

The Use Of Pin Bearing Data For The Preliminary Design Of "Bolted" Joints

Summary

A major design issue remains in progressing from the "coupon" strengths measured in these tests to bolted joints in different applications. A method is described for the approximate determination of a multiple, "bolted" joint loaded in tension, based on the integration of the various steps as outlined.

Comparison with experimental results show that the method can be used as a first approximation for design purposes and would be suitable for inclusion in the National Physical Laboratory (NPL) composites design analysis software (CoDA).

An overview of minimum fastener spacing is presented in Chapter 5 for the use of designing composite double lap joints.

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1 Introduction

Most of the data for the design of composite bolted joints is based on the testing of simple coupons having pin/hole combinations. Currently, three coupon methods covering open hole tension [1], open/filled hole compression [2] and pin bearing [3] have been proposed by the UK as ISO new work items (NWI).

Although these tests are often considered to belong to the aerospace industry and do appear in their material specifications, all types of composites contain holes and bolted or other fittings. For example, the new Comite European de Normalisation (CEN) (TC249/SC2/WG6) specification standard being developed, includes a required figure for both longitudinal and transverse pin bearing strength as a mandatory requirement within a set of properties.

There are various modes of failure for a pin loaded joint [4], of these only two, bearing and tension through the hole are desirable, as the others give premature failures at lower loads. For laminates loaded in double shear, the distinction between bearing and tension is established largely by the joint geometry, particularly the width to diameter ratio (w/d), with tension failure being the most prominent for ratios of $w/d < 4$.

In design, it is customary to allow for higher stress concentration at a loaded hole than at an unloaded hole, also a distinction must be made between bearing and non-bearing or by-pass loads, to be able to characterise the internal loads in multi-row joints. In the following sections, the different stages of an analysis from plain hole to multi-row pinned joint is shown, considering in turn the following problems:

- distribution of load in the pins and plate (by-pass load) for a multiple pin array
- prediction of stress concentration in a pin-loaded hole
- prediction of failure load for a multiple pin loaded strap.

Finally, the analysis is compared and validated with experimental data,

2 Load Sharing Analysis

In this Section, the analysis for the determination of the load sharing between individual bolts in a line of bolts in a symmetrical lap joint is outlined. The analysis is based on the model used by Tate and Rosenfeld [5].

2.1 Geometry And Model Assumptions

A schematic diagram of the bolted joint is shown in [Figure 2.1](#). The upper part of the diagram shows co-linear bolts in a strap of width w in line with the total applied load P . The lower part of the diagram shows the construction of the joint in which a plate of uniform thickness t_p is situated between two straps of uniform thickness t_s . The assumptions of the model are as follows:

- The ratio of stress to strain is constant.
- The stress is uniformly distributed over the cross sections of main plate and butt straps.
- The effect of friction is negligible
- The bolts fit the holes initially, and the material in the immediate vicinity of the holes is not damaged or Stressed in making the holes or inserting the bolts.
- The relationship between bolt deflection and bolt load is linear in the elastic range.
- Each bolt is assumed to be a short loaded fixed end beam; allowances are made for shearing, bending and bearing effects.

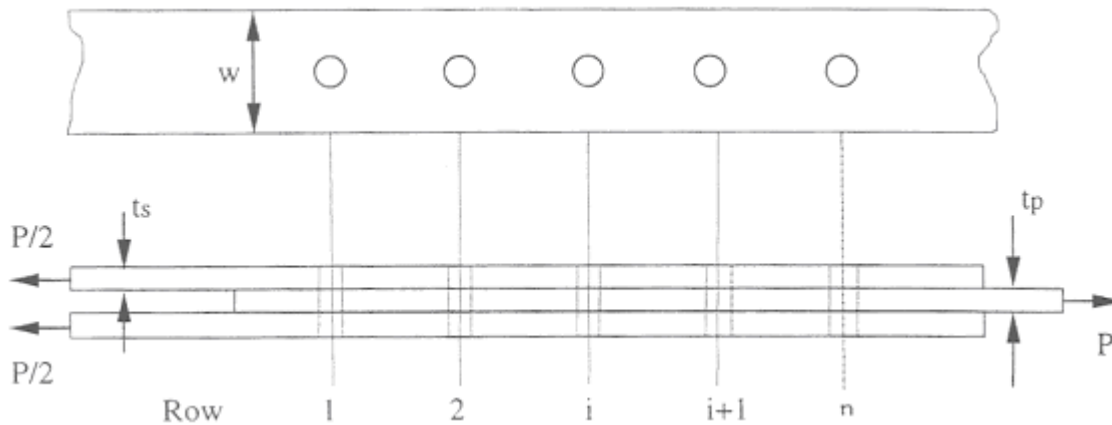


Figure 2.1 Symmetrical butt joint with pins in line of applied load

2.2 Model Equations

The relative displacement of plate and straps between the i th and $(i+1)$ th bolts yields for $i=1, 2, \dots, (n-1)$ is given below:

$$R_{i+1} = \frac{C_i}{C_{i+1}} R_i + \frac{(2K_p + K_s)}{C_{i+1}} R_i - \frac{2K_p}{C_{i+1}} P + \frac{(2K_p + K_s)}{C_{i+1}} \sum_{r=1}^{i-1} R_r \quad (2.1)$$

where R_i is the i th bolt load and p the external applied load. C_i denotes a constant for the i th bolt dependent upon material properties, plate and strap dimensions and includes effects of bending, shear and bearing. K_p and K_s are given by:

$$K_p = \frac{P}{wt_p E_p}, K_s = \frac{P}{wt_s E_s} \quad (2.2)$$

where p is the hole spacing or pitch and E_p and E_s denote Young's modulus for the plate and strap respectively. The overall equilibrium condition gives:

$$P = \sum_{r=1}^{n-1} R_r \quad (2.3)$$

Each bolt constant C_j has contributions from the various beam mechanisms in operation:

$$C_i = C_{bs} + C_{bb} + C_{bbr} + C_{pbr} \quad (2.4)$$

where

$$C_{bs} = \frac{2t_s + t_p}{3G_b A_b} \quad (\text{shear effect})$$

$$C_{bb} = \frac{8t_s^3 + 16t_s^2 t_p + 8t_s t_p^2 + t_p^3}{192E_{bb} I_b} \quad (\text{bending effect})$$

$$C_{bbr} = \frac{2t_s + t_p}{t_s t_p E_{bbr}} \quad (\text{bearing effect})$$

$$C_{pbr} = \frac{1}{t_s E_{pbr}} + \frac{2}{t_p E_{pbr}} \quad (\text{plate bearing effect})$$

and $A_b = \pi d^2/4$, $I_b = \pi d^4/64$ and d is the bolt diameter

In the above equations, E_{bb} and E_{bbr} are Young's moduli for the bolt under bending and bolt under bearing respectively. E_{sbr} and E_{pbr} are the bearing moduli for the strap and plate, assumed equal to the compressive moduli of the respective materials. Assuming that the bolt diameter d is the same for each, then bolts of the same material imply that $C_i = C_{i+1} = C$.

For purposes of the present implementation, it is assumed that:

$$E_{pbr} = E_p, E_{sbr} = E_s, E_{bbr}, E_{bb} = E_b \quad (2.5)$$

The analysis can be extended to allow for the presence of bolts in several lines parallel to the applied load as shown in [Figure 2.2](#). In addition to the previous assumptions, it is assumed that the bolts in any transverse row are loaded equally and that the bolts in any transverse row are identical, although this latter assumption is not explicitly stated in the Tate and Rosenfeld paper [\[5\]](#).

The corresponding form of (2.1) is:

$$R_{i+1} = \frac{C_i}{C_{i+1}} R_i + N_i \frac{(2K_p + K_s)}{C_{i+1}} R_i - \frac{2K_p}{C_{i+1}} P + \frac{(2K_p + K_s)}{C_{i+1}} \sum_{r=1}^{i-1} R_r \quad (2.6)$$

where N_i denotes the number of bolts in the i th row.

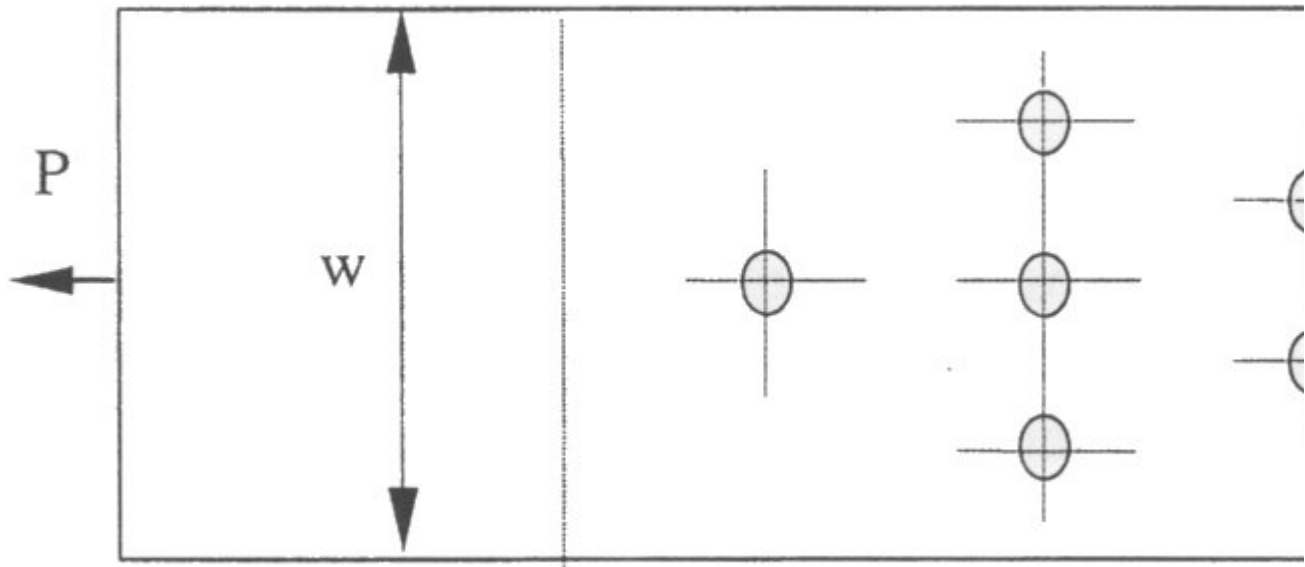


Figure 2.2 Symmetrical joint with multiple pins

2.3 Solution Of The Model Equations

To enable R_j to be evaluated, Equation 2.1 is rearranged:

$$R_i = [R_{i+1} - \lambda \sum_{r=1}^{r=i-1} R_r + \mu] / (1 + \lambda), \quad i = 1, 2, \dots, n-1 \quad (2.7)$$

where:

$$\lambda = \frac{2K_p + K_s}{C}, \quad \mu = \frac{2K_p P}{C}$$

As the bolts are normally of the same material.

These equations are solved using a form of Gauss Seidel iteration as follows. An initial estimate to each R_i is made; an obvious choice is to use $R_i = P/n$. Using $i = 1, 2$ etc in turn in Equation 2.7 enables new estimates to be made for the R_j up to R_{n-1} . Finally Equation 2.8 provides a new estimate for R_n .

$$R_n = P - \sum_{r=1}^{n-1} R_r \quad (2.8)$$

This whole process is repeated until relative changes from one iteration to the next are less than some specified tolerance, usually 1×10^{-6} , at which point the calculation for the bolt loads R_j is considered completed. The bypass loads P_{S_i} are determined from:

$$P_{S_i} = \sum_{j=i+1}^{j=n} R_j \quad (2.9)$$

2.4 Validation Of Tate And Rosenfeld

Finite element analysis (PEA) as well as electronic speckle pattern interferometry (ESPI) [6] have been used to validate the Tate and Rosenfeld method [5]. Table 2.1 compares load share values between Tate and Rosenfeld and non-linear contact PEA (using orthotropic values) for a double lap four pin pultruded joint.

Table 2.1 Tate and Rosenfeld (T&R) Prediction compared to FEA

Pin Number	Bearing load on pin	
	T&R	FEA
1	0.374	0.35
2	0.217	0.22
3	0.177	0.19
4	0.232	0.22

3 Failure Loads In Multi-Row Joints

The main objective of using multi-row joints is to minimise the peak bearing load, avoiding the cut-off due to bearing as shown in Figure 3.3 [7]. To achieve improved strength the joint has to be designed to ensure even load sharing between the fasteners. If the joint has been correctly designed to reduce bearing loads, failure under tensile loading will occur in net tension. However there are two sources of tensile stress at the edge of a fastener hole:

- the load reacted out on the fastener by bearing
- the load bypassing the fastener to be reacted out on other fasteners along the joint

An experimental approach to finding the allowable load on a composite joint is to produce a plot, such as in Figure 3.4, in which the outer envelope of allowable gross strain e_u (away from the fastener hole) is shown as a function of the bearing stress σ_b . All that is needed is to establish by analysis the peak bearing stress at the critical hole in the joint. In Figure 3.4 the allowable gross strain for pure by-pass is shown as 4000 microstrain, reducing to around 3000 microstrain at the bearing cut off. These envelope plots can be produced by measuring:

- the failure stress of a joint element with an unloaded hole that should fail in tension at the fastener hole. Methods such as the Whitney Nuismer [8-9] failure criteria can also be used if the characteristic distance is known,
- the bearing stress in a wide strip (six times the diameter of the hole) forced to fail in bearing,
- the failure stress of a narrow strip (three times the hole diameter) designed to fail in tension at the hole at some combinations of by-pass and bearing.

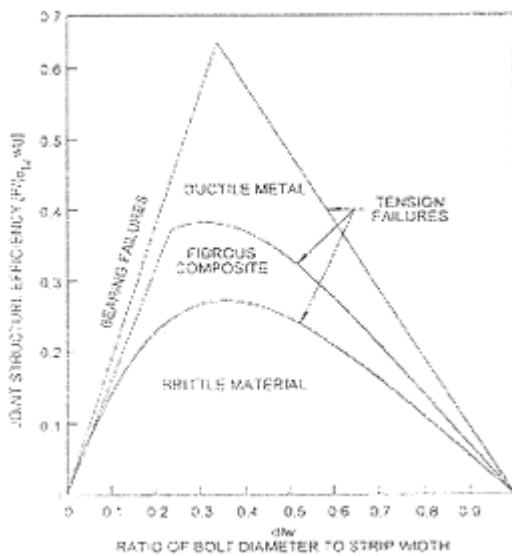


Figure 3.3 Plot of joint efficiency versus d/w for single fastener joints [7]

For tension loading the conditions for onset of damage are similar to those shown schematically in [Figure 3.4](#), indicating a cut-off of tension reacted bearing strength for net tension failures as the magnitude of the by-pass stress increases. The results are reasonably explained by two straight line plots, one for bearing failures and the other for net tension.

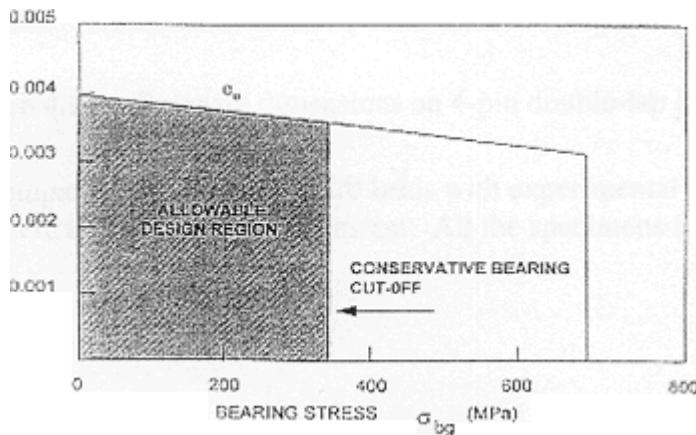


Figure 3.4 Design allowable for bearing-bypass interaction under tensile loading. [7]

Under compression loading, the behaviour is markedly different as the bolt (if the clearances are small) can support the walls of the hole, transmitting part of the load directly and thus delaying the onset of net compression. The predominant failure mode is therefore compression reacted bearing failure.

An analytical approach for multi-pin joints under tension in a strip w can be used, based on the hypothesis of linear interaction equation between the net tension by-pass and bearing stresses (found with the Tate and Rosenfeld method) using the relationship as shown in Equation 3.1 :

$$\sigma_{tu} = K_{te}\sigma_t + K_{tb}\sigma_b \quad \sigma_b \leq \sigma_{bultimate} \quad (3.1)$$

In which σ_{tu} is the basic laminate strength, a is the by-pass stress at the hole under consideration K_{tb} is the bearing stress concentration factor and K_{tc} is the by-pass concentration factor found from experimentation or using the Whitney Nuismer failure criteria.

Based on the work of others Hart-Smith [4] showed that the stresses associated with σ_t the elastic concentration factor (K_{te}) for an unloaded hole in an isotropic strip width is given by:

$$K_{te} = 2 + \left[1 - \frac{d}{w} \right]^3 \quad (3.2)$$

To allow for stress reduction in composites at failure loads, an approximate linear relationship is used between K_{te} and K_{tc} . Since the two coefficients must be equal at $K_{te} = 1$ the equation used is:

$$C = \frac{K_{te} - 1}{K_{tc} - 1} \quad (3.3)$$

where c is the correlation coefficient for that particular laminate, environmental conditions and geometry of the joint. A value of $C=0$ indicates full relief of the stress concentration, whereas $C=1$ indicates a brittle material.

Using this c factor the stress concentration factor K_{tc} for a loaded hole can be found combining Equations 3.3 and 3.4 where the isotropic stress concentration factor for a loaded hole is:

$$K_{te} = \frac{w}{d} + 1 - 1.5 \left[\frac{\frac{w}{d} - 1}{\frac{w}{d} + 1} \right] \quad (3.4)$$

As the tensile failure load with no by-pass can be expressed either in terms of net section or in terms of the contact area dt we have:

$$P = \frac{\sigma_{tu}(w-d)t}{K_{te}} = \frac{\sigma_{tu}dt}{K_{bc}} \quad (3.5)$$

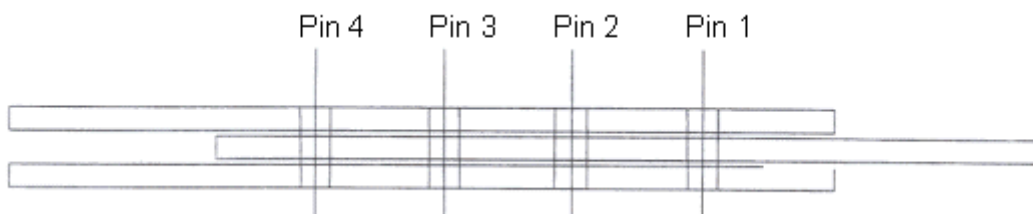
This gives an expression (Equation 3.6) for the bearing concentration factor:

$$K_{tb} = \frac{K_{te}}{\left(\frac{w}{d} \right) - 1} \quad (3.6)$$

4 Four Pin Double-Lap Tests

To validate the above method, a variety of tests were carried out from single row multi pin tests to multi row arrays of pin arrangements with differing w/d ratios, pitch distances, numbers of pins and materials.

Initially four pin double lap joints were tested at a range of w/d values as shown in [Figure 4.1](#). A pin loaded specimen was chosen as this is the simplest stress distribution as well as the worst case loading condition. Four pins were chosen so that the load sharing could be favourably demonstrated.



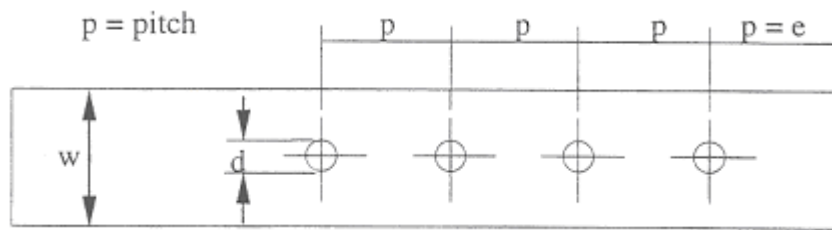


Figure 4.1 Relevant dimensions on 4-pin double-lap joint.

Figures 4.2 to 4.4 compare the predicted failure loads with experimental values obtained for the three materials were the pitch p is kept constant. All the specimens failed in net tension at pin 1, as expected.

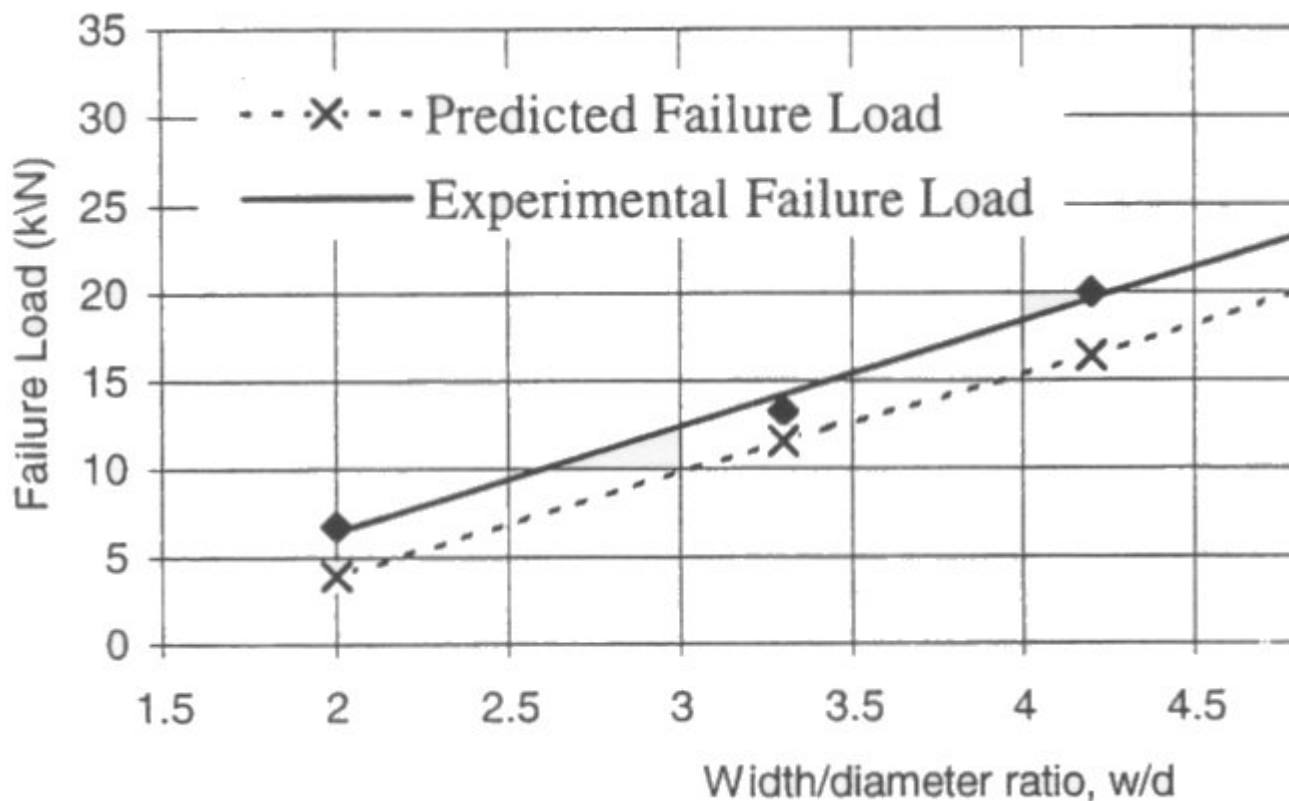


Figure 4.2 Results of pultruded 4-pin double-lap pin tests

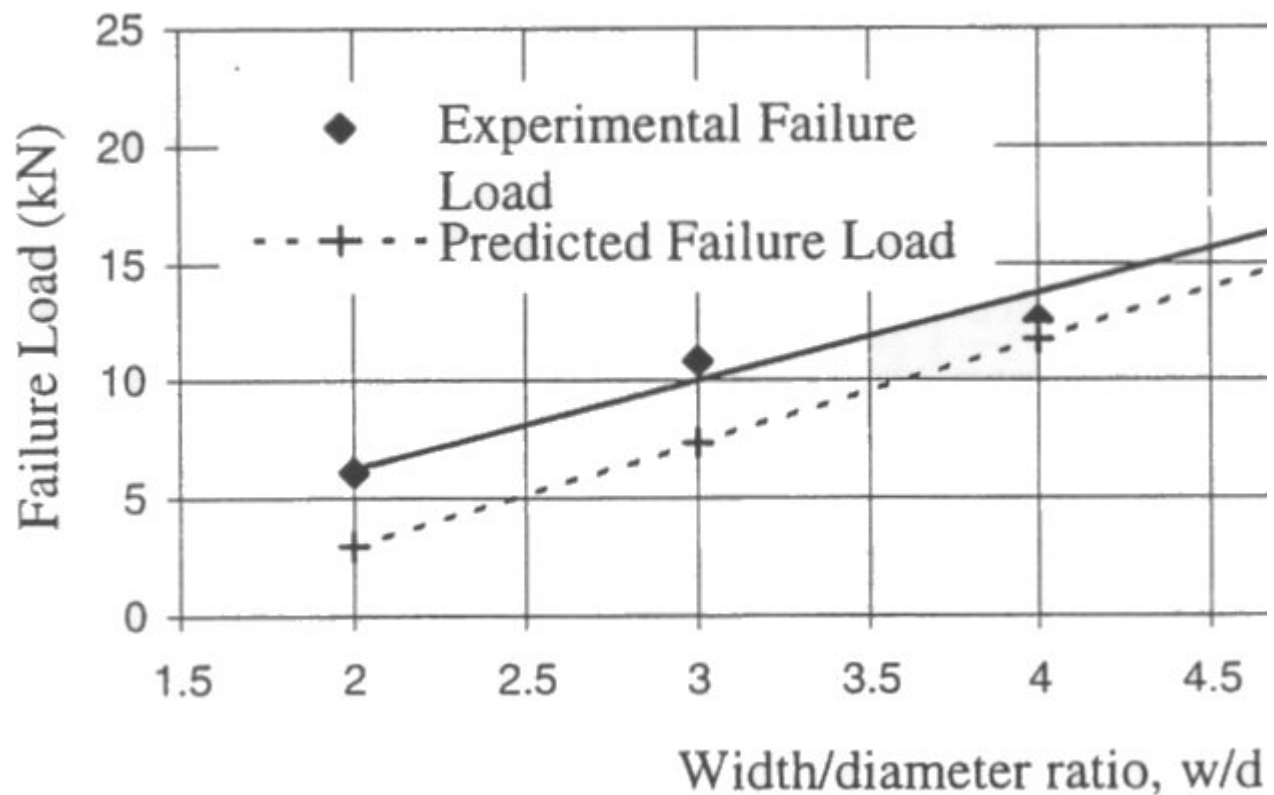


Figure 4.3 Results of woven glass-fibre epoxy 4-pin double-lap pin tests

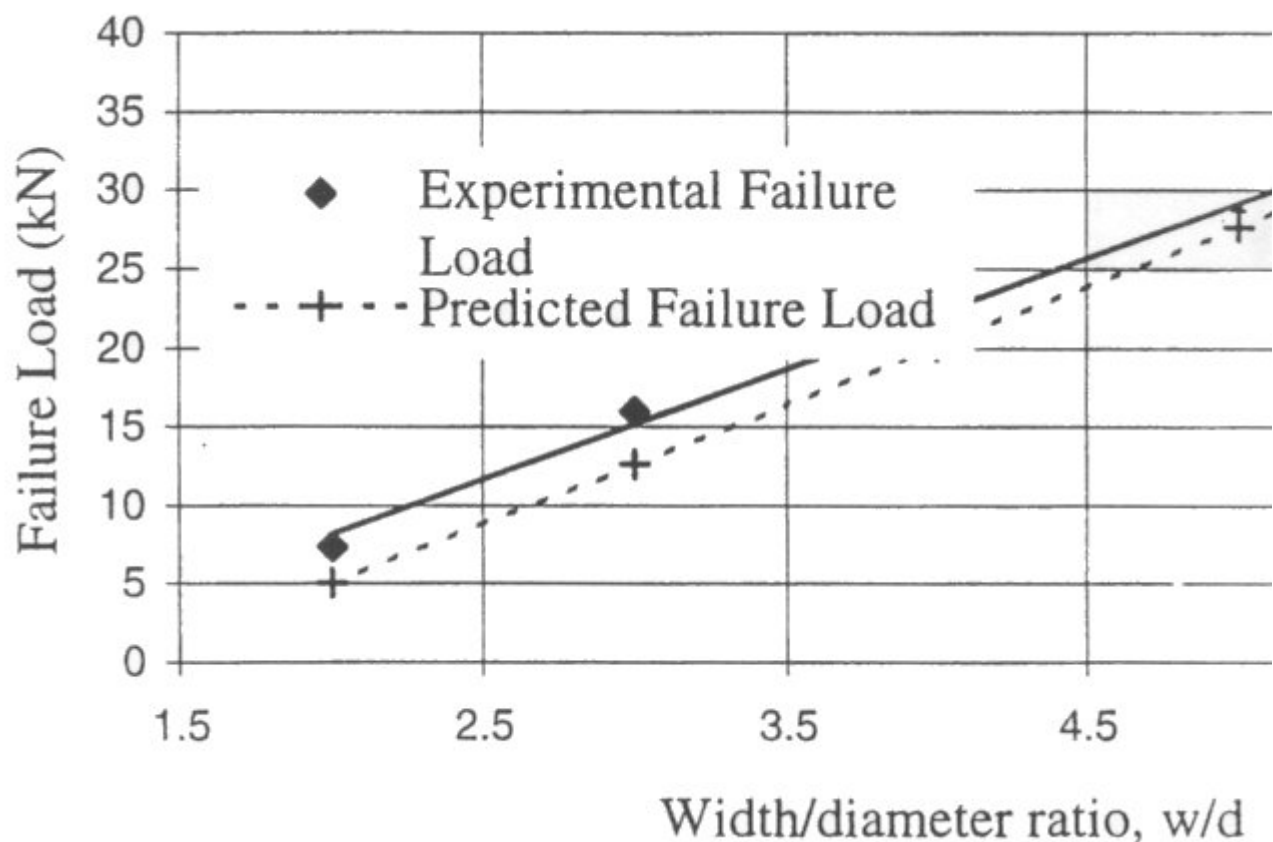


Figure 4.4 Results of carbon-fibre/epoxy 4-pin double-lap pin tests

Calculations for four pin single row joints were compared to pin and bolted joints as shown in [Figure 4.5](#). NPL has

shown [10] that torquing up a bolt can increase the bearing capacity of a joint by over 50% for certain materials and lay-ups. The methods described in Chapters 2 and 3 predict the maximum pin loaded stress of the joint, as this would be the worst case for design.

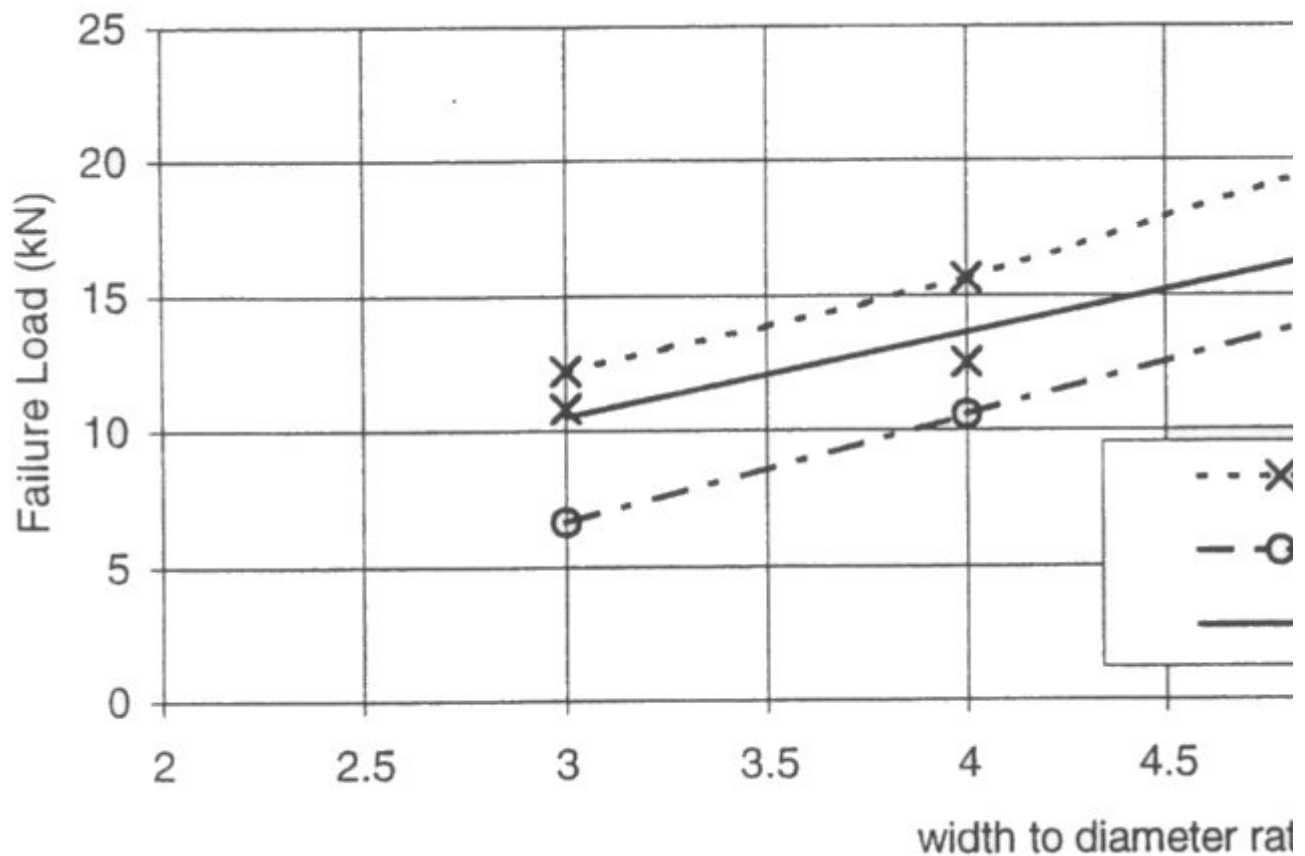


Figure 4.5 Comparison of pin/bolted and predicted loads for a glass-fibre epoxy material

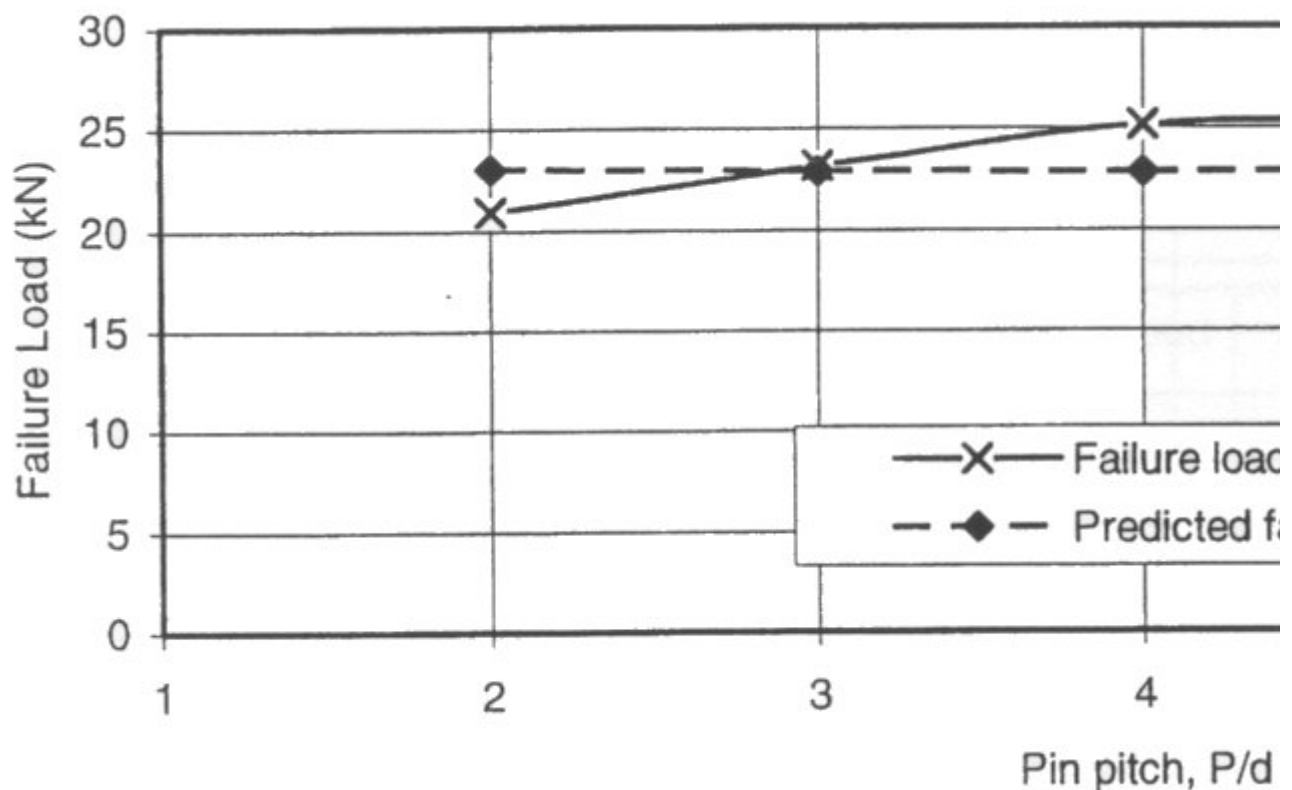


Figure 4.6 Comparison of failure load with varying pitch for a glass-fibre epoxy material

Figure 4.6 shows that by increasing the pitch (P) between the pins has an effect on the strength of a joint. A minimum P/d ratio of 4 is recommended to achieve the maximum strength of a joint. Figure 4.6 indicates that the design method described gives reasonable correlation for this minimum pitch to diameter of hole ratio.

Figure 4.7 shows that the design method predicts the strength of a multi pinned joint with three or more fasteners accurately, but must not be used to predict a joint of only two holes as the failure mode is in bearing. For bearing failures Equation 4.1 should be used:

$$P = dt\sigma_b \quad (4.1)$$

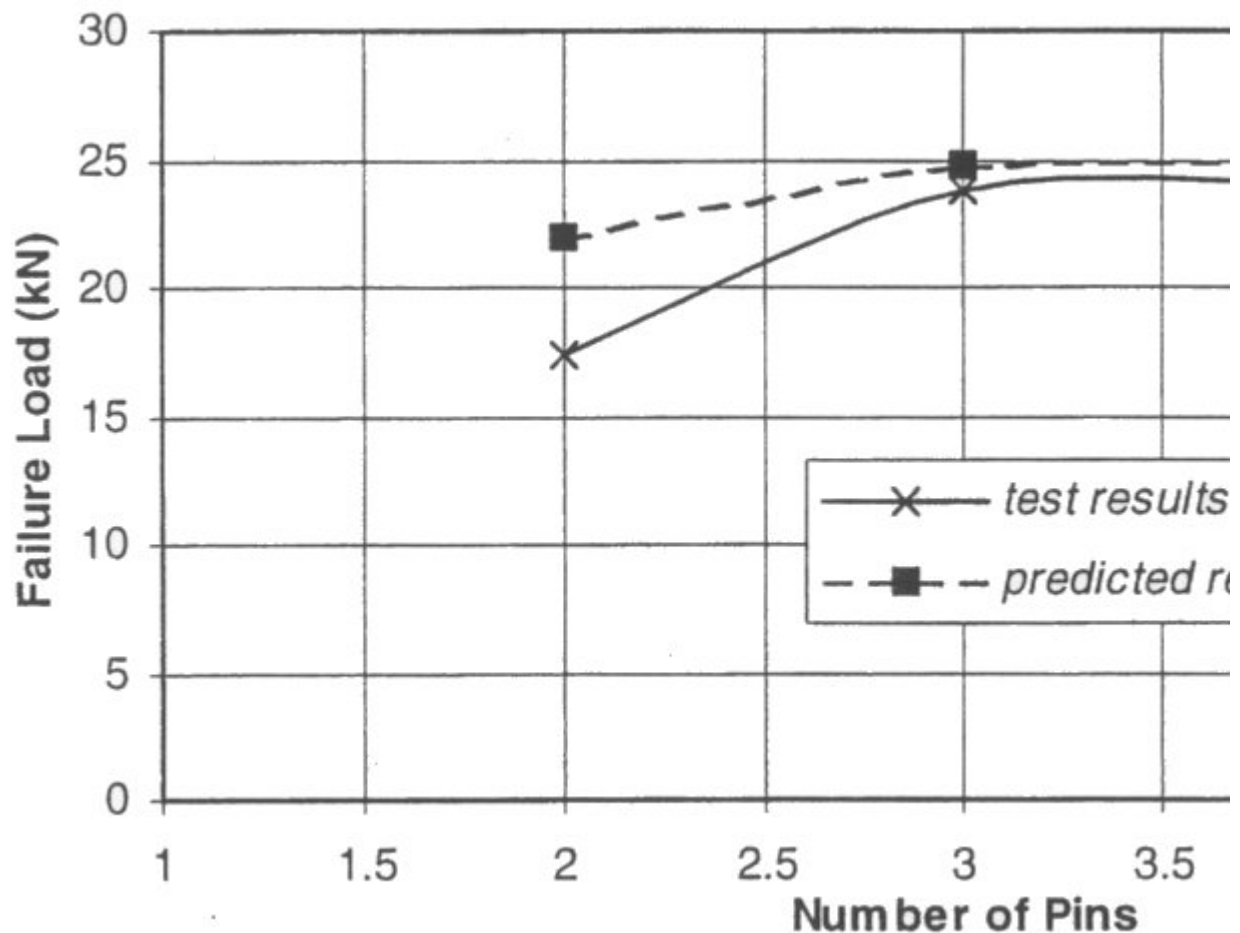


Figure 4.7 Predicted and actual failure load (kN) with varying number of pins for a woven glass-fibre epoxy

Figure 4.8 shows the effect of fastener spacing in a multi array joint it was found that the minimum side distance for an effective joint was found to be 5d.

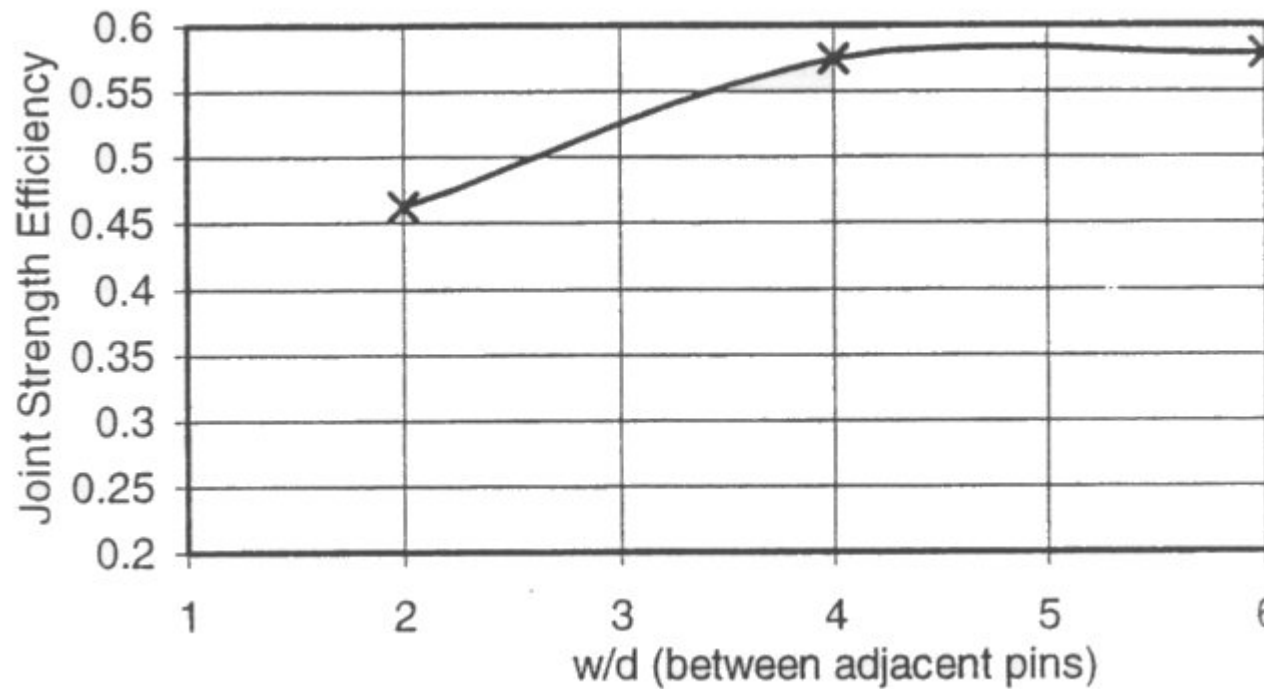


Figure 4.8 Efficiency of a joint with respect to side distance between pins

Further tests were carried out to validate the design method described on multi pin arrays. Tables 4.1-4.3 show the predicted to actual failure values and modes. As can be seen from the [Table 4.1](#) the strongest joints predicted and tested for the woven glass epoxy specimens was the 3/3/3 pin configuration (three pins per row and three rows). For the pultruded specimens ([Figure 4.2](#)), again the 3/3/3 configuration was found to be the strongest both in testing and in prediction. For the carbon-fibre epoxy specimens ([Figure 4.3](#)) the 3/3/2 was found to be the strongest with the 2/3/3 and 3/3/3 combinations closely following. Because of these results the recommended configuration for a multi fastener array would be the 3/3/3 as shown in [Figure 5.1](#).

Table 4.1 Predicted and actual loads for woven glass-fibre/epoxy multi pin array

Pin Configuration	Actual Max. Load (kN)	Predicted Max. (kN)
3/3/3	49.74	42.02
2/3/3	48.23	37.46
1/2/3	41.17	26.03
3/3/2	49.09	42.02
3/2/1	43.48	41.56
2/3/2	44.27	37.46

Table 4.2 Predicted and actual loads for pultruded glass-fibre/epoxy multi pin array

Pin Configuration	Actual Max. Load (kN)	Predicted Max. Load (kN)	Actual Failure Mode	Predicted Failure Mode
3/3/3	53.84	46.35	Net tension at row 1	Net tension at row 1
2/3/3	52.31	42.83	Net Tension at row 1	Net Tension at row 1
1/2/3	41.28	32.12	Net Tension at row 1	Bearing
3/3/2	52.15	46.35	Net Tension at row 1	Net Tension at row 1
3/2/1	40.03	45.96	Bearing	Net Tension at row 1
2/3/2	47.56	42.83	Bearing	Net Tension at row 1

Table 4.3 Predicted and actual loads for quasi-isotropic carbon-fibre/epoxy multi pin array

Pin Configuration	Actual Max. Load (kN)	Predicted Max. Load (kN)	Actual Failure Mode	Predicted Failure Mode
3/3/3	76.87	63.44	Net tension at row 1	Net tension at row 1
2/3/3	77.64	54.62	Net Tension at row 1	Net Tension at row 1
1/2/3	70.44	35.87	Net Tension at row 1	Net Tension at row 1
3/3/2	80.93	63.44	Net Tension at row 1	Net Tension at row 1
3/2/1	67.64	51.05	Bearing	Bearing row 3
2/3/2	71.39	54.62	Net Tension at row 1	Net Tension at row 1

5 Mechanical Joint Guidelines

- In general, the best fastened joints in fibrous composites still impose a loss in strength of about half the basic material strength
- If a laminate is dominated by *d*) fibres with few transverse fibres it is most likely to fail by shearing out of the composite. The minimum edge distance used for multi directional composites should be $e/d > 3$ (see [Figure 5.1](#)) to avoid composite shear out.
- A minimum of 40% of $\pm 45^\circ$ plies should be used with a minimum 10% of 90° to achieve highest bearing stresses. Optimum lay-up patterns for maximised fastener strength are found when a quasi-isotropic lay-up is used.
- Net tension failure ([Figure 3.3](#)) is influenced by the tensile strength of the fibres at fastened joints, which is maximised when the fibre spacing is approximately four to five times the fastener diameter ([Figure 4.8](#)). Smaller spacing results in cutting to many fibres while larger spacing results in bearing failures. Minimum fastener spacing is shown in [Figure 5.1](#).
- Use two row joints where possible. The low ductility of composite materials confines most of the load transfer to the outer fasteners, if more than two rows are needed bearing lby-pass procedures such as the one in Chapters [2](#) and [3](#) should be used.
- If possible double lap joints should be used in designing composite joints as the eccentricity in the load path of single lap joints leads to a non uniform bearing load on the fasteners reducing the strength of the joint considerably.

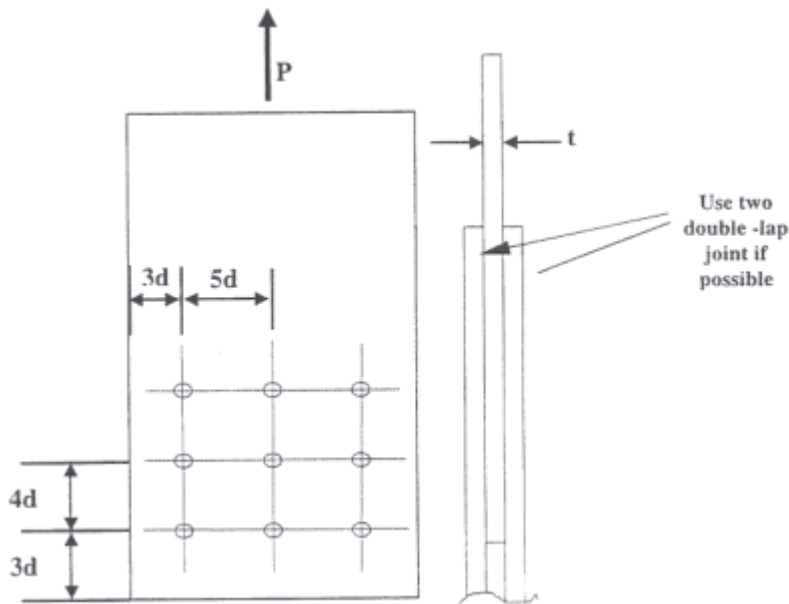


Figure 5.1 Minimum fastener spacing and edge distance

6 Conclusions

A method for the approximate determination of a multiple array joint loaded in tension has been described, based on the integration of the various steps shown in Sections 2 to 3. Comparisons with experimental results, shown in Section 4, indicated that the method can be used as an approximation for design purposes for a strip that is designed to fail in net tension. It is planned to use this approach to prepare a design module within the CoDA software [11] for bolted joints.

An overview of minimum fastener spacing has been validated in Chapter 4 and presented in Chapter 5 for the use of designing composite double lap joints.

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