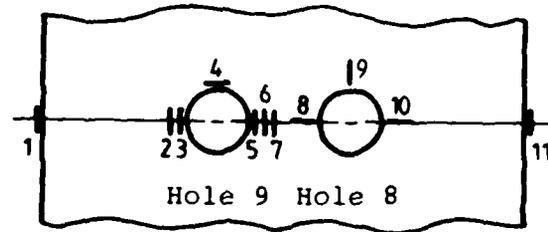
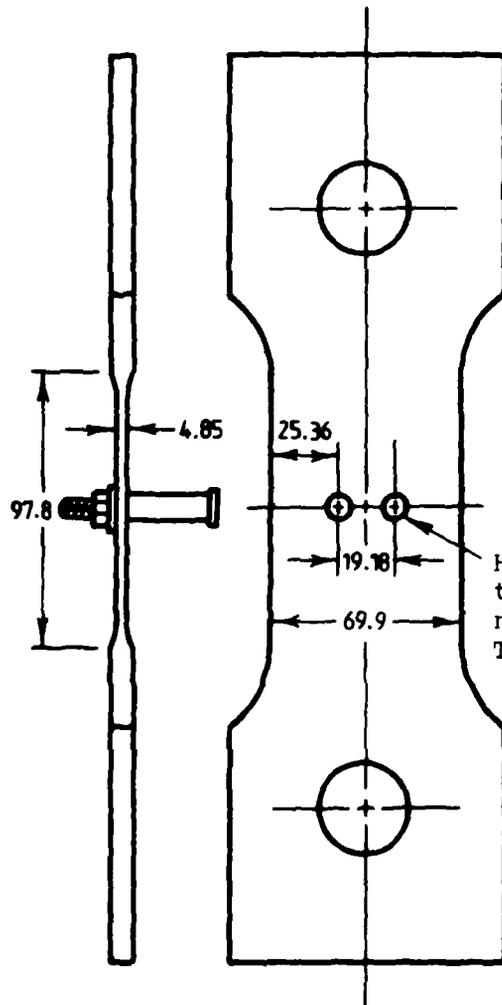
Gauge No.	Centreline distance from hole edge (mm)
1	0.76
2	0.91
3	0.99
4	1.02

Plate thickness = 4.55 mm

Holes taper-reamed to suit 9.525 mm nominal diameter Taper-Lok bolts.

Hole diameters at insertion face are 9.701 mm (hole 3) and 9.709 mm (hole 4)

FIG. 3 CONFIGURATION AND GAUGE LOCATIONS - SPECIMEN F.



Centreline distances from hole edge (mm).

Gauge No.	1	20.4
2	2.41	
3	0.96	
4	1.12	
5	1.21	
6	2.63	
7	3.83	
8	0.70	
9	0.67	
10	0.56	
11	20.4	

Hole diameters at insertion face are 9.747 mm (hole 8) and 9.764 mm (hole 9).

All dimensions in mm.

FIG. 4 CONFIGURATION AND GAUGE POSITIONS SPECIMEN 8/9.

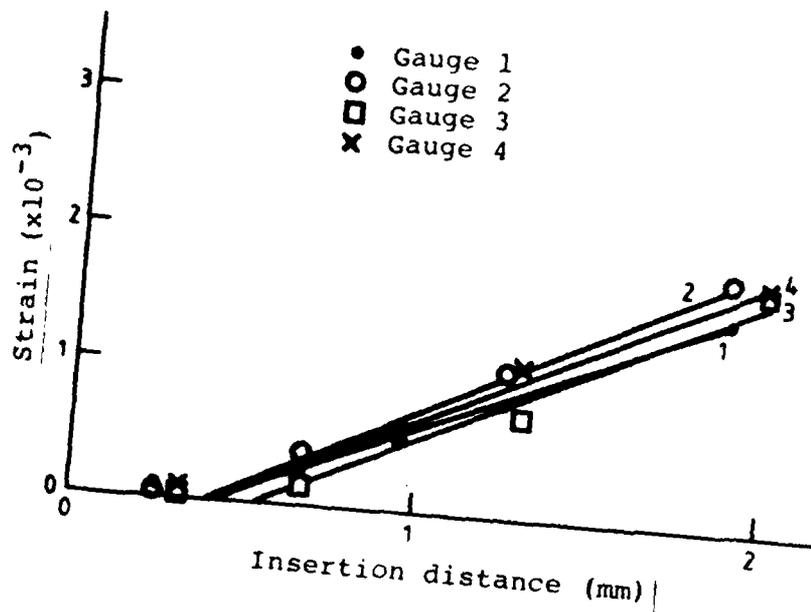


FIG. 5 STRAIN VERSUS INSERTION DISTANCE - SPECIMEN F.

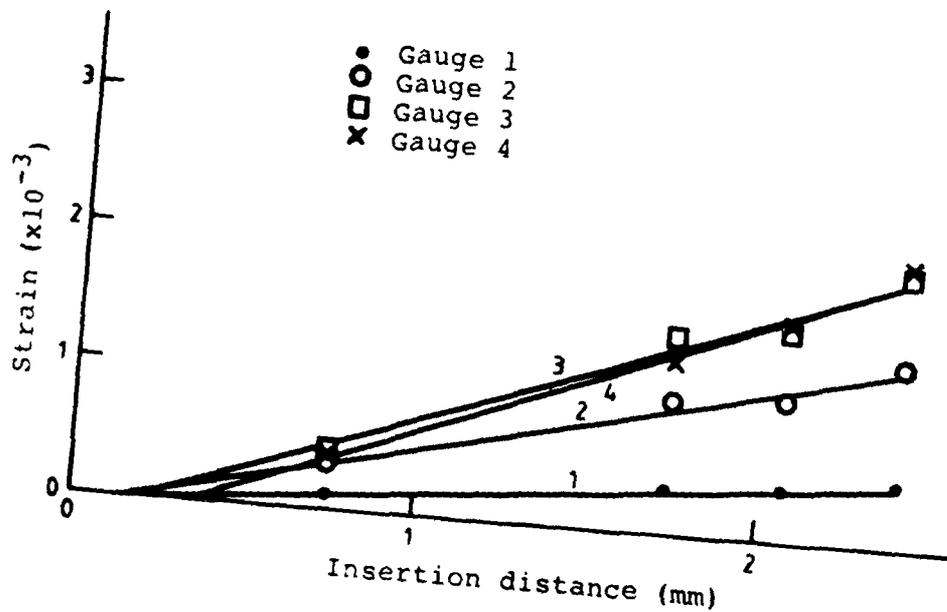


FIG. 6 STRAIN VERSUS INSERTION DISTANCE - SPECIMEN 8/9

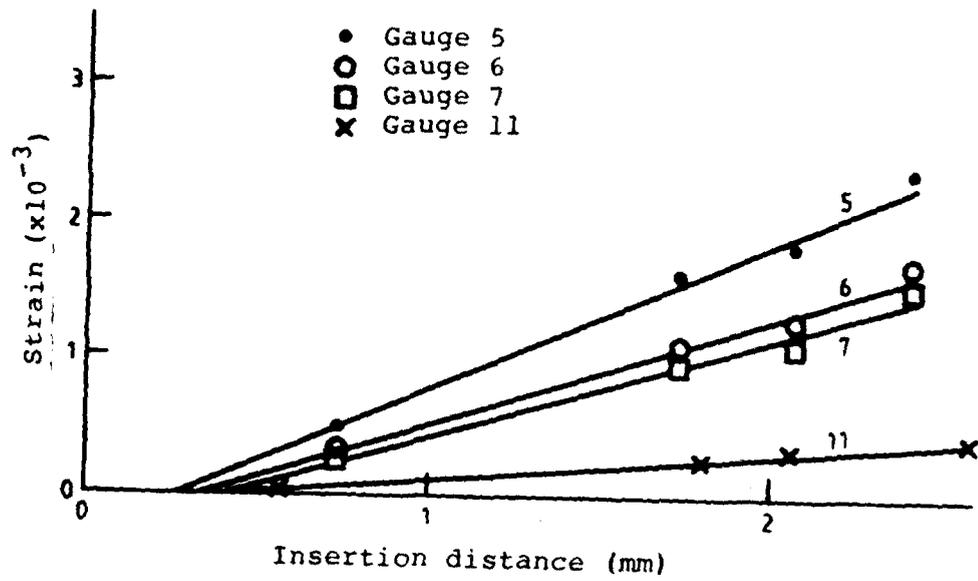


FIG. 7 STRAIN VERSUS INSERTION DISTANCE - SPECIMEN 8/9.

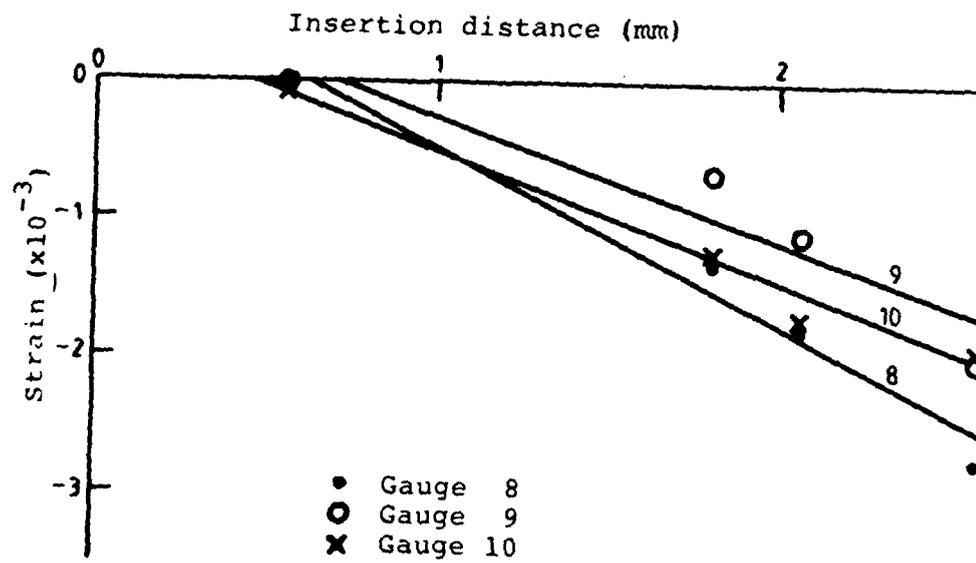


FIG. 8 STRAIN VERSUS INSERTION DISTANCE - SPECIMEN 8/9.

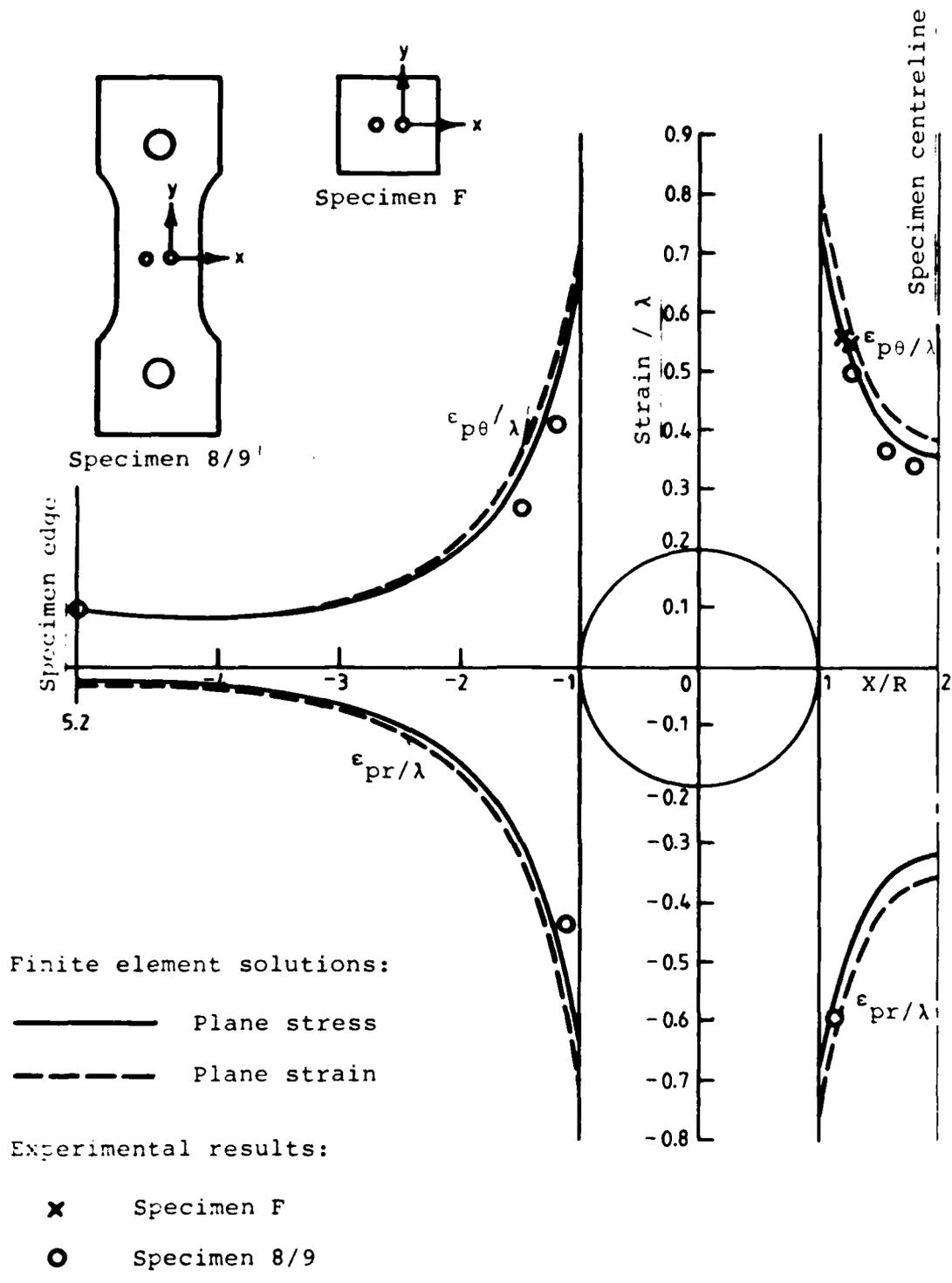


FIG. 9 CIRCUMFERENTIAL AND RADIAL STRAIN RATIOS ON LINE JOINING CENTRES OF CLOSELY-SPACED FASTENERS - COMPARISON OF EXPERIMENTAL RESULTS WITH FINITE ELEMENT SOLUTIONS.

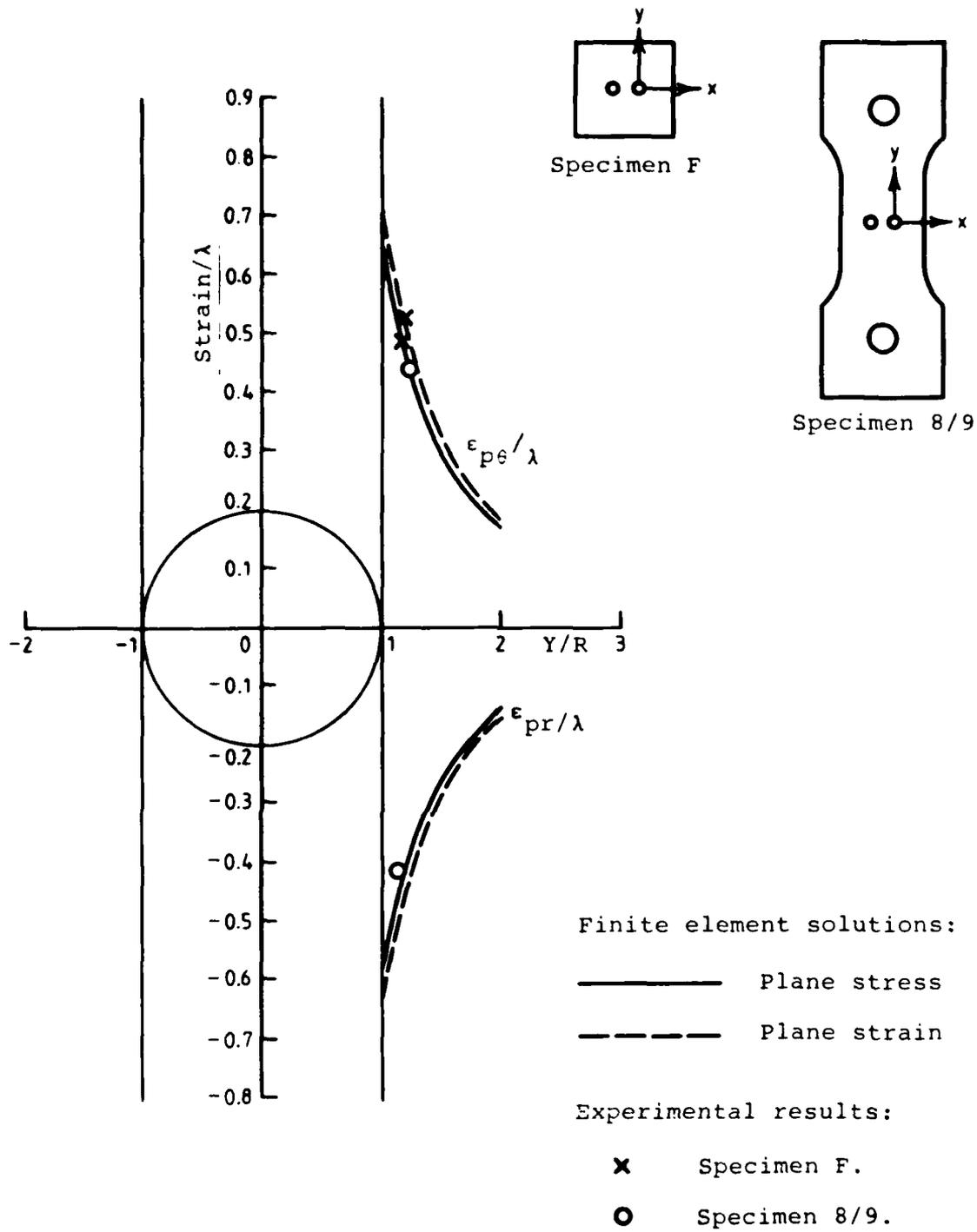


FIG. 10 CIRCUMFERENTIAL AND RADIAL STRAIN RATIOS ON PERPENDICULARS TO LINE JOINING FASTENER CENTRES - EXPERIMENTAL RESULTS VERSUS FINITE ELEMENT SOLUTIONS.

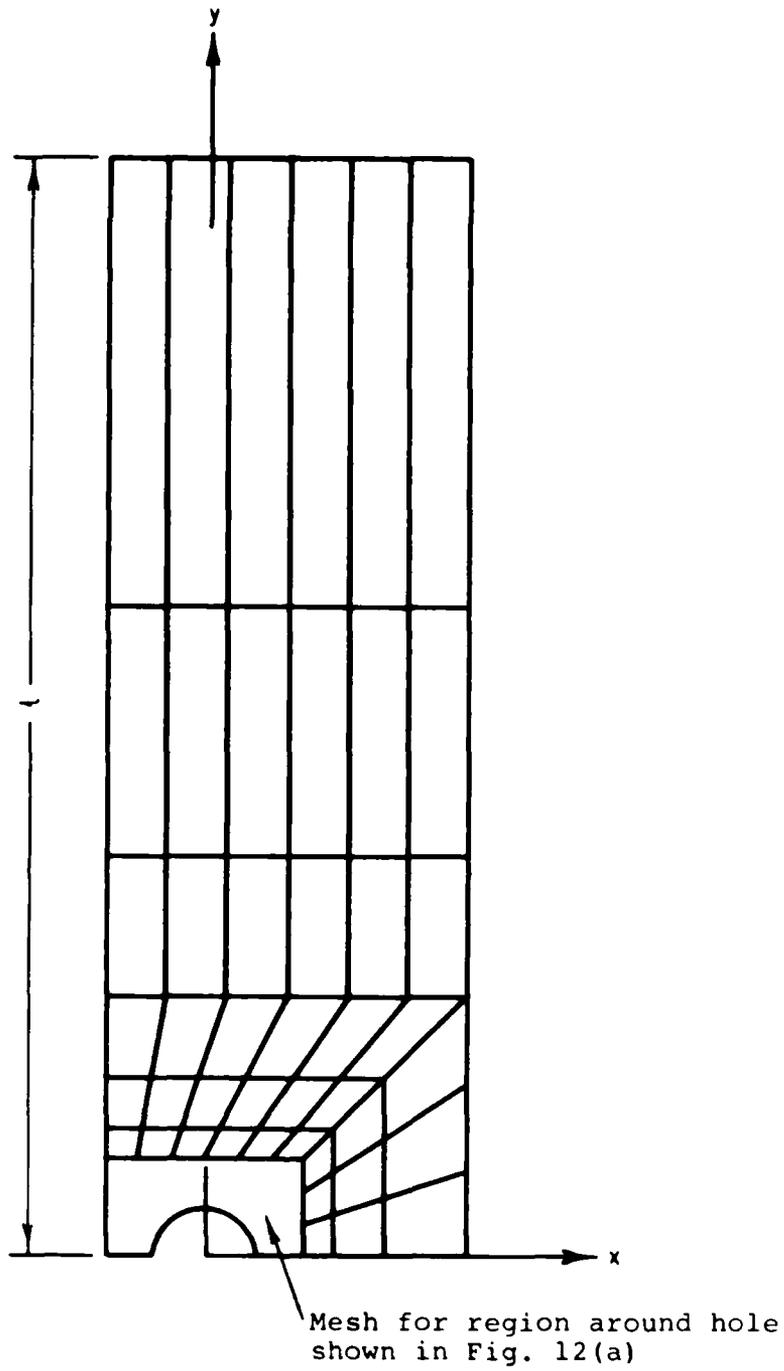
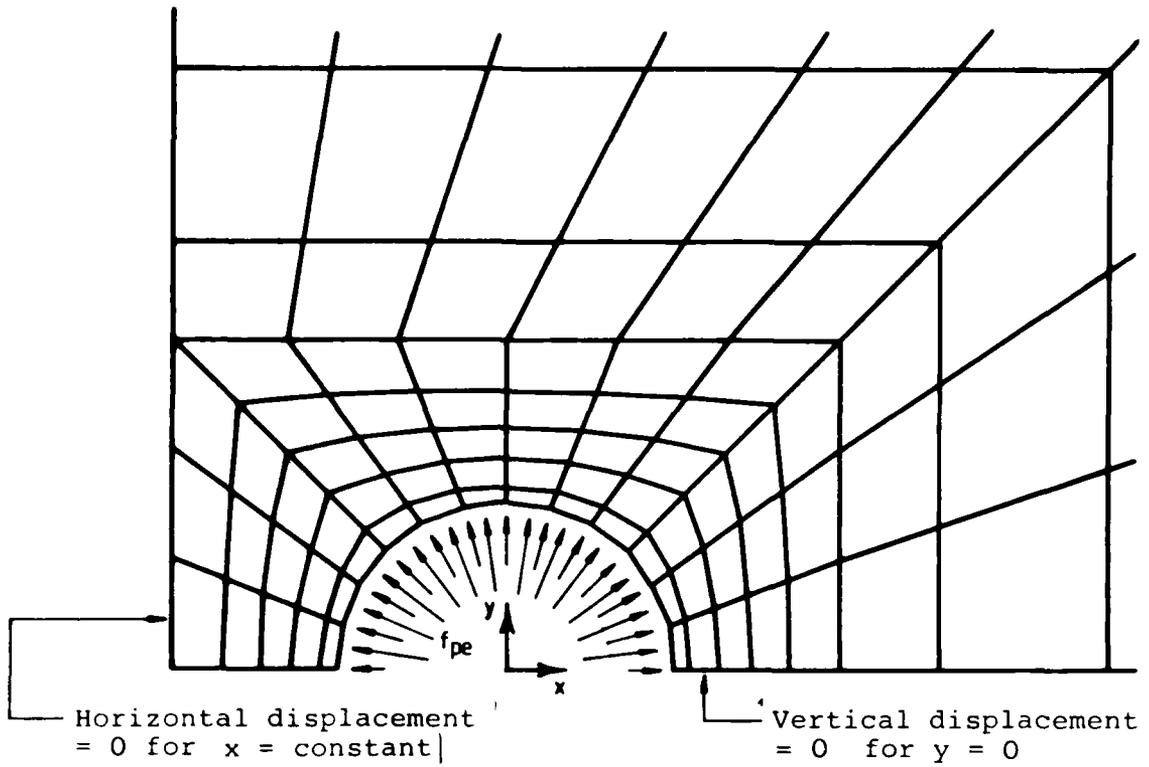
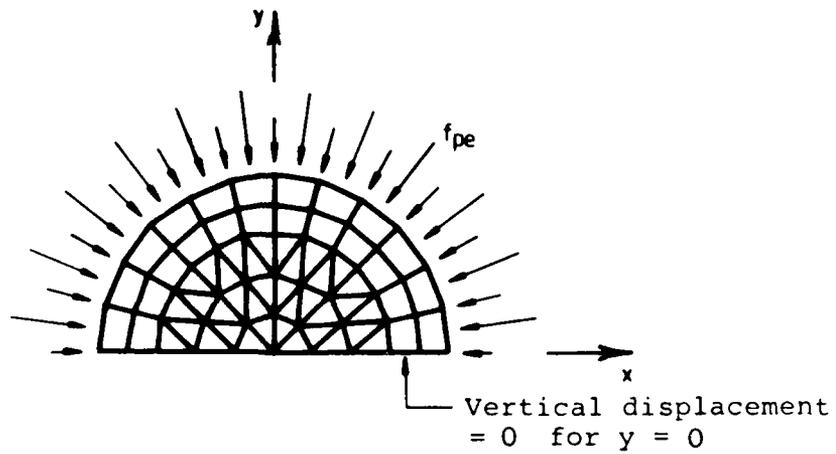


FIG. 11 FINITE ELEMENT MESH FOR PLATE.



(a) Plate



(b) Fastener

FIG. 12 FINITE ELEMENT MESH IN REGION OF HOLE - INTERACTION BETWEEN FASTENER AND PLATE.

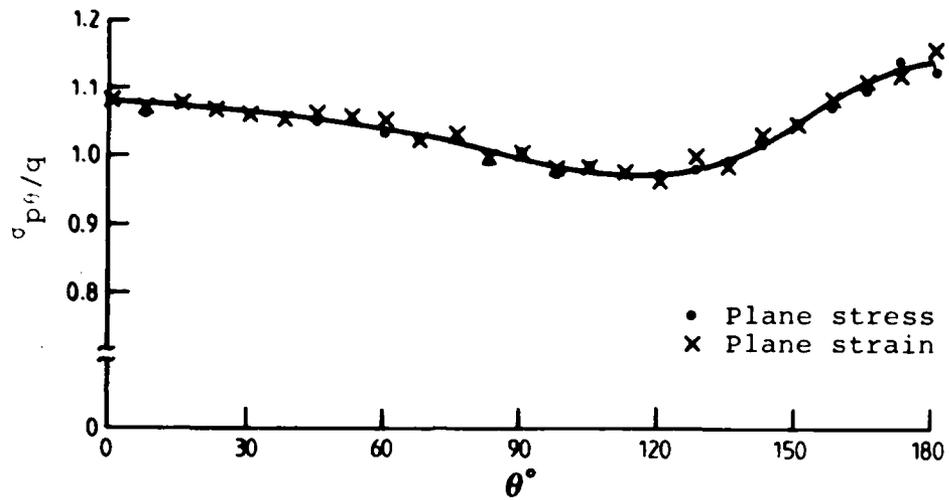


FIG. 13 CIRCUMFERENTIAL STRESS RATIOS AT HOLE PERIMETER IN PLATE - FROM FINITE ELEMENT ANALYSES.

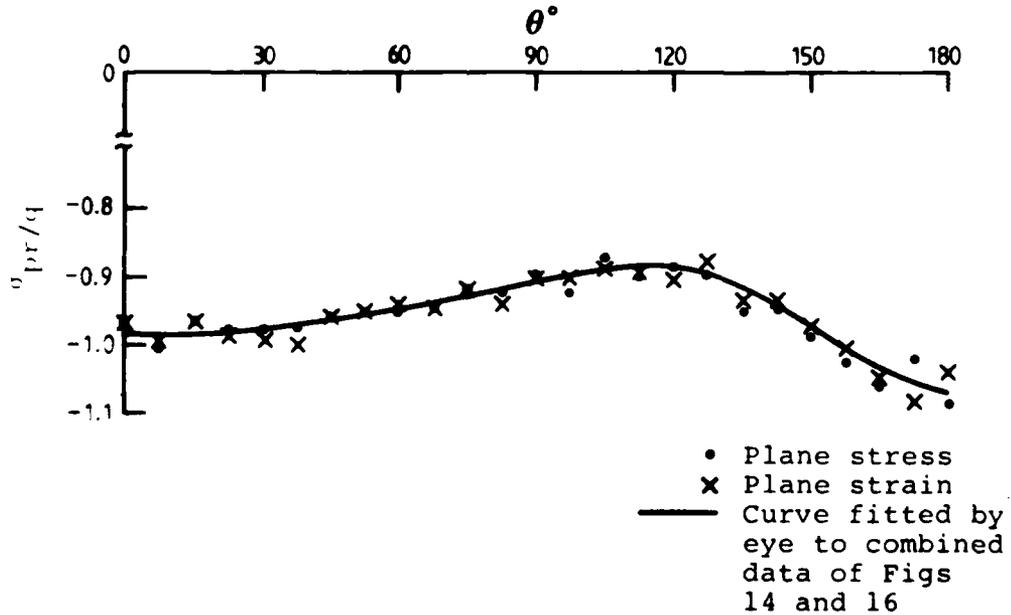


FIG. 14 RADIAL STRESS RATIOS AT HOLE PERIMETER IN PLATE - FROM FINITE ELEMENT ANALYSES.

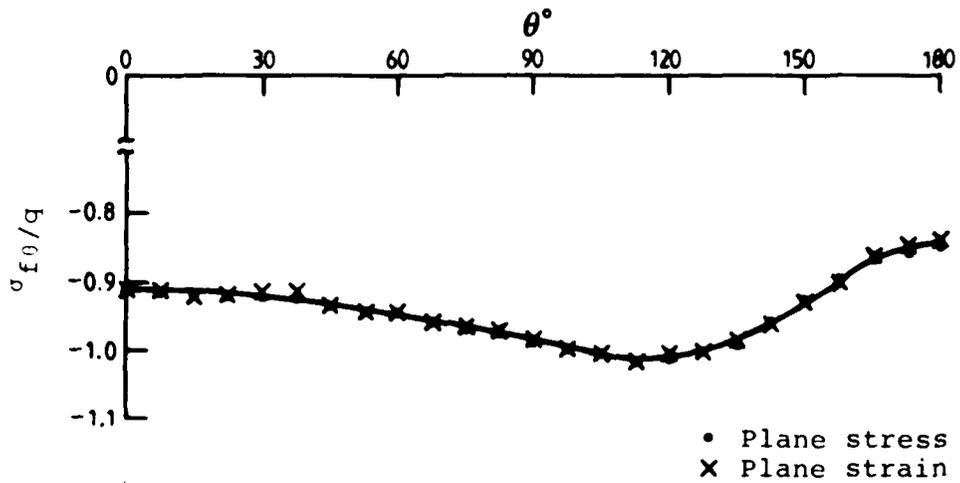


FIG. 15 CIRCUMFERENTIAL STRESS RATIOS ON FASTENER PERIMETER - FROM FINITE ELEMENT ANALYSES.

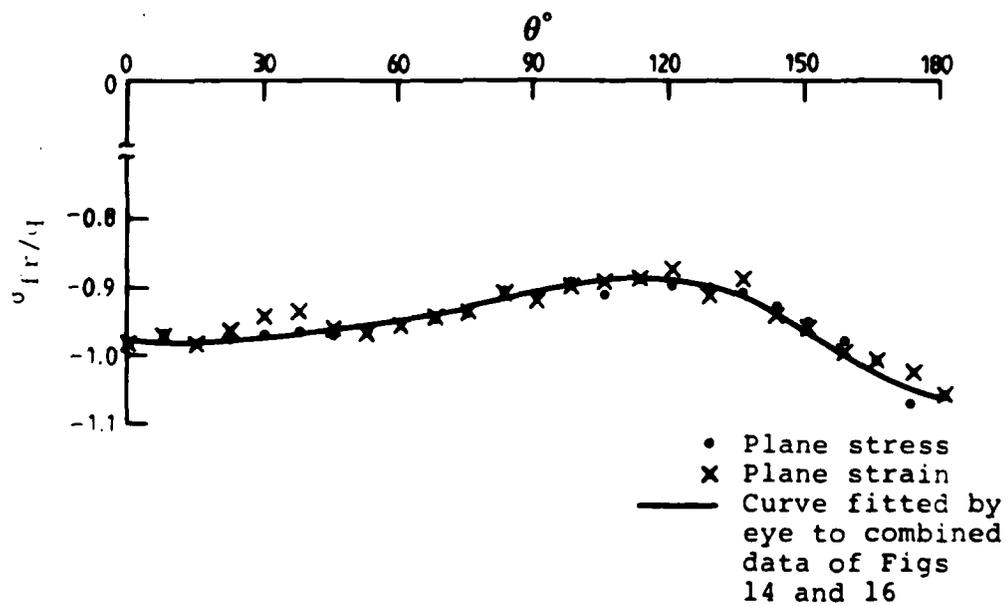


FIG. 16 RADIAL STRESS RATIOS ON FASTENER PERIMETER - FROM FINITE ELEMENT ANALYSES.

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