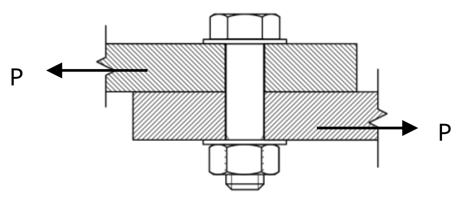
7.2. Mechanically Fastened Joints

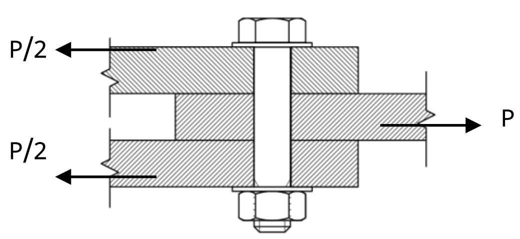
'Joint strength' considers the failure of all the components and all failure modes within the joint. Joint strength is determined by the first critical failure mode of the joint.

The in-plane or shear strength of a mechanically fastened joint includes consideration of the bearing failure of all items in the joint and the shear failure of the shank of the mechanical fastener. Depending on the joint configuration the in-plane strength may also include consideration of the bending failure of the shank of the mechanical fastener. If large out of plane deflections occur before failure the fastener head can pull through the plates in the joint. The critical failure load of the joint will often include a combination of all of the above effects – but we call it bearing strength and assume the propensity for failure varies proportionally to the thickness (within limits).

The calculation of the in plane strength of a joint may not include friction between items in the joint, the adhesive effects of liquid shim or fay surface sealant. *This is not because these effects do not exist but rather that there is no process control applied during manufacturing to these aspects of the joint that guarantee any level of strength.*



Single Shear



Double Shear

Figure 77: Mechanically fastened Shear (in plane) loaded joints

The out-of-plane or tension strength of the joint includes the pull out strength of the sheets, the tension strength of the fastener (the least of the minimum tension area strength of the fastener shank/threaded portion and the shear out of the threads) and the tension/pullout strength of the nut, nut-plate or collar.

Note: There is a more complex method of determining the joint tension strength where the fastener pre-tension is accounted for. However the effect of the fastener pre load on the joint strength is not considered significant in this methodology

Note: The published tension strengths for rivets are usually higher than the desirable tension load. A tension load in excess of 20% of the rivet shear strength can result in the deformation of the formed tail of the rivet and loss of joint clamp-up which significantly degrade the fatigue life of the riveted joint.

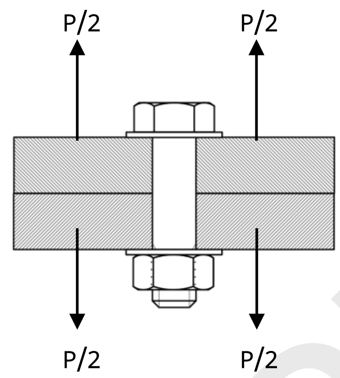


Figure 78: Mechanically fastened Tension (out of plane) loaded joints

Note: The strength of joints is subjected to the 1.5 ultimate load factor used for all structural analysis per 23.303/25.303. Joints at fittings are also subjected to an additional 1.15 fitting factor per 23.625/25.625.

Engineering judgment should be used when considering which checks to apply to a joint. If the tension load is low and is not likely to significantly affect the overall margin of safety of the joint then is it acceptable to quote the shear strength margin of safety without considering the tension load effects - and visa-versa, if the applied shear load is much less than the tension load the tension margin of safety may be quoted without considering shear load effects.

The lower the overall load magnitude is, the greater the margin and the more leeway may be used in discriminating against particular lesser load effects in order to simplify the stress analysis. The converse is also true, if the margin of safety from one load effect alone is low then the other, much smaller, load effect may have to be considered and interacted with the primary load in order to ensure the structural integrity of the joint.

The following diagram shows how the in plane strength of a mechanical joint varies with thickness of the fastener sheets for protruding head and countersunk fastener in metal and for a general fastener in a fiber composite laminate

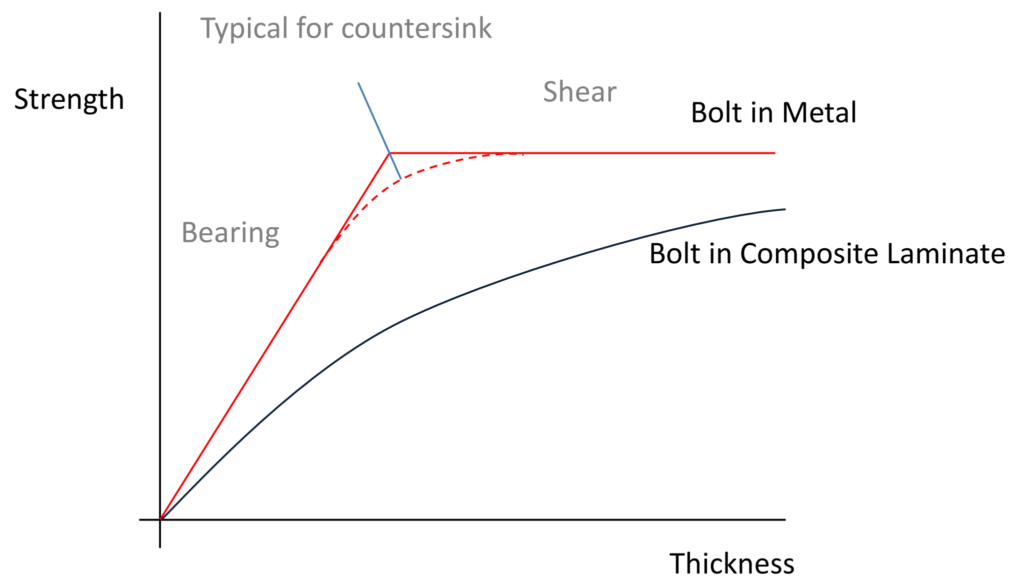


Figure 79: How in-plane Joint strength changes with thickness

- The bearing strength (for protruding head Fasteners) in metal components can be reliably calculated by the bearing area (Sheet thickness x Fastener Diameter) multiplied by the bearing strength of the sheet material (F_{bru})
- Note that shear strength portion of Figure 79 for metal joints is solely a function of the strength of the fastener shank in shear. That is the cross sectional area of the shank multiplied by its ultimate shear strength (F_{su}). This strength value is not affected by the thickness of the sheets in the joint.
- Note that in a joint between sheets of composite laminates the fastener shank shear strength is not reached. The implication of this is that in a mechanically fastened joint

between composite laminate sheets the joint failure mode will always be a failure in the composite laminate sheets. This is not always the case (it depends on the precise joint configuration: sheet thickness, sheet material, environmental condition, hole fit, fastener size and fastener material) but for typical mechanical joints in composite laminate components it is a reasonable assumption.

- It can also be seen that the failure behavior of a mechanical fastener in a composite laminate sheet is not a simple bearing failure similar to that seen in metal sheets - the failure mode can be a combination of bolt bending and local bearing failure due to non-uniform bearing stresses combined with brittle bearing failure of the composite laminate material. There is no accurate way to develop these failure loads theoretically, therefore the strength of mechanical fasteners in composite laminates must be determined by test

7.2.1. Mechanical Joints in Metal Plates

The in plane strength of mechanical joints in metal plates can be reliably predicted using available and approved strength data for the bearing strength of the sheet and the shear and tension strengths of the fasteners.

7.2.1.1. Out of Plane Strength for Mechanical Joints in Metal Plates

For metal sheets - the pullout strength is determined by calculation. the pull out strength is calculated as a 'shear-out' allowable.

The circumference of the outer diameter of the fastener head or collar/bolt (or washer if used) is calculated and combined with the thickness and shear allowable of the sheet material to determine the sheet tension allowable at the fastener. This is done for each sheet.

Sheet tension strength = Nominal fastener head outer diameter x π x sheet thickness x Fsu

$$Shear\ Through\ Strength = D \cdot \pi \cdot t \cdot F_{su}$$

Where

D	Nominal Fastener Head Outer Diameter, or Washer Diameter
t	Sheet Thickness
F_{su}	Sheet Ultimate Shear Strength

7.2.2. Mechanical Joints in Composite Plates

In general composite mechanical joint strength has to be based on a comprehensive test program. So specific strength data cannot be provided as there are many different materials and layup combinations.

However, general design guidelines are provided and analysis methodologies can be discussed.

7.2.2.1. Mechanical Joints in Composite Plates – Design Guidelines

The makeup of the laminate is critical. In general, quasi-isotropic laminate is preferred although deviations from perfect quasi-isotropy are permissible and will have limited effect on joint strength.

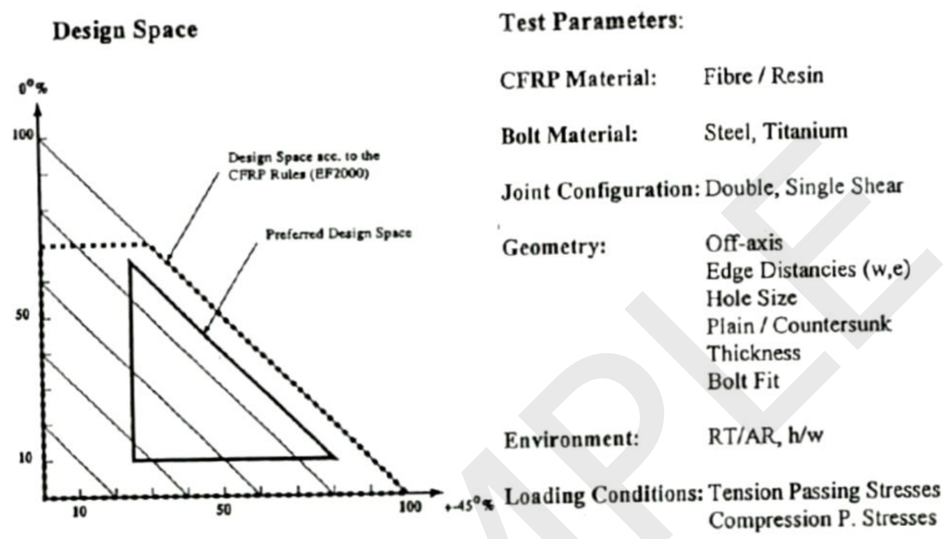


Figure 80: Lay-up Suitability for Bolt Installation (AGARD-CP-590, 1996)

There are several versions of the appropriate envelope to use for layups. The recommended layup range is given in the spreadsheet below:

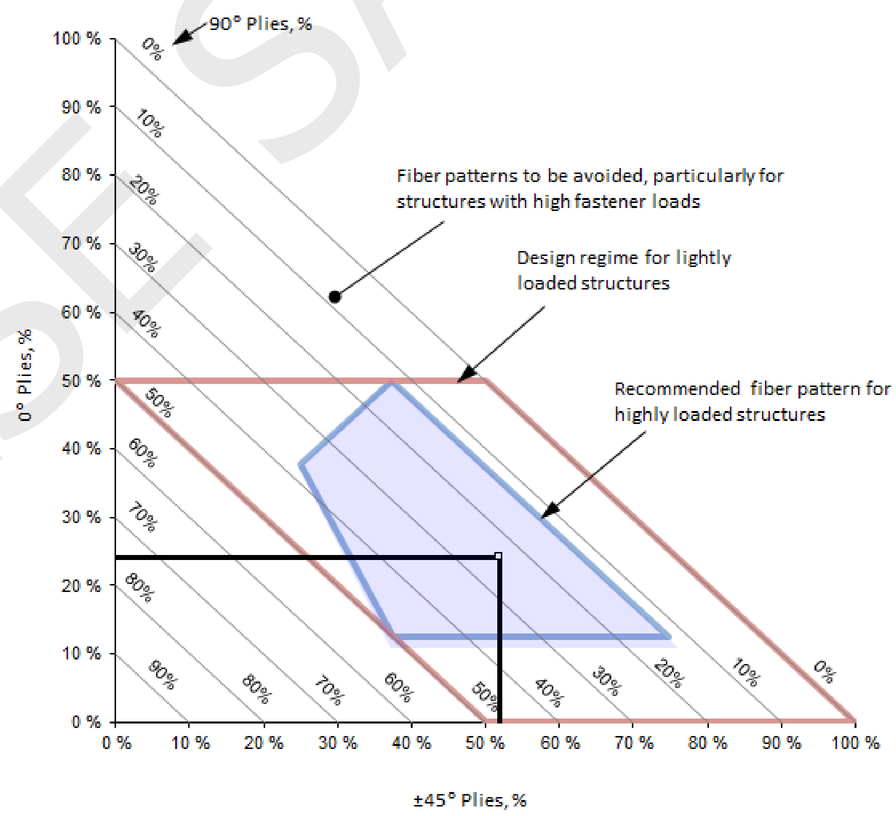


Figure 81: Recommended Lay-up Suitability for Bolt Installation

AA-SM-101-009 Layup Suitability for Bolt Installation.xlsx

Bearing failure in a composite place is progressive and non-linear, this is shown in the following figure.

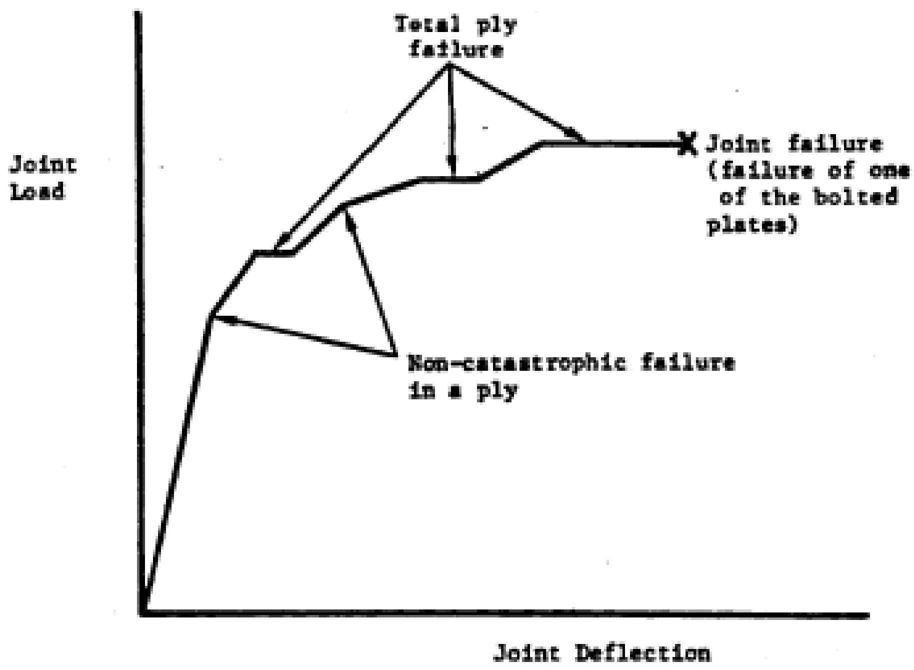


Figure 82: Failure Sequence of Mechanical Fastener in Composite Plate (AFWL-TR-86-3035, 1986)

When testing and assessing specific joint configurations it is recommended that the first failure is taken as the limit load level for the joint. This approach will demonstrate compliance with 23.305(a) and 25.305(a) *The structure must be able to support limit loads without detrimental, permanent deformation. At any load up to limit loads, the deformation may not interfere with safe operation.*

The final joint failure load level can be taken as the ultimate strength.

Design Check List:

Solid rivets and Blind rivets should not be used to react significant tension loads.

The tail diameter for solid rivets can be assumed to be 1.5x the shank diameter.

Note that aluminum (commonly used for solid rivets and blind rivets) has galvanic corrosion potential with carbon fiber and steel. Galvanic corrosion potential is increased by the presence of moisture and further increased by the presence of salt water.

Do not mix different types of fasteners (i.e. solid rivets and bolts) in the same joint.

In access panels and removable doors use a consistent grip length for all fasteners.

Avoid design that places fastener threads in bearing for joints that carry significant load. (See Section 7.2.3.2)

In both metal and laminate composites that carry significant in-service loads the countersunk depth should not exceed 70% of the sheet thickness.

The edge distance in metal plates should be 2x Fastener Shank Diameter + positional tolerance.

Where the service loads are low the fastener edge distance in metal components may be reduced to 1.5x the nominal fastener shank diameter + positional tolerance.

'Spot facing' in internal radii of machined metal components to allow for adequate fastener head/tail/collar/nut clearance is not recommended.

Fastener pitch should be no less than 4 x the nominal fastener shank diameter

Design Guidance Specific to Mechanically Fastened Laminate Composite Joints:

Note: The ideal laminate for mechanical joints is quasi-isotropic

Fastener edge distance in composite components should be a minimum of 3x the nominal fastener shank diameter with allowance for hole positional tolerance. This gives a minimum fastener edge distance in composite laminate components of 3x Diameter + .05in (1.27mm).

In special circumstances the fastener edge distance in composite laminate components may be reduced to 2.5x the nominal fastener shank diameter + .05in (1.27mm), in this case additional analysis or other substantiation is required.

Interference fits may not be used in composite components. It is recommended that clearance fit holes are used in all mechanical joints in composite laminate sheets.

When using large diameter bolts (>0.25in) in composite laminates the installation torque should be limited to avoid crushing the laminate.

To avoid galvanic corrosion recommended fastener materials for use in carbon laminate composite sheets are Titanium, A286, PHI13-8MO, Monel or PH17-4. Titanium fasteners are preferred as they are the most galvanically compatible with carbon fiber composite.

In composite to metal locations corrosion barriers like fiberglass layers must be used.

Do not buck rivets in composite structure.

Countersunk fasteners develop greater in plane strength than shear head fasteners in composite laminate joints.

In fuel containment areas joints must be sealed to be leak proof. Fasteners must also be sealed to prevent arcing within the fuel cell in the event of a lightning strike.

Fastener pitch must not to less than 4 x Shank Diameter as the interaction of the KT effect around each hole will interact and cause premature failure

Note: Best efforts should be made to follow these guidelines, deviation will require additional analysis and/or testing to validate.

7.2.2.2. In Plane Strength for Mechanical Joints in Composite Sheets

For most structures the following simple joint strength for composite laminates can be used.

If the fastener size or joint configuration is not included in the specific project approved test results, 50ksi can be used as a general bearing stress allowable for carbon fiber laminates in their worst environmental condition and 40ksi for glass fiber laminates in their worst environmental condition. As long as the t/D (thickness/Diameter) ratio is between 0.5 and 2.

Note that the in-plane strengths for composite laminate sheets for protruding head fasteners can be applied to countersunk fasteners without modification.

7.2.2.3. Out of Plane Strength for Mechanical Joints in Composite Sheets

There is no reliable analytical method to determine the pull through strength of fasteners in Laminated composites. There is some public domain data that can serve as a useful sizing guide. (NASA-TM-87603, 1985) is one of these.

The allowable data in this reference is slightly greater than that I have seen from test on several programs. Be sure to use the 1.15 fitting factor for calculations using values from this reference.

This data is developed using a 3/16in diameter 100^{o} countersunk head titanium Huck fastener.

This testing was done with laminates from .244in to .317in thick and also for 2000 and 7000 series aluminum for comparison.

This testing was done at room temperature. Environmental factors should be applied to these results.

System	Material	Laminate	Nominal Thickness, Inch
1	T300/5208	$[(\pm45/0_2)_2/\pm45/0/90]_{2S}$	0.244
2	AS4/3502	$[(\pm45/0_2)_2/\pm45/0/90]_{2S}$	0.257
3	T300/5208 (Kevlar Stitch)	$[(\pm45/0_2)_2/\pm45/0/90]_{2S}$	0.312
4	AS4/3502	$[(\pm45/\pm45)_2/+45/0_2/-45_2/0_2/+45_2/0_2/-45_2/0_2/+45_2/90_2/-45]_S$	0.298
5	Kevlar 29/3501-6	$[(\pm45/0_2)_2/\pm45/0/90]_{2S}$	0.246
6	Kevlar 49/3501-6	$[(\pm45/0_2)_2/\pm45/0/90]_{2S}$	0.256
7	Kevlar 49/SP328	$[(\pm45/\mp45)_2/90_2\pm45/\mp45/\pm45/\mp45]_S$	0.312
8	T300/CIBA-4	$[(\pm45/0_2)_2/\pm45/0/90]_{2S}$	0.317
9	AS4/PEEK (APC2)	$[+45/0/-45/90]_{6S}$	0.254
10	HSCelion/CIBA-2566	$[+45/0/-45/90]_{6S}$	0.265
11	AS4/3502/FM1000	$[+45/0/-45/90]_{5S}$@	0.309
12	Aluminum 2024-T4		0.251
13	Aluminum 7075-T651		0.256
14	AS4/3502 (0, 90) Kevlar 285 Fabric/ 5208 (±45)	$[(\pm45/0_2)_2/\pm45/0/90]_{2S}$	0.246

Table 2 Laminate Definition for Fastener Push-Through Testing (NASA-TM-87603, 1985)

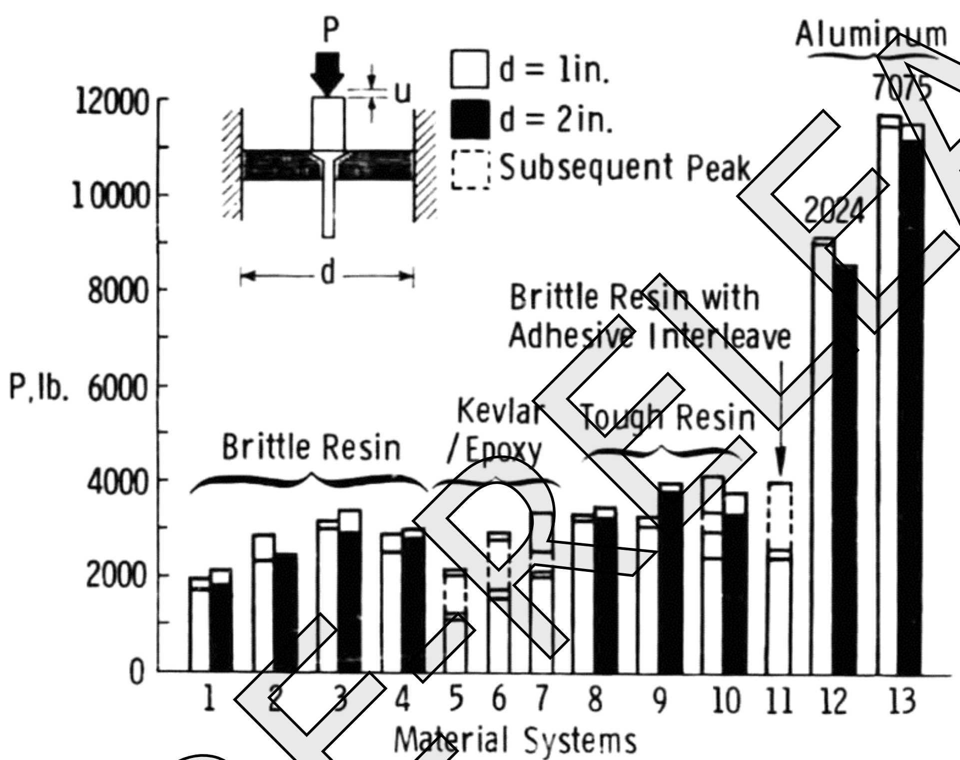


Figure 83: Results for 3/16 Fastener push through Test (NASA-TM-87603, 1985)

It is recommended that for a 3/16in countersunk fastener in 1/4in thick carbon laminate a room temperature strength of 1000lb is used.

It can be assumed that this value varies linearly with fastener diameter and sheet thickness. *However, all strength values of this nature must be confirmed by test.*

7.2.3. Mechanical Joints, Examination of the Fastener

7.2.3.1. Fastener Thread Pull-Out/Shear Out:

The fastener and nut/collar combination can fail at the minimum tension area location of the shank or thread or the threads can pull out or the threads on the collar/nut side can pull out.

The minimum tension strength for fasteners and nuts/collars is usually stated on the specification for the fastener, nut or collar. If the tension strength is not given for the fastener nut or collar, or the attachment is using a custom tapped thread in a part, the pull out strength can be calculated using (NASA-RP-1228, 1990) Page 21 'Calculating Pullout Load for Threaded bolt'

$$P = \frac{\pi\, d_m F_s L}{3}$$

Where

P Pullout Strength
d_m mean diameter of threaded hole, in (can be taken as the thread pitch diameter)
F_s Material Ultimate Shear Stress
L Thread engagement

AA-SM-005-004 Bolted Connections - 4- Bolt Thread Pull Through.xlsx

The Joint in tension should also be checked against the published tension strength of both the fastener and the nut, nut-plate or collar.

7.2.3.2. Fastener Interaction of Shear Load and Tension Load Effects

Traditional Analysis methods only cover the interaction effects of shear on the shank of the fastener and tension alone on the threaded portion. The most recent and best reference for this interaction effect can be found in (NASA-TM-2012-217454, 2012)

This analysis is valid for bolts installed in metal and composite components as it considers only the fastener in isolation

This reference examines fasteners loaded at the shank in combined shear and tension loads and also in the threaded portion in combined shear and tension loads.

All of the testing represented in this reference was on 3/8in diameter bolts and lubricant was used to minimize the potential for load transfer by friction.

The first round of testing done was to compare the ultimate combined strength of the bolt with and without preload.

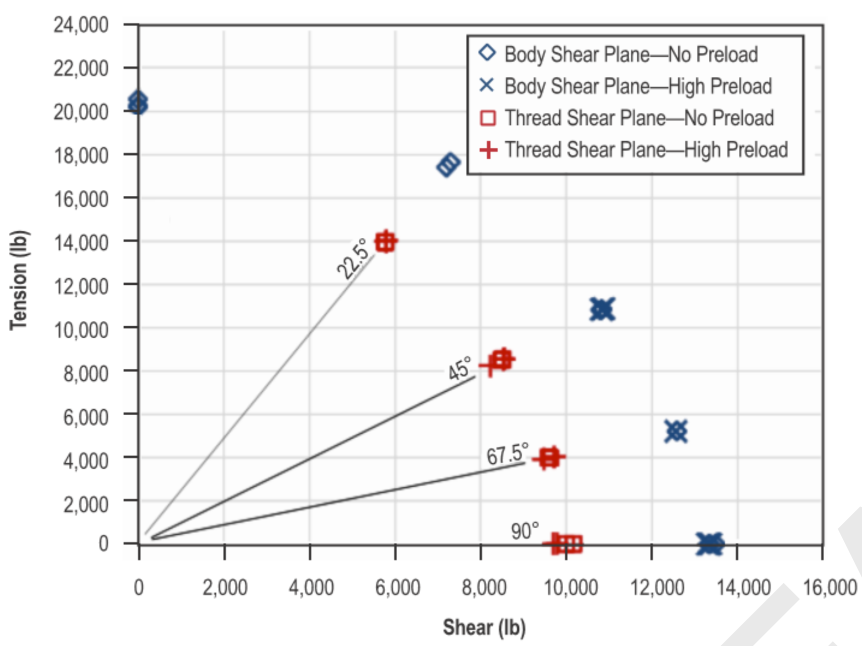


Figure 84: Bolt combined shear and tension test results, with and without preload (NASA-TM-2012-217454, 2012)

When a fastener is subjected to both Tensile and Shear loading simultaneously, the combined load must be compared with the total strength of the fasteners. Load ratios and interaction curves are used to make this comparison The load ratios are

$$R_S(\text{or } R_1) = \frac{\text{Actual shear load}}{\text{Allowable shear load}} \qquad R_T(\text{or } R_2) = \frac{\text{Actual tensile load}}{\text{Allowable tensile load}}$$

From Figure 84 two clear conclusions can be drawn

1. The presence or lack thereof of preload does not affect the ultimate strength of the joint in tension or shear.[1]
2. The strength of the joint is significantly affected if the fastener is loaded in the threaded area

The reference gives some further guidance on the definition of the interaction of shear and tension.

For fasteners loaded in the unthreaded shank (recommended for primary structure)

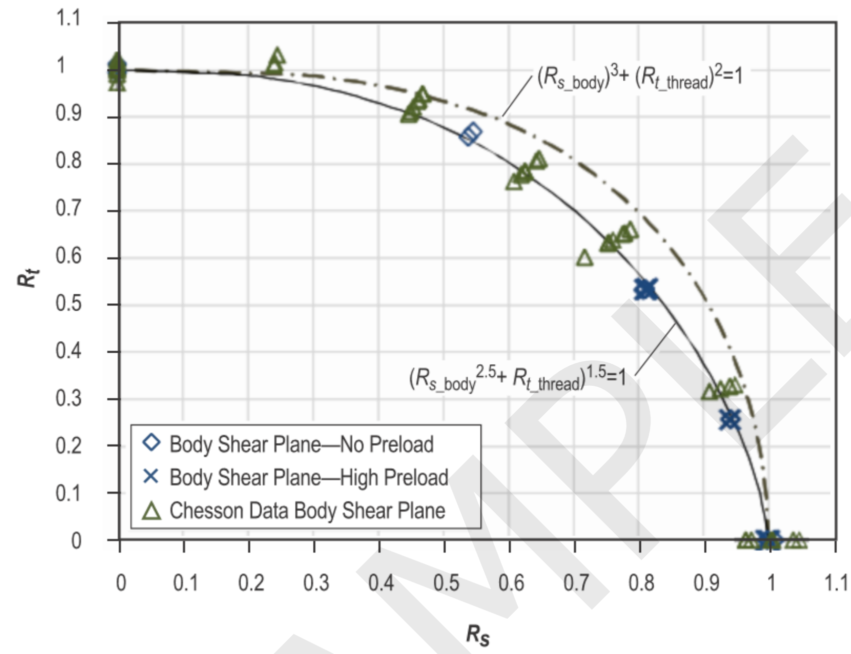


Figure 85: Interaction Curves for shear plane in the shank of the fastener (NASA-TM-2012-217454, 2012)

The interaction equation for fasteners loaded in the unthreaded shank is:

$$R_{s\_body}{}^{2.5} + R_{t\_thread}{}^{1.5} = 1$$

ABBOTT AEROSPACE SEZC LTD SPREADSHEETS ABBOTTAEROSPACE.COM

AA-SM-005-001 Bolted Connections - Combined Shear And Tension on Shank.xlsx

For fasteners loaded in the threaded portion:

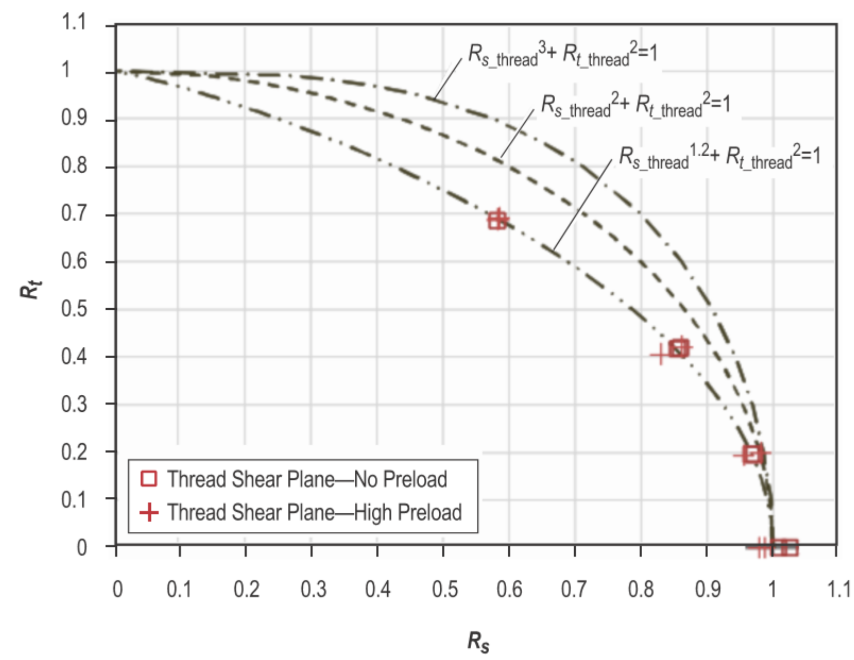


Figure 86: Interaction Curves for shear plane in the threaded portion of the fastener (NASA-TM-2012-217454, 2012)

The interaction equation for fasteners loaded in the unthreaded shank is:

$$R_{s\_thread}{}^{1.2} + R_{t\_thread}{}^{2} = 1$$

[1] The preload on a joint *is* important as it has a critical effect on the fatigue life of the joint

ABBOTT AEROSPACE SEZC LTD SPREADSHEETS ABBOTTAEROSPACE.COM

AA-SM-005-002 Bolted Connections - Combined Shear And Tension on Threads.xlsx

7.2.3.3. Fastener Bending

When very thick sheets are fastener together and non-working shims are used the offset introduced can create a bending effect on the fasteners in the joint.

These effects rarely occur in a well-designed joint using a field of fasteners. In a fastener field the bending moment caused by an offset between load paths is carried over the area of the joint and not by an individual fastener or fasteners.

For a joint that relies on a single highly loaded fastener, has very few fasteners, has unusual geometry, has a poor fastener fit (for example, a slotted hole), with low installation torque, in thick items then bolt bending may be considered, but it is almost always conservative to do so. Bolt bending is more typically considered for lug analyses, see section 7.2.6.1

7.2.4. Mechanical Joints – Tension Clip Installations

Tension clips are used when it is not possible to transfer load as shear in a fastener. There are two basic types, single angle and double angle.

- Clips are only used when the load is small, machined tension fittings should be used for applications in the primary load path
- Thin clips tend to fail in bending of the clip, thick clips tend to fail the fastener in the base of the clip due to prying action tension.
- Tension fasteners (not rivets) should be used
- Keep the bolt head as close to the radius as possible

A tension clip installation has 2 significant failure modes, failure of the clip in bending and tension failure of the fastener.

7.2.4.1. Tension Clip - Flange Bending Strength Failure

The analytical methods to determine the strength of tension clip installation has been limited to proprietary data. There is a well-known Lockheed stress memo that provides a method for the analyst and in recent years "Aircraft Stress Analysis and Sizing" by Michael Niu has given an analogous analysis method.

In the development of this book both of these methods were examined and it was determined that the curves in these two methods were not derived by test but were analytically derived.

The derivation of the tension clip strength method is below:

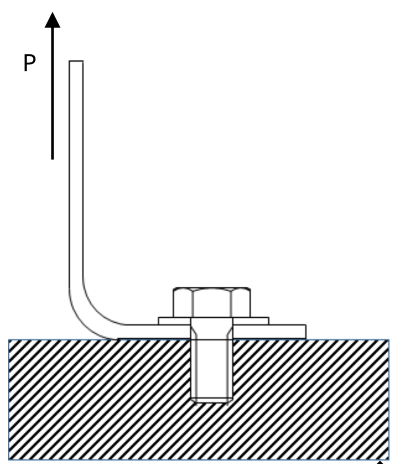
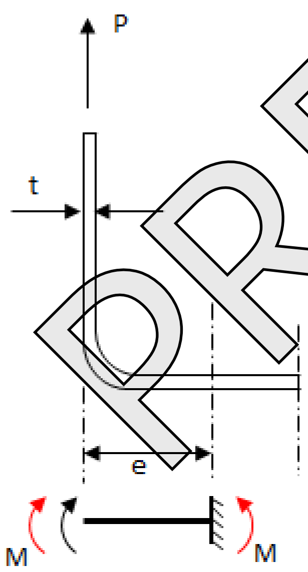


Figure 87: Tension Clip idealized installation

The methodology assumes that the clip is installed onto an effectively rigid foundation, and that the loaded outstanding flange is also retrained in rotation.

Therefore the clip can be idealized as a simple beam built at the fastener and restrained in rotation at the loaded flange:



The eccentricity of the clip (e) is the length of the idealized beam and P is the load applied at the end restrained in rotation only.

For this arrangement the critical bending moment occurs at both ends of the beam and equal to

$$M = P \cdot e / 2$$

The allowable load per one inch of angle (where the pitch of the fasteners attaching the angle to the support structure in greater than 1 inch the allowable value is assumed to be per fastener. i.e. each fastener spreads the load over 1 inch of angle length)

The 2nd moment of area for a unit length of angle flange is equal to:

$$I = 1 \cdot t^3 / 12$$

And the distance from the flange cross section neutral axis to the outer fiber

$$y = t/2$$

Taking the material yield strength F_{ty}, the allowable moment to yield is given by

$$M_0 = F_{ty} \cdot I/y$$

Therefore the allowable applied load to yield the angle is:

$$P_0 = 2 \cdot M_0/e$$

Formed Sheet Aluminum Tension Clips

The allowable load for formed sheet aluminum clips can be found with the following factors. Factor from yield allowable to ultimate allowable (general minimum for aluminum) = 1.33, the shape factor for a rectangular section is 1.5. The combination of these two factor is 2.00. The P_0 term above can be multiplied by 2.0 to give an ultimate allowable.

Figure 88 should be used for formed sheet aluminum with F_{ty} = 40,000psi. The values from this figure can be modified for other grades and tempers of aluminums in the following way.

$$P_0' = P_0 \cdot \frac{F_{ty}'}{40{,}000}$$

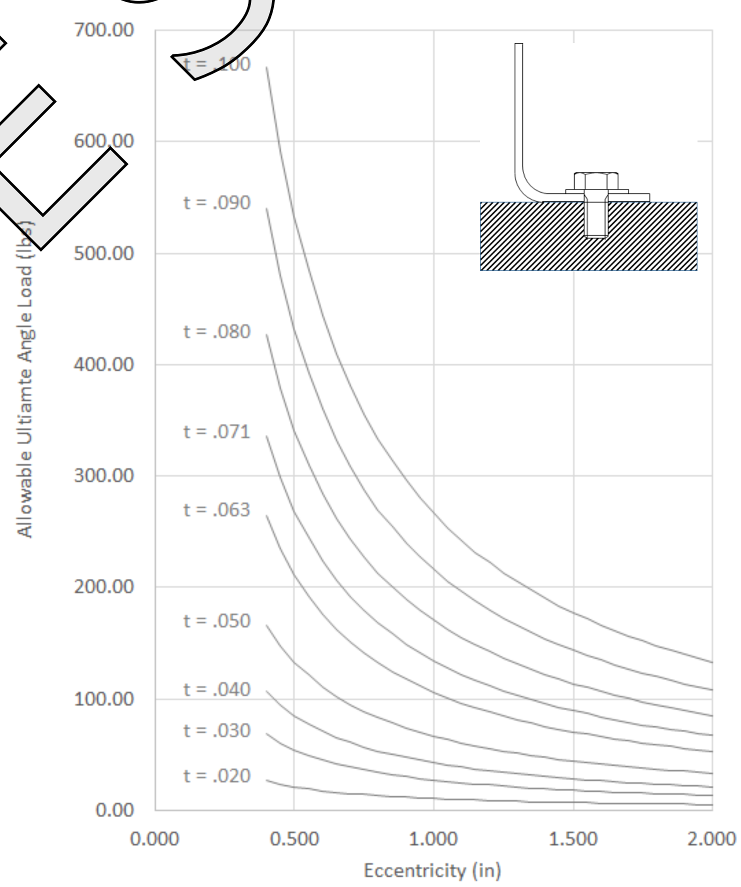


Figure 88: Allowable Ultimate Load for Formed Sheet Aluminum Tension Clip

ABBOTT AEROSPACE SEZC LTD SPREADSHEETS ABBOTTAEROSPACE.COM

AA-SM-027-003 Tension Clips - Formed Aluminum - Abbott Aerospace Method.xlsx

Extruded Aluminum Tension Clips

The allowable load for extruded aluminum clips can be found with the following factors.
Factor from yield allowable to ultimate allowable (minimum for extruded aluminum) = 1.167, the shape factor for a rectangular section is 1.5. The combination of these two factors is 1.75. The P_0 term above can be multiplied by 1.75 to give an ultimate allowable.

Figure 88 should be used for formed sheet aluminum with F_{ty} = 42,000psi. The values from this figure can be modified for other grades and tempers of aluminums in the following way.

$$P_0' = P_0 \cdot \frac{F_{ty}'}{42{,}000}$$

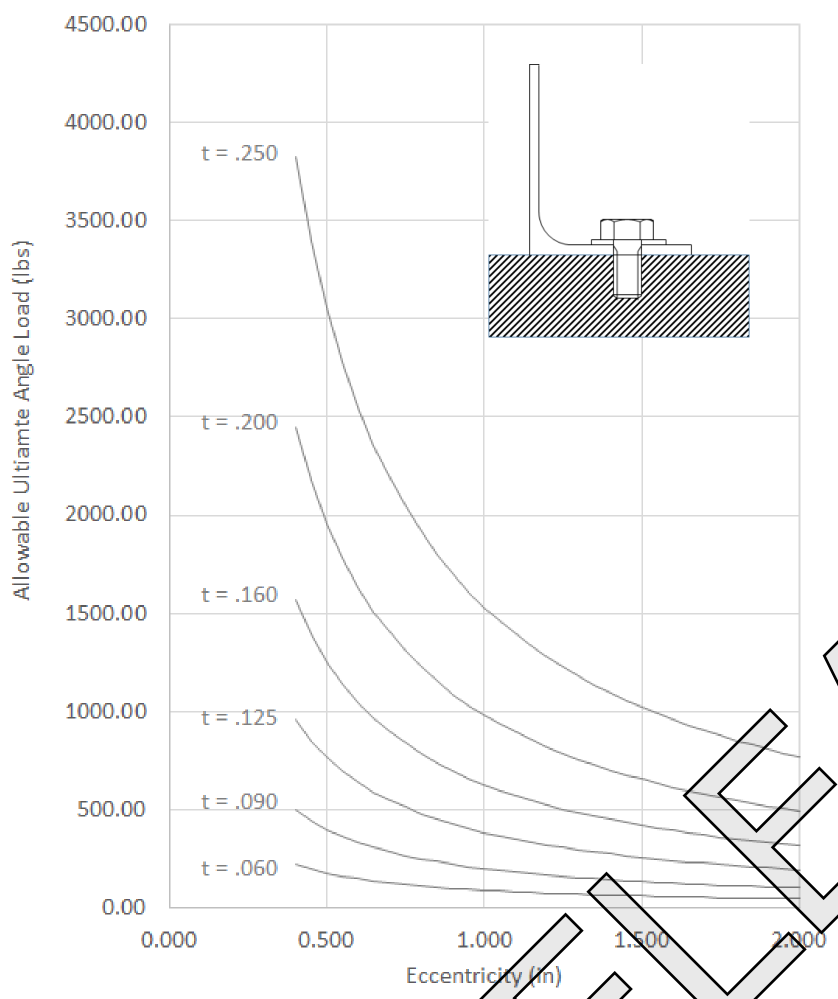
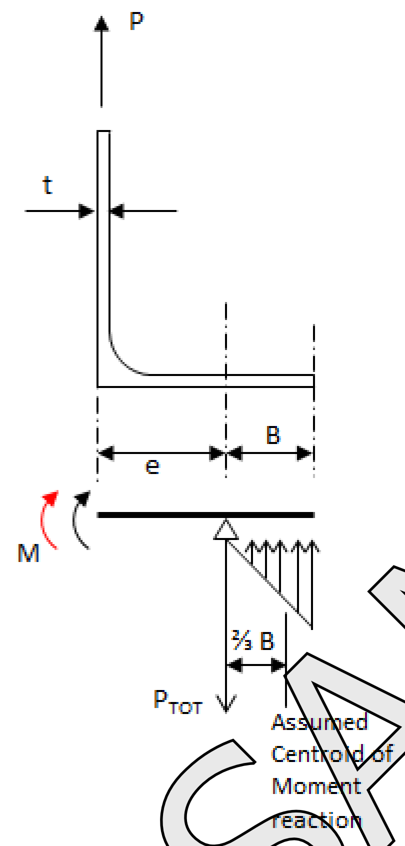


Figure 89: Allowable Ultimate Load for Extruded Aluminum Tension Clip

AA-SM-027-003 Tension Clips - Extruded Aluminum - Abbott Aerospace Method.xlsx

7.2.4.2. Tension Clip - Fastener Tension Failure

The second failure mode of a tension clip installation is a tension failure of the fastener. A fastener tension failure is only likely, for a well-designed joint, by consideration of the heel-toe effect of the fastener location and the edge of the angle flange.
In order for this load amplification effect to be significant the clip has to have a relatively high bending stiffness. This failure mode is less likely for thinner angles. However there is no way to determine (other than comprehensive testing) when this effect becomes significant so it is cautious to check this effect in every case.



The assumption of how the flange reacts moment created by the applied load is the same as for the bending check.

The moment reaction at the fastener location is replaced by a couple reaction between the fastener position and as assume triangular bearing reaction between the underside of the clip and the support structure.

Simplistically the moment is calculated at the fastener position assuming rotational fixity at the fastener position and the outstanding web.

$$M = P \cdot e / 2$$

The additional force due to the couple reaction is calculated:

$$P_{add} = M / \left(\frac{2}{3} B\right)$$

Where 2/3B is the centroid of the assumed triangular reaction force.

The total bolt load is the sum of the applied load and the additional load:

$$P_{tot} = P + P_{add}.$$

AA-SM-027-005 Tension Clips - Fastener Tension - Abbott Aerospace Method.xlsx

7.2.5. Mechanical Joints - Lugs

Most of this section can be cited to (NASA TM X-73305, 1975)

- A lug can be described as a 'single bolt fitting' - typically used to transmit large loads and provide a joint that can quickly be disconnected.
- In a typical bolted joint the hole created by the presence of the bolt does not play a significant role in the overall strength of the joint - i.e. the net section strength of the sheet item is not significantly less than the gross strength of the sheet, and in any case the tension strength of the sheet is typically not the critical measure of strength in a typical bolted joint. However in a lug the bolt hole has a significant effect on the strength of the joint.

Lug Load Nomenclature:

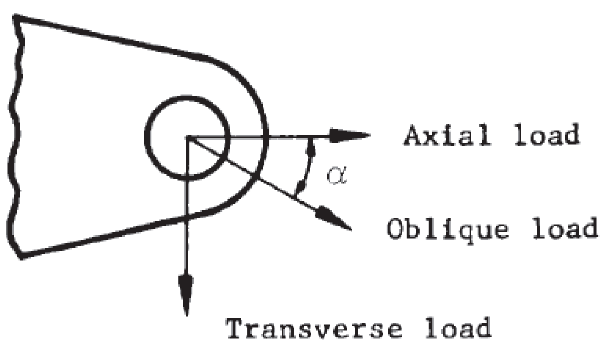


Figure 90: Lug Load direction Nomenclature

It is recommended that for all lug analyses a 10% off axis load effect is considered combined with the major load direction. This gives allowance for misalignment on installation and the effect of deflection under load of the wider structural assembly.

The lug can fail in any of the following failure modes:

- Tension across the net section
- Shear tear out or bearing
- Shear of the pin
- Bending of the pin
- Side load on the lug (checked by conventional beam method)

Lug dimension nomenclature:

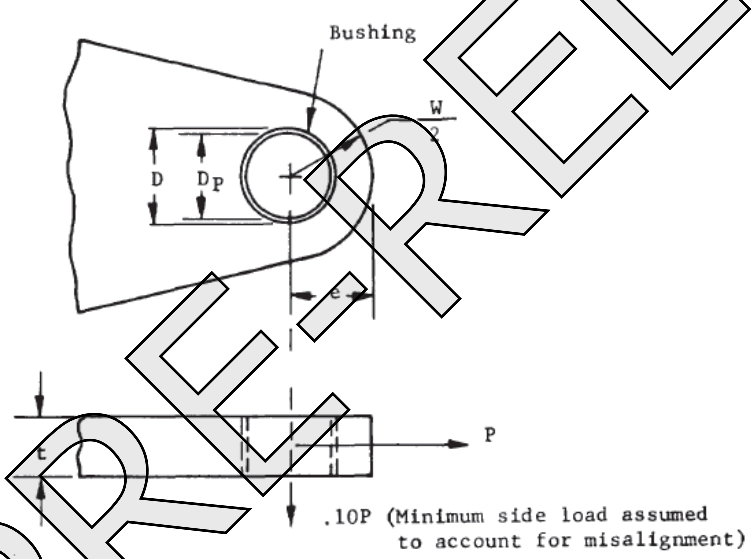


Figure 91: Lug Dimension Nomenclature

Derived Lug dimensional terms:

Bearing Area: $A_{br} = D \cdot t$

Net Tension Area: $A_t = (W - d) \cdot t$

7.2.5.1. Shear Tear out or Bearing Failure

The ultimate allowable load for Shear Bearing Failure: $P'_{bru} = K_{br} \cdot F_{tux} \cdot A_{br}$

Where F_{tux} = Ultimate tensile strength of lug material in the transverse direction.
K_{br} is taken from the following figure.

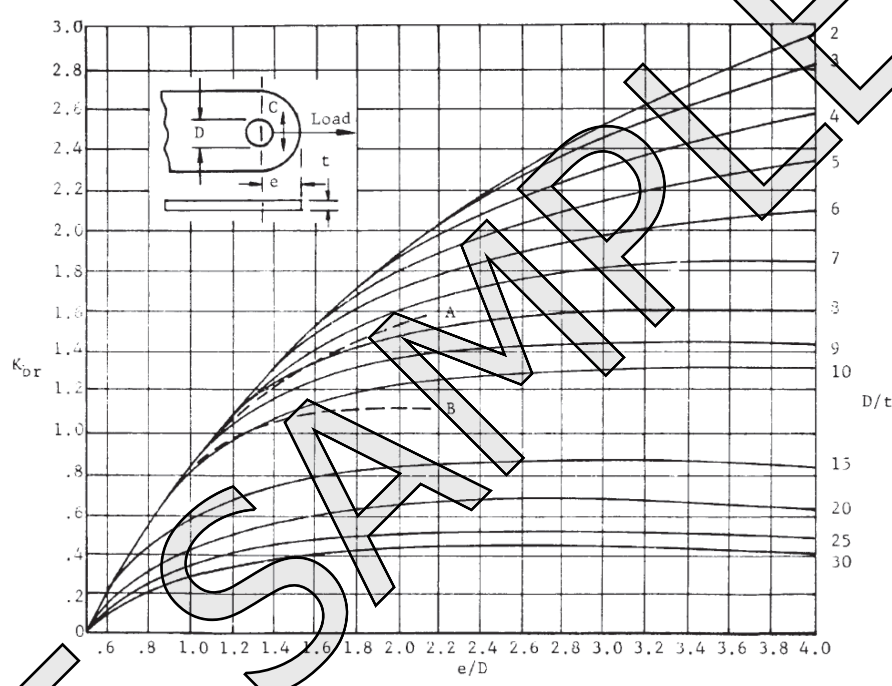


Figure 92: Shear Bearing Kt (NASA TM X-73305, 1975)

AA-SM-009-001 Lug Analysis - Shear Bearing Strength.xlsx

7.2.5.2. Tension Across the Net Section

The ultimate allowable for tension failure: $P'_{tu} = K_t \cdot F_{tu} \cdot A_t$

Where F_{tu} = Ultimate tensile strength of lug material.
K_t is taken from the following figure.

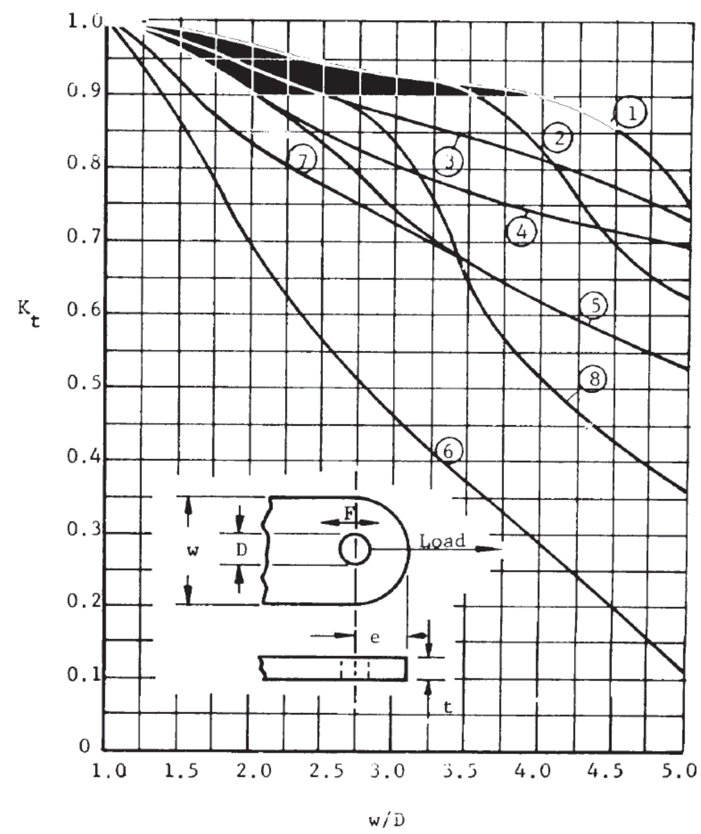


Figure 93: Axial Loading Kt (NASA TM X-73305, 1975)

Notes for Figure 93:

L = longitudinal, T = long transverse, N = short transverse (normal)

Curve 1
4130, 4140, 4340 and 8630 steel
2014-T6 and 7075-T6 plate _ 0.5 in (L,T)
7075-T6 bar and extrusion (L)
2014-T6 hand forged billet _ 144 sq. in. (L)
2014-T6 and 7075-T6 die forgings (L)

Curve 2
2014-T6 and 7075-T6 plate > 0.5 in., _ 1 in.
7075-T6 extrusion (T,N)
7075-T6 hand forged billet _ 36 sq.in. (L)
2014-T6 hand forged billet > 144 sq.in. (L)
2014-T6 hand forged billet _ 36 sq.in. (T)
2014-T6 and 7075-T6 die forgings (T)
17-4 PH, 17-7 PH-THD

Curve 3
2024-T6 plate (L,T)
2024-T4 and 2024-T42 extrusion (L,T,N)

Curve 4
2024-T4 plate (L,T), 2024-T3 plate (L,T)
2014-T6 and 7075-T6 plate > I in.(L,T)
2024-T4 bar (L,T)
7075-T6 hand forged billet > 36 sq.in. (L)
7075-T6 hand forged billet _ 16 sq.in. (T)

Curve 5
195T6, 220T4, and 356T6 aluminum alloy casting
7075-T6 hand forged billet > 16 sq.in. (T)
2014-T6 hand forged billet > 36 sq.in. (T)

Curve 6
Aluminum alloy plate, bar, hand forged billet, and die forging (N). Note: for die forgings, N direction exists only at the parting plane. 7075-T6 bar (T)

Curve 7
18-8 stainless steel, annealed

Curve 8
18-8 stainless steel, full hard, Note: for 1/4, 1/2 and 3/4 hard, interpolate between Curves 7 and 8.

AA-SM-009-002 Lug Analysis - Axial Strength.xlsx

7.2.5.3. Transverse Lug Strength

The ultimate allowable for transverse failure: $P'_{tru} = K_{tru} \cdot F_{tux} \cdot A_{br}$

The transverse strength of the lug depends on the shape parameter of the lug. This parameter is expressed as:

Shape parameter =

$$\frac{A_{av}}{A_{br}}$$

Where
A_{br} is the bearing area – Dt
A_{av} is the weighted average area given by

$$A_{av} = \frac{6}{\left(\frac{3}{A_1}\right) + \left(\frac{1}{A_2}\right) + \left(\frac{1}{A_3}\right) + \left(\frac{1}{A_4}\right)}$$

The areas A_1, A_2, A_3 and A_4 are defined as:

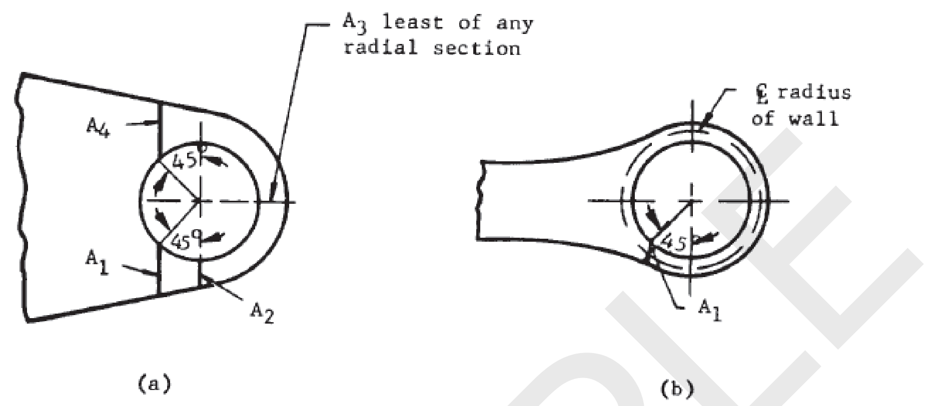


Figure 94: Idealization of typical lug for transverse load (NASA TM X-73305, 1975)

A_3 is is the least area on any radial section around the hole.
Thought should always be given to assure that the areas A_1, A_2, A_3 and A_4 adequately reflect the strength of the lug. For lugs with an unusual shape or a sudden change in cross section a conservative equivalent lug should as assumed.

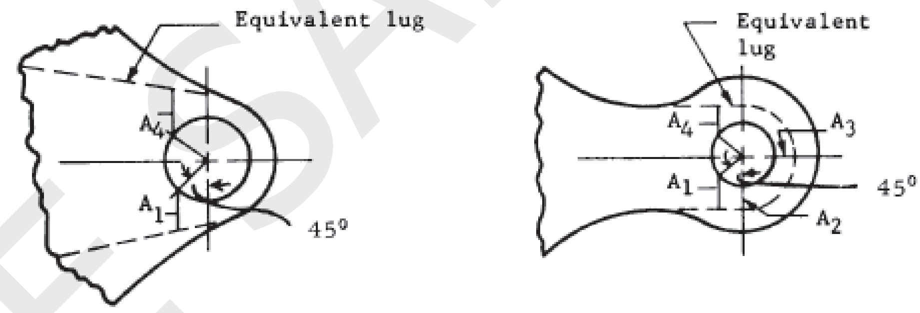


Figure 95: Idealization of unusual lugs for transverse load (NASA TM X-73305, 1975)

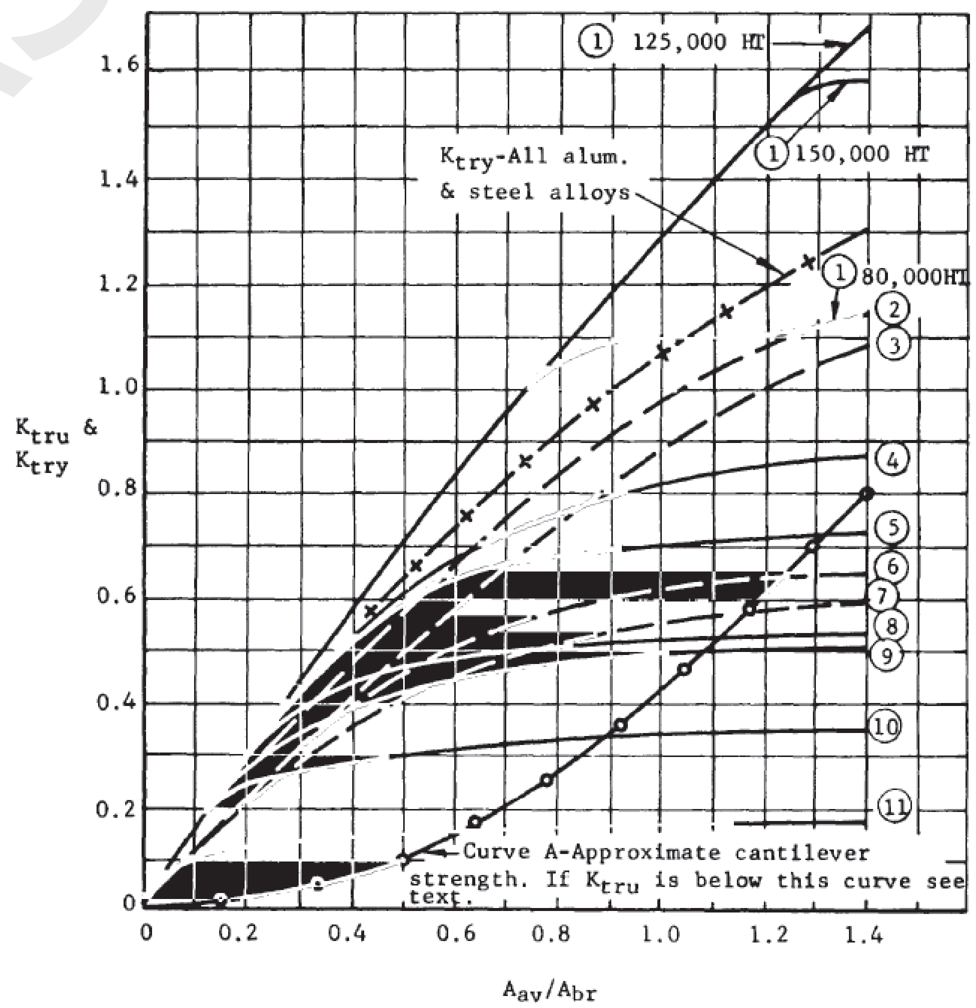


Figure 96: Transverse Loading Kt (NASA TM X-73305, 1975)

Notes for Figure 96:

Curve 1:
4130, 4140, 4340, and 8630 steels, heat treatment as noted. - Curve (a) for 125Ksi HT, Curve (b) for 150Ksi HT, Curve (c) for 80Ksi HT
Curve 2:
2024-T4 and 2024-T3 plate =< 0.5 in.
Curve 3:
220-T4 aluminum alloy casting
Curve 4 :
17-7 PH (THD)
Curve 5 :
2014-T6 and 7075-T6 plate =< 0.5 in.
Curve 6:
2024-T3 and 2024-T4 plate >0.5 in., 2024-T4 bar
Curve 7:
195-T6 and 356-T6 aluminum alloy casting
Curve 8:
2014-T6 and 7075-T6 plate>0.5 in.,=<1 in.
7075-T6 extrusion
2014-T6 hand forged billets =< 36 sq. in.
2014-T6 and 7075-T6 die forgings
Curve 9:
2024-T6 plate
2024-T4 and 2024-T42 extrusion
Curve 10:
2014-T6 and 7075-T6 plate > 1 in.
7075-T6 hand forged billet =<16 sq. in.
Curve 11:
7075-T6 hand forged billet >16 sq. in.
2014-T6 hand forged billet >36 sq. in.

AA-SM-009-003 Lug Analysis - Transverse Strength.xlsx

7.2.5.4. Lugs – Combined Axial and Transverse (Oblique) Load

In analyzing a lug the load applied should be broken into axial and transverse components (denoted by subscripts "a" and "tr" respectively) relative to the idealized lug. The two separate cases should be analyzed and the results combined using the interaction equation:

$$R_a^{1.6} + R_{tr}^{1.6} = 1.$$

Where:

$$R_a = \frac{P_a}{\min(P'_{bru}, P'_{tu})}$$

And

$$R_{tr} = \frac{P_{tr}}{P'_{tru}}$$

The Margins of Safety is calculated as follows:

$$MS = \frac{1}{[R_a^{1.6} + R_{tr}^{1.6}]^{0.625}} - 1$$

AA-SM-009-005 Lug Analysis - Compact Oblique.xlsx

7.2.6. Mechanical Joints - Lugs - Additional checks

7.2.6.1. Pin Bending

The pin used in the lug joint should be checked for pin bending. To obtain the effective moment arm of the pin compute the following for the inner lug

$$r = \left[\left(\frac{e}{D}\right) - \frac{1}{2}\right] \cdot \frac{D}{t_2}$$

Where e, D and t_2 are the lug edge distance, hole/pin diameter and thickness respectively defined in Figure 91.
Take the smaller of P'_{bru} and P'_{tu} for the inner lug as $(P'_u)_{min}$ and compute the following expression

$$(P'_u)_{min}/(A_{br} \cdot F_{tux})$$

Obtain the reduction factor 'γ' from the following figure:

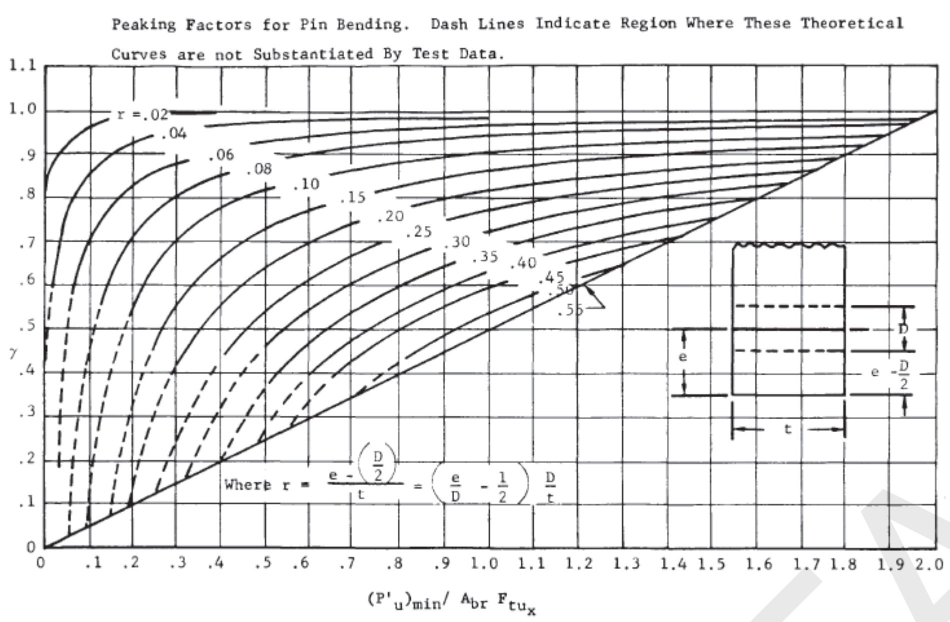


Figure 97: Peaking Factors for Pin Bending (NASA TM X-73305, 1975)

The effective moment arm can then be calculated using the following expression:

$$b = \frac{t_1}{2} + g + \gamma\left(\frac{t_1}{4}\right)$$

Where the terms in the expression are defined in the figure below

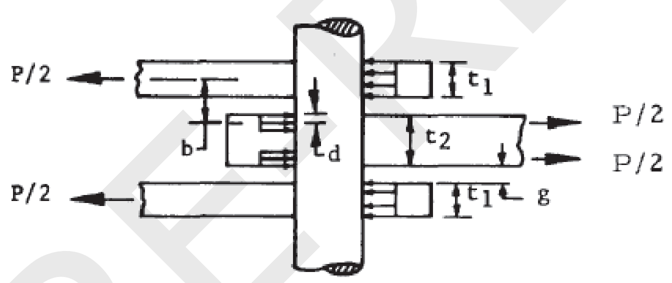


Figure 98: Parameters to calculate effective moment arm for pin bending (NASA TM X-73305, 1975)

Calculate Pin bending moment from the equation

$$M = P \cdot \left(\frac{b}{2}\right)$$

Calculate the bending stress resulting from "M" assuming the standard My/I distribution.

The resulting bending stress can be compared to the pin plastic bending allowable.

Note: A fitting factor per the regulations of at least 1.15 should be used. Some OEMs require a minimum margins of safety of 0.25 for lugs, or an effective fitting factor of 1.25.

ABBOTT AEROSPACE SEZC LTD SPREADSHEETS ABBOTTAEROSPACE.COM

AA-SM-009-004 Lug Analysis - Pin Bending.xlsx

7.2.6.2. Stresses due to Press Fit Bushings

The method in this section is referenced to (AFFDL-TR-69-42, 1986) Section 9.16. Note that several errors in the source material have been corrected. *There are errors in the source material for this section. The expression for the maximum tangential stress for the bushing: The 'p' and 'B' should be in regular font, therefore the numerator becomes '2pB²' and the denominator of this expression should read 'B²-A²'*

Pressure between a lug and a bushing assembly having negative clearance can be determined by consideration of the radial displacements. This method assumes the lug acts as if it is a uniform ring around the bushing. After assembly, the increase in inner radius of the ring (lug), plus the decrease in the outer radius of the bushing equals the difference between the radii of the bushing and ring (lug) before assembly.

$$\delta = u_{ring} - u_{bushing}$$

Where
δ = Difference between outer radius of bushing and inner radius of the ring
u = Radial displacement, positive away from the axis of the ring or bushing

Radial displacement at the inner surface of a ring subjected to internal pressure p is

$$u = \frac{D_p}{E_{ring}} \cdot \left[\frac{C^2 + D^2}{C^2 - D^2} - \mu_{ring}\right]$$

Radial displacement at the outer surface of a bushing subjected to external pressure p is

$$u = -\frac{B_p}{E_{bush}} \cdot \left[\frac{B^2 + A^2}{B^2 - A^2} - \mu_{bush}\right]$$

Where

A	Inner radius of bushing
B	Outer radius of bushing
C	Outer radius of ring (lug)
D	Inner radius of ring (lug)
E	Modulus of elasticity
μ	Poisson's ratio

Combining these equations and substituting into the first equation and solving for p gives the following expression

$$p = \frac{\delta}{\frac{D}{E_{ring}} \cdot \left(\frac{C^2 + D^2}{C^2 - D^2} + \mu_{ring}\right) + \frac{B}{E_{bush}} \cdot \left(\frac{B^2 + A^2}{B^2 - A^2} - \mu_{bush}\right)}$$

Maximum radial and tangential stresses for a ring (lug) subjected to internal pressure occur at the inner surface of the ring (lug).

Maximum radial stress for lug (the pressure on the interface between the lug and the bushing),

$$F_r = -p$$

Maximum tangential stress for lug,

$$F_t = p \cdot \left[\frac{C^2 + D^2}{C^2 - D^2}\right]$$

Positive sign indicates tension. The maximum shear stress at this point in the lug is,

$$F_s = \frac{F_t - F_r}{2}$$

The maximum radial stress for a bushing subjected to external pressure occurs at the outer surface of the bushing and is

$$F_r = -p$$

The maximum tangential stress for a busing subjected to external pressure occurs at the inner surface of the bushing and is

$$F_t = -\frac{2 \cdot p \cdot B^2}{B^2 - A^2}$$

Acceptable stress levels:

- Stress Corrosion. This maximum allowable press fit stress in magnesium alloys should not exceed 8000psi. For all aluminum alloys the maximum press fit stress should not exceed $0.50F_{ty}$.
- Static Fatigue. For steels heat treated to above 200ksi, where there is any risk of hydrogen embrittlement the press fit stress should not exceed $0.25F_{tu}$.
- Ultimate Strength. F_{tu} should not be exceeded. However, it is rare to create stresses of this magnitude in a press fit busing installation.

ABBOTT AEROSPACE SEZC LTD SPREADSHEETS ABBOTTAEROSPACE.COM

AA-SM-010 Stress Due to Interference fit bushing installation.xlsx

7.2.7. Other Mechanical Connections

7.2.7.1. Beam In a Socket Analysis

A beam in a socket type analysis is usually applicable for cantilevered (single shear) pins in fittings. The nature of these joints means that the engagement length of the pin in the 'socket' is usually some multiple of the pin diameter. This is required to reduce the peak bearing load between the pin and the socket to an acceptable level.

The method is predicated on a continuous contact between the pin and the socket and a uniform bearing load distribution between the pin and the socket. This method is reference to R. Burandt in 1959, and an expanded method (that gives essentially the same results) is defined in (NASA-CR-4608, 1994).
This method was provided to me by Bosko Zdanski in October 2013.

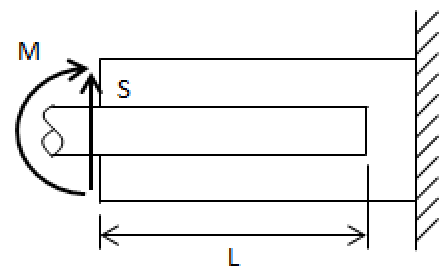


Figure 99: Beam in a Socket Configuration

Distributed socket reaction to shear load:

$$w_s = \frac{S}{L}$$

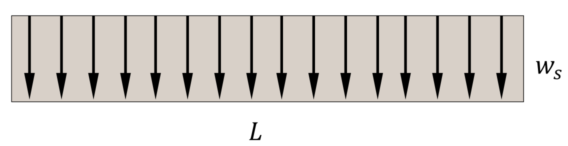


Figure 100: Beam in a Socket – Distributed Shear Load

Moment at Socket Center:

$$M_T = M + S.\frac{L}{2}$$

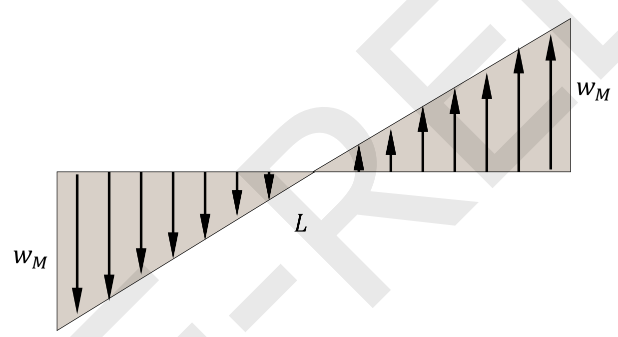


Figure 101: Beam in a Socket – Distributed Moment Load

Distributed socket reaction to Moment Load at Socket Center:

$$w_M = \frac{6}{L^2}\cdot\left(M + S\cdot\frac{L}{2}\right)$$

Socket Reaction at Outer End:

$$w_1 = w_M + w_s$$

This expands to:

$$w_1 = \frac{M}{L^2}\cdot\left(4\cdot\frac{S\cdot L}{M} + 6\right)$$

Introducing:

$$K_1 = 4\cdot\frac{S\cdot L}{M} + 6$$

The resulting expression is:

$$w_1 = K_1\cdot\frac{M}{L^2}$$

Socket reaction at bottom end:

$$w_2 = w_M - w_s$$

This expands to:

$$w_2 = \frac{M}{L^2}\cdot\left(2\cdot\frac{S\cdot L}{M} + 6\right)$$

Introducing:

$$K_2 = 2\cdot\frac{S\cdot L}{M} + 6$$

The resulting expression is:

$$w_2 = K_2\cdot\frac{M}{L^2}$$

From a linear load distribution it follows that:

$$\frac{a}{w_2} = \frac{L-a}{w_1}$$

$$a = L\cdot\frac{w_2}{w_1 + w_2}$$

Introducing:

$$K_a = \frac{w_2}{w_1 + w_2} = \frac{1 + \frac{S\cdot L}{3\cdot M}}{2 + \frac{S\cdot L}{M}}$$

The resulting expression is:

$$a = K_a\cdot L$$

Where 'a' is the distance from the 'bottom' of the socket to the point of zero shear load

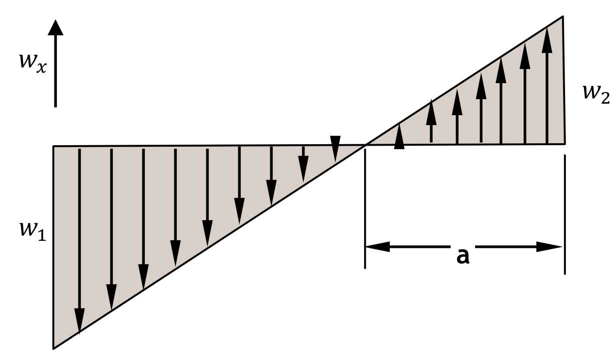


Figure 102: Beam in a Socket – Bearing Load Summation

Local distributed reaction along socket is given by:

$$w(x) = \frac{w_1 + w_2}{L}\cdot x - w_1$$

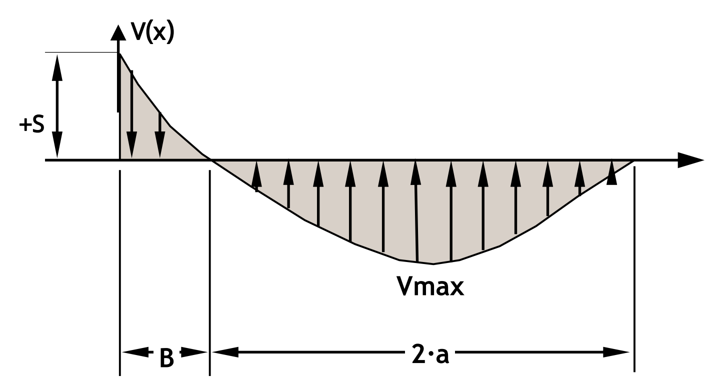


Figure 103: Beam in a Socket – Pin Internal Shear Load

Pin maximum shear load:

$$V_{MAX} = -\frac{a \cdot w_2}{2}$$

Introducing

$$K_V = \frac{K_2 \cdot K_a}{2} = \frac{2 \cdot \frac{S \cdot L}{M} + 6}{2} \cdot \frac{1 + \frac{1}{3} \cdot \frac{S \cdot L}{M}}{2 + \frac{S \cdot L}{M}}$$

The resulting expression is:

$$V_{MAX} = -K_V \cdot \frac{M}{L}$$

The point of pin zero shear load is given by:

$$B = L - 2 \cdot a = L \cdot (1 - 2 \cdot K_a)$$

Introducing:

$$K_B = 1 - 2\frac{1 + \frac{1}{3} \cdot \frac{S \cdot L}{M}}{2 + \frac{S \cdot L}{M}} = \frac{\frac{S \cdot L}{M}}{2 + \frac{S \cdot L}{M}}$$

The resulting expression is:

$$B = K_B \cdot \frac{L}{3}$$

The location 'B' is also where the maximum pin internal moment occurs

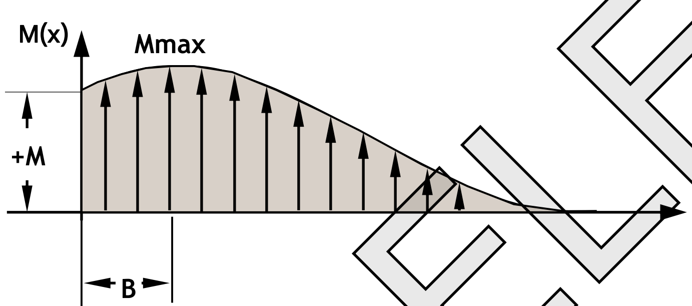


Figure 104: Beam in a Socket – Pin Internal Moment

The expression for the pin internal shear load is:

$$V(x) = S + \int_0^x w(x)dx$$

This can be expanded to:

$$V(x) = S + \frac{M}{L^2} \cdot \left[\left(3 \cdot S \cdot \frac{L}{M} + 6\right) \cdot \frac{x^2}{L} - \left(4 \cdot S \cdot \frac{L}{M} + 6\right) \cdot x\right]$$

The expression for the pin internal shear Moment is.

$$M(x) = M + \int_0^x V(x)dx$$

This can be expanded to:

$$M(x) = M + S \cdot x + \frac{M}{L^2} \cdot \left[\left(S \cdot \frac{L}{M} + 2\right) \cdot \frac{x^3}{L} - \left(2 \cdot S \cdot \frac{L}{M} + 3\right) \cdot x^2\right]$$

The maximum pin moment occurs are point 'B' coincidental with the point of zero internal pin shear:

$$M_{MAX} = M \cdot \left[1 + \frac{K_B}{3} \cdot \frac{S \cdot L}{M} + \left(\frac{K_B}{3}\right)^3 \cdot \left(\frac{S \cdot L}{M} + 2\right) - \left(\frac{K_B}{3}\right)^2 \cdot \left(2 \cdot \frac{S \cdot L}{M} + 3\right)\right]$$

This method is available in our standard spreadsheet format here:

AA-SM-003 Beam in a Socket.xlsx

7.3. General Treatment of Contact Stresses

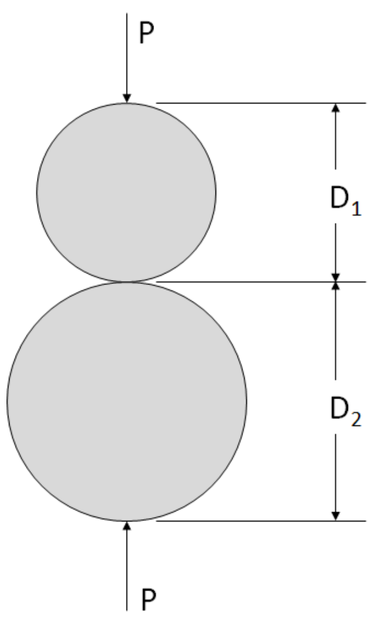
This section is largely taken from (AFFDL-TR-69-42, 1986) chapter 11. The analysis methods in this section are applicable only to isotropic materials in the elastic range. The methods are applicable for static load only and cannot be used for dynamic contact.

The stresses develop when two elastic bodies are forced together are termed bearing stresses, the stresses are localized on the surface of the material and can be high due to the small areas in contact.

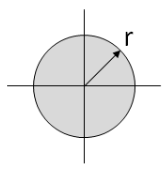
For specialized or vendor ball and roller bearings the vendor information/product specification should be consulted for allowable load levels.

7.3.1. Formulas for Stress and Deformations Due to Pressure Between Elastic Bodies

7.3.1.1. Sphere on Sphere



Shape of Contact Area:



$$r = 0.721 \cdot \sqrt[3]{P \cdot \left(\frac{D_1 \cdot D_2}{D_1 + D_2}\right) \cdot \left[\frac{1-\mu_1{}^2}{E_1} + \frac{1-\mu_2{}^2}{E_2}\right]}$$

Deflection:

$$\delta = 1.04 \cdot \sqrt[3]{\frac{P^2 \cdot (D_1 + D_2)}{D_1 \cdot D_2} \cdot \left[\frac{1-\mu_1{}^2}{E_1} + \frac{1-\mu_2{}^2}{E_2}\right]^2}$$

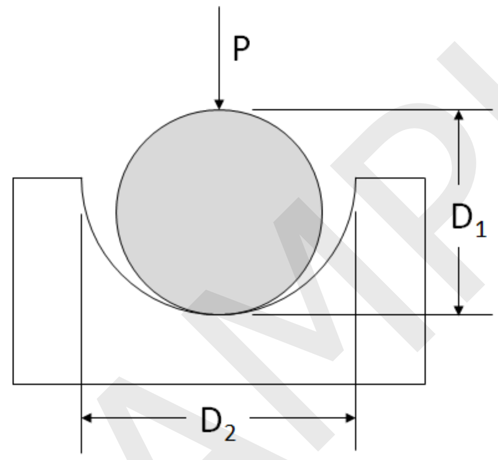
Maximum Bearing Compression Stress:

$$f_{brc} = 0.918 \cdot \sqrt[3]{\frac{P \cdot \left(\frac{D_1 - D_2}{D_1 \cdot D_2}\right)^2}{\left[\frac{1-\mu_1{}^2}{E_1} + \frac{1-\mu_2{}^2}{E_2}\right]^2}}$$

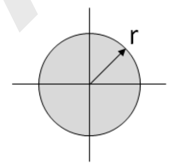
ABBOTT AEROSPACE SEZC LTD SPREADSHEETS ABBOTTAEROSPACE.COM

AA-SM-008-001 Contact Stresses - Sphere on a Sphere.xlsx

7.3.1.2. Sphere in Spherical Socket



Shape of Contact Area:



$$r = 0.721 \cdot \sqrt[3]{P \cdot \left(\frac{D_1 \cdot D_2}{D_1 - D_2}\right) \cdot \left[\frac{1-\mu_1{}^2}{E_1} + \frac{1-\mu_2{}^2}{E_2}\right]}$$

Deflection:

$$\delta = 1.04 \cdot \sqrt[3]{\frac{P^2 \cdot (D_1 - D_2)}{D_1 \cdot D_2} \cdot \left[\frac{1-\mu_1{}^2}{E_1} + \frac{1-\mu_2{}^2}{E_2}\right]^2}$$

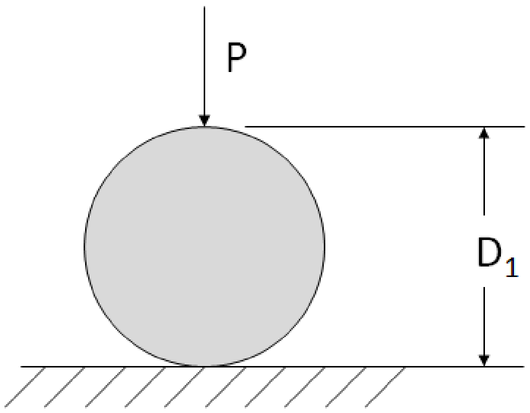
Maximum Bearing Compression Stress:

$$f_{brc} = 0.918 \cdot \sqrt[3]{\frac{P \cdot \left(\frac{D_1 + D_2}{D_1 \cdot D_2}\right)^2}{\left[\frac{1-\mu_1{}^2}{E_1} + \frac{1-\mu_2{}^2}{E_2}\right]^2}}$$

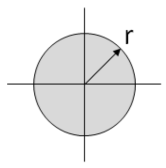
ABBOTT AEROSPACE SEZC LTD SPREADSHEETS ABBOTTAEROSPACE.COM

AA-SM-008-002 Contact Stresses - Sphere in a Spherical Socket.xlsx

7.3.1.3. Sphere on a Flat Plate



Shape of Contact Area:



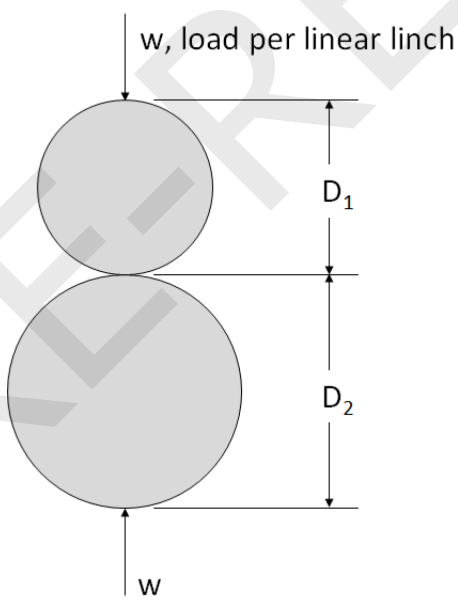
$$r = 0.721 \cdot \sqrt[3]{P \cdot D \cdot \left[\frac{1-\mu_1{}^2}{E_1} + \frac{1-\mu_2{}^2}{E_2}\right]}$$

Maximum Bearing Compression Stress:

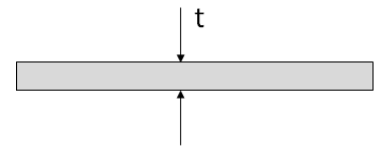
$$f_{brc} = 0.918 \cdot \sqrt[3]{\frac{P}{D^2 \cdot \left[\frac{1-\mu_1{}^2}{E_1} + \frac{1-\mu_2{}^2}{E_2}\right]^2}}$$

AA-SM-008-003 Contact Stresses - Sphere on a Flat Plate.xlsx

7.3.1.4. Cylinder on a Cylinder with Axes Parallel



Shape of Contact Area:



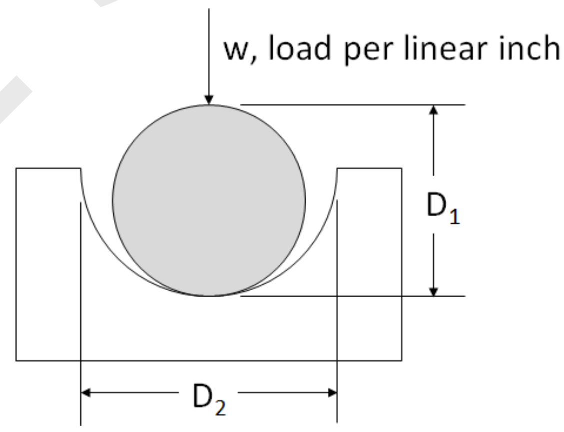
$$t = 1.6 \cdot \sqrt{\frac{w \cdot D_1 \cdot D_2}{D_1 + D_2} \cdot \left[\frac{1-\mu_1{}^2}{E_1} + \frac{1-\mu_2{}^2}{E_2}\right]}$$

Maximum Bearing Compression Stress:

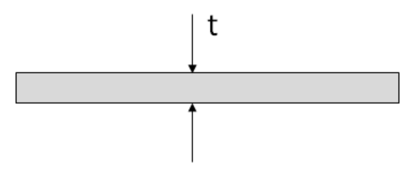
$$f_{brc} = 0.798 \cdot \sqrt{\frac{\frac{w \cdot (D_1 + D_2)}{D_1 \cdot D_2}}{\left[\frac{1-\mu_1{}^2}{E_1} + \frac{1-\mu_2{}^2}{E_2}\right]^2}}$$

AA-SM-008-004 Contact Stresses - Cylinder on a Cylinder - Parallel.xlsx

7.3.1.5. Cylinder in a Cylindrical Groove



Shape of Contact Area:



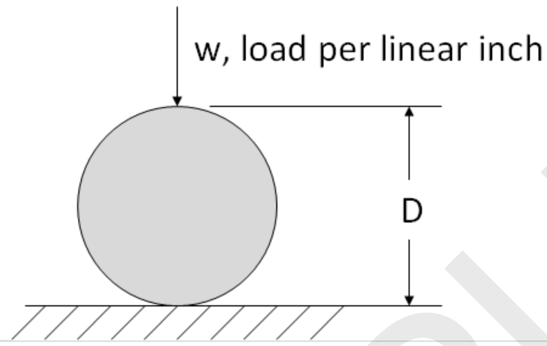
$$t = 1.6 \cdot \sqrt{\frac{w \cdot D_1 \cdot D_2}{D_1 - D_2} \cdot \left[\frac{1-\mu_1{}^2}{E_1} + \frac{1-\mu_2{}^2}{E_2}\right]}$$

Maximum Bearing Compression Stress:

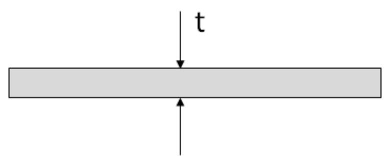
$$f_{brc} = 0.798 \cdot \sqrt{\frac{\frac{w \cdot (D_1 - D_2)}{D_1 \cdot D_2}}{\left[\frac{1-\mu_1{}^2}{E_1} + \frac{1-\mu_2{}^2}{E_2}\right]^2}}$$

AA-SM-008-005 Contact Stresses - Cylinder in a Cylindrical Groove.xlsx

7.3.1.6. Cylinder on a Flat Plate



Shape of Contact Area:



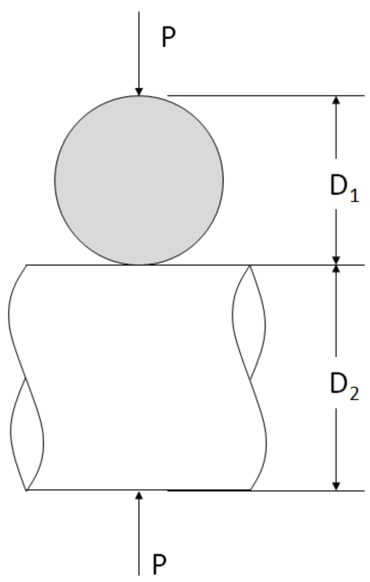
$$t = 1.6 \cdot \sqrt{w \cdot D \cdot \left[\frac{1-\mu_1^{\,2}}{E_1} + \frac{1-\mu_2^{\,2}}{E_2}\right]}$$

Maximum Bearing Compression Stress:

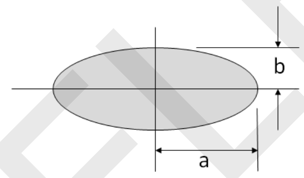
$$f_{brc} = 0.798 \cdot \sqrt{\frac{w}{D \cdot \left[\frac{1-\mu_1^{\,2}}{E_1} + \frac{1-\mu_2^{\,2}}{E_2}\right]^2}}$$

AA-SM-008-006 Contact Stresses - Cylinder on a Flat Plate.xlsx

7.3.1.7. Cylinder on a Cylinder with Axes Perpendicular



Shape of Contact Area:



The contact area between the two cylinders are derived using the following 3 parameters: K_1, K_2 and K_3.

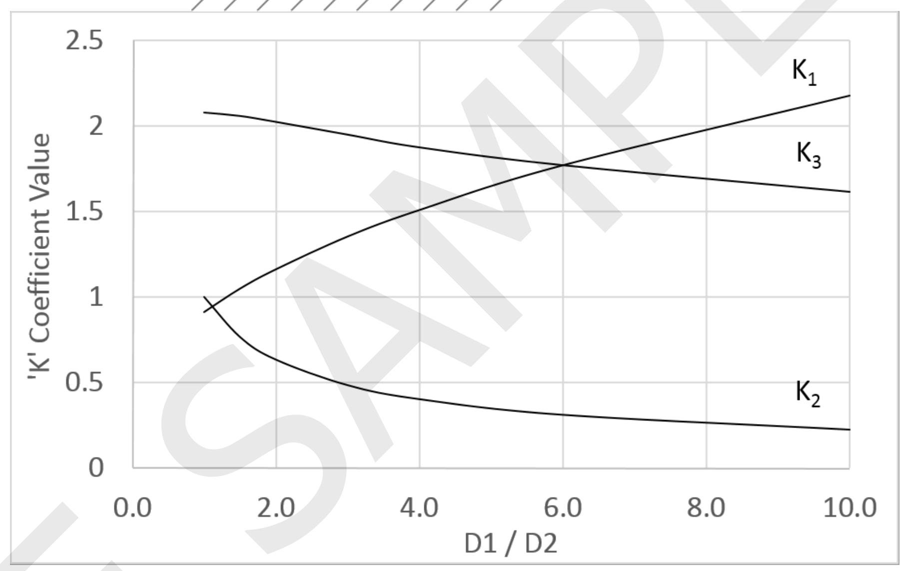


Figure 105: Contact Regions Parameters

$$a = K_1 \cdot \sqrt[3]{P \cdot \left(\frac{D_1 \cdot D_2}{D_1 + D_2}\right) \cdot \left[\frac{1-\mu_1^{\,2}}{E_1} + \frac{1-\mu_2^{\,2}}{E_2}\right]}$$

$$b = K_2 a$$

Deflection:

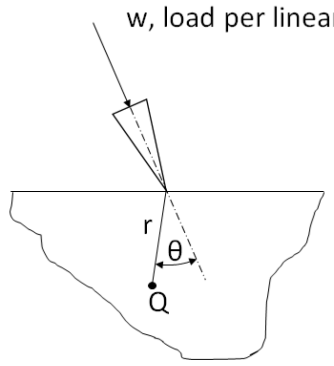
$$\delta = K_3 \cdot \sqrt[3]{\frac{(D_1 + D_2)}{D_1 \cdot D_2} \cdot \frac{P^2}{\left[\frac{E_1}{1-\mu_1^{\,2}} + \frac{E_2}{1-\mu_2^{\,2}}\right]^2}}$$

Maximum Bearing Compression Stress:

$$f_{brc} = \frac{1.5P}{\pi a b}$$

AA-SM-008-007 Contact Stresses - Cylinder on a Cylinder - Perpendicular.xlsx

7.3.1.8. Rigid Knife Edge on a Plate



Maximum Bearing Compression Stress, at any point Q:

$$f_{brc} = \frac{2 \cdot w \cdot \cos\theta}{\pi \cdot r}$$

ABBOTT AEROSPACE SEZC LTD SPREADSHEETS ABBOTTAEROSPACE.COM

AA-SM-008-008 Contact Stresses - Rigid Knife Edge on Plate.xlsx

7.3.2. Allowable Stresses for Contacts

7.4. Strength of Brazed Joints

Brazing is a joining process that is not in general use for larger structure for aerospace applications because of process cost. There is no standard method for analysis of a brazed joint, however NASA has made some efforts to create a reliable analysis methodology. This work has been under the stewardship of Dr Yuri Flom and his work forms the basis for most of this section.

The prevailing opinion is that the brazed joint (if the joint is well designed and the parent materials and the filler metal are well selected) has an equal or greater strength than the parent metal.

This level of joint strength depends on the braze being 'perfect'. The work that Flom has done at NASA covers the interaction of direct and shear load effects and gives a simple assumption to cover the likely quality variability of the brazed joint.

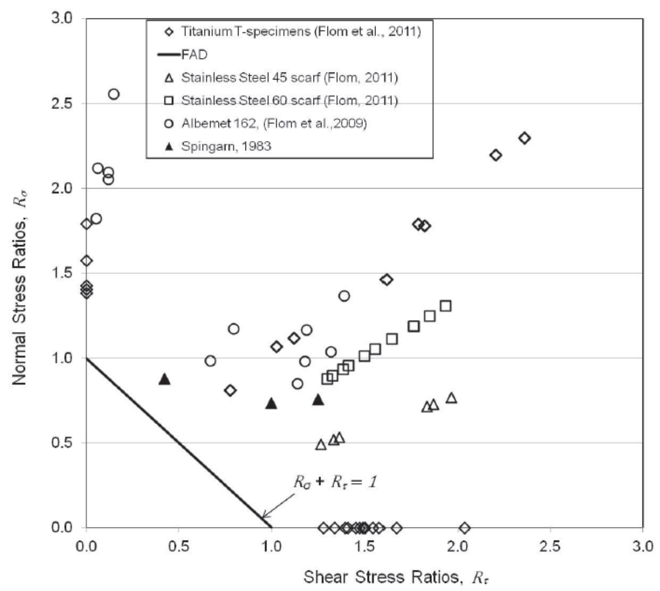


Figure 106: Combined result for studies of brazed joints under combined axial and shear loads (NASA 20120008328, 2012)

Where:

$$R_\sigma = \frac{\sigma}{\sigma_0} \text{ and } R_\tau = \frac{\tau}{\tau_0}$$

The test results used for this figure are determined by test in the following way.

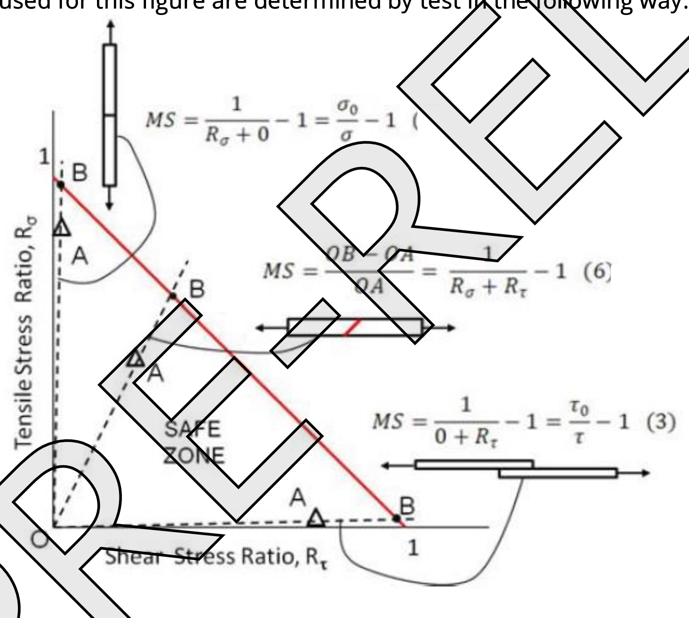


Figure 107: Graphical representation of Margin of Safety for Butt Brazed, Scarf and Lap Shear Brazed Joints (NASA 20120008328, 2012)

Determination of appropriate allowable shear strength values is given as follows. (NASA 20120008193, 2012) gives the following guidance for the relationship between brazed strength test mean values and B-basis strength.

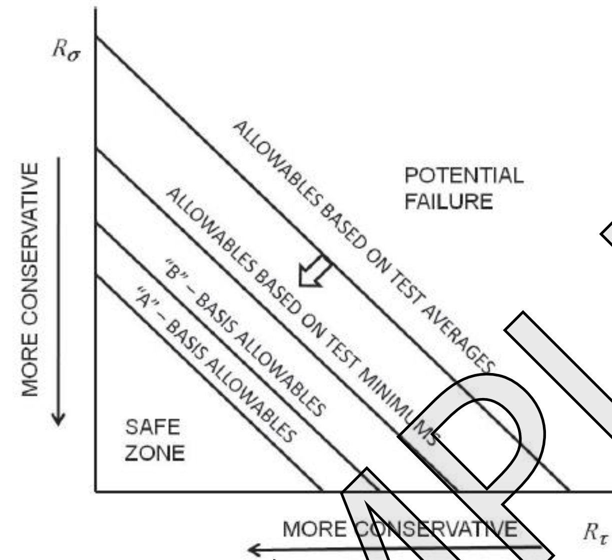


Figure 108: Relationship of average test results to statistical basis allowables for brazed joints (NASA 20120008193, 2012)

The effective B-basis brazed strength should be assumed to be 0.5 of the test mean strength for both tension strength and shear strength.

If it can be assumed that the pristine brazed joint develops the same strength of the parent material, preliminary margins of safety at ultimate load level can be generated with the following expression:

$$MS = \frac{1}{\frac{\sigma}{0.5 \cdot F_{tu}} + \frac{\tau}{0.5 \cdot F_{su}}} - 1$$

Where F_{tu} and F_{su} are the material strength of the parent material.

Note: This analysis method is approximate only. All critical joint strengths must be based on relevant test data.

ABBOTT AEROSPACE SEZC LTD SPREADSHEETS ABBOTTAEROSPACE.COM

AA-SM-020 Brazed Joints.xlsx