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Study of Behaviour of Bolted Joint in Sandwich Panels

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Abstract

Bolted joints are widely used in various kinds of aerospace and spacecraft structures. Accurate estimation of bolt forces is the prime requirement for the design and verification of joints when they are subjected to external loads. Usually, the bolts and joining components (flanges) are made of different materials having different mechanical properties like modulus, strength, surface finish qualities (friction coefficient), thermal co-efficient of expansions etc. Hence the integrity of the joint has to be maintained for thermal and external loads acting of the connecting elements and bolts. The change in the bolt force is a function of the stiffness of joining materials and type of the joint. In literature empirical equations are available for the analysis of bolted joints, but the methods are limited to the analysis of joints where flanges are clamped using a bolt and a nut. In case of spacecraft structure, the different sub-assemblies are connected to main structure using bolt and potted insert configuration. In this paper the behavior of such type of joints in the sandwich panels structures are studied.

Commonly used empirical formulae are used to analyze typical bolted joint in sandwich panels and finite element analysis were carried out to understand behavior of the joint in detail. An Axisymmetric model using 2D solid elements is generated for bolted joints with different types of inserts to understand the bolt load variations due to temperature. The behavior of such joints due to external loads is also studied using the 3D finite element models. Variation in the bolt forces of different configurations are estimated using the formulae and FE analysis are discussed in this paper.

Keywords: *Bolted joints, Axisymmetric stress model, used elements -2D solid elements, 3D HEX elements.*

Nomenclature

Symbol	Description
δ_b, δ_f, δ_w	Elongation/Deflection, mm
L_b, L_f, L_w	Length, mm
E_b, E_f, E_w	Young's moduli, N/mm^2
A_b, A_f, A_w	Cross-sectional area, mm^2
Suffices b,f,w	Bolt, flange and washer respectively
ΔT	Change in temperature, ^{o}c
ΔP	Change in preload, N
F_b	Bolt force, N
K_b	Bolt stiffness, N/mm
K_m	Member stiffness, N/mm
K_j	Total joint stiffness, N/mm
JGL	Joint Grip Length, mm
α	Frustum angle, degrees
d_b	Nominal Bolt diameter, mm
d_h	Bolt head diameter, mm
d_{avail}	Available flange diameter, mm
r	d_b/L_f
s	d_{avail}/d_w
L_j	Joint length, mm
Suffices i1,i2	Threaded insert flange and insert web
Suffices core, ipc	Insert core and insert-potting compound

1. INTRODUCTION

Bolted joints are one of the most basic elements in construction and machine design. They consist of fasteners that will capture and join other parts, and they are secured with the mating of screw threads. Typically, bolt is tensioned (preloaded) by the application of a torque to either the nut or the bolt head. The preload developed in a bolt is due to the applied torque and is a function of the bolt diameter, the geometry of the threads, length and the coefficients of friction that exist between the threads and under the bolt head or nut. The stiffness of the components joined by the bolt is not related to the preload that is developed by the torque. But the relative stiffness of the bolt and the clamped joint components, do however, determine the part of the external tension load that the bolt will carry and that in turn determines preload needed to prevent joint separation and by that means to reduce the range of stress the bolt experiences as the tension load is repeatedly applied to the joint. This determines the durability of the bolt when subjected to repeated loads. Maintaining a sufficient joint preload also prevents relative slippage of the joint components that would produce fretting wear that could result in a fatigue failure of those parts when subjected to in-plane shearing forces. [1]

The clamp load, also called as preload, of a fastener is created when a torque is applied to the bolt, and so develops a tensile preload that is generally a substantial percentage of the fastener's proof strength. In this work 75% of proof load is considered. A fastener is manufactured to various standards that define its strength and clamp load. Charts are available to identify the required torque for a fastener based on its property class or grade.

When a fastener is torqued, a tension preload develops in the bolt and a compressive preload develops in the parts being fastened. This can be modelled as a spring-like assembly that has some assumed distribution of compressive strain in the clamped joint components which will be discussed further. As the tension load is applied it relieves the compressive strains induced due to preload, hence the preload acting on the compressed joint components provides the external tension load a path other than the bolt shank. As long as the forces acting on the clamped parts will not exceed the preload, the bolt is not subjected to an increase in its tensile load.

This however, is a simplified model that is only valid when the clamped parts are much stiffer than the bolt. In reality, the fastener carries a small amount of the external load even if that external load does not exceed the clamp load. When the clamped parts are less stiff than the bolt, this model breaks down and the bolt is subjected to a tension load that is the sum of the tension preload and the external tension load.

In some applications, joints are designed so that the fastener eventually fails before more expensive components, just like a fail-safe design. In this case, replacing an existing bolt with a higher strength fastener may result in equipment damage. Thus, it is usually good practice to replace old bolts with new bolts of the same strength grade.

[3] Honeycomb sandwich structures are used extensively in spacecraft structures because of the combination of excellent mechanical properties and lightness. Mechanical joining is the most important method of assembling structural elements in the aerospace industry.

Bolted joints for high responsibility applications on sandwich structures should be carefully designed. The enhanced stress intensity factor at the surrounding of the hole and the weakness of the core material, make the designing and assembly process more critical than in those based on metallic components. Structural safety should be ensured; thus the study of bolted joints in structural sandwich and composite components has received considerable attention in both scientific literature and aeronautical standards. The joint performance depends on different parameters, mainly the joint geometry, materials, clearance, friction between different elements of the joint, temperature, load path, and bolt torque.

[4]The bolted joint presents many problems in practice, this is because it is alive, it keeps changing its response to service and environmental conditions, The purpose of bolt or group of bolts in all tensile and most shear joints are to create a clamping force between two or more things, which is called as joint members. A preloaded joint must meet (as a minimum) the following three basic requirements:

a. The bolt must have adequate strength.

b. The joint must demonstrate a separation factor of safety at limit load. This usually requires that no joint separation occur.

c. The bolt must have adequate fracture and fatigue life.

2. HAND CALCULATION FORMULAE

Most studies and hand analyses start from an equation of the form

$$P_{bolt} = \varphi P_{ext} + P_{ini} + \Delta P \quad (1)$$

Where, φ is a joint stiffness ratio

P_{ext} – External load on the joint in N

P_{ini} – Initial bolt pretension in N

ΔP – change in pretension in N

[5]All of the analytic approaches implicitly assume an axisymmetric stress field. The general approach is to idealize a bolted joint into a pair of springs in parallel. One spring represents the bolt and other represents the clamped material. If an estimate can be obtained for the stiffness of the bolt and the clamped material, then externally applied axial loads can be partitioned appropriately between the two and factors of safety can be computed to determine if the joint design is sufficient.

It is generally assumed that the clamped material can be viewed as a set of springs in series and an overall stiffness of the material km, can be computed as

$$\frac{1}{k_m} = \frac{1}{k_{f1}} + \frac{1}{k_{f2}} + \frac{1}{k_{f3}} + \ldots + \frac{1}{k_{fi}} \quad (2)$$

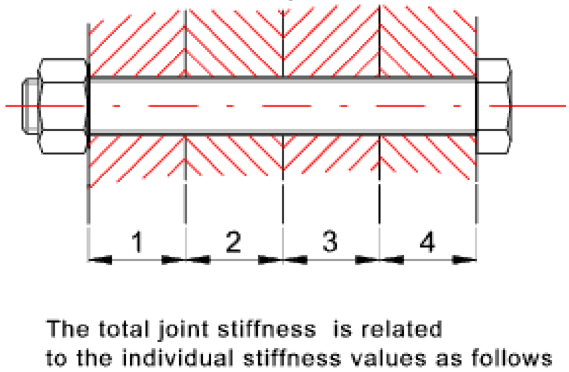


Fig. 1.1 Computation of joint stiffness

Where k_{fi} is the stiffness of i^{th} flange given by

$$k_{fi} = \frac{A_{fi} E_{fi}}{L_{fi}} \quad (3)$$

The bolt stiffness k_b can be calculated in terms of the length of the bolt L_b, young's modulus for the bolt E_b, and the cross sectional area of the bolt, A_b.

$$k_b = \frac{A_b E_b}{L_b} \quad (4)$$

The total stiffness of the joint, k_j, can be computed as (assuming two springs in parallel)

$$k_j = k_b + k_m \quad (5)$$

Now to determine the bolt and the clamped material stiffness the effective area and the effective length must be calculated. Effective area for the bolt can be calculated using the bolt diameter, but the effective area of the flange depends on the distribution of compressive stress due to the preload along the thickness of the flange. Various approaches are available in the literature to calculate the effective area of the flange.

The effective length of both the bolt and the flange also varies with the type of joint.

2. 1. Effective Area Of The Flange

The stress distribution within the material under the bolt has a complex geometry. The compressive stress in the material directly under the bolt is highest and reduces as it moves laterally away from the bolt centre line. At some lateral distance from the centre line, the compressive stress at the joint interface tends to zero and beyond that region, the joint tends to separate since it cannot sustain a tensile stress.

2.2. Cylindrical stress field approach (Q factor)

[6]In this method it is assumed the true 'barrel shaped' stress field can be approximated as a cylinder of diameter d_c (see Figure 2.1), in which dc equals Q_d.

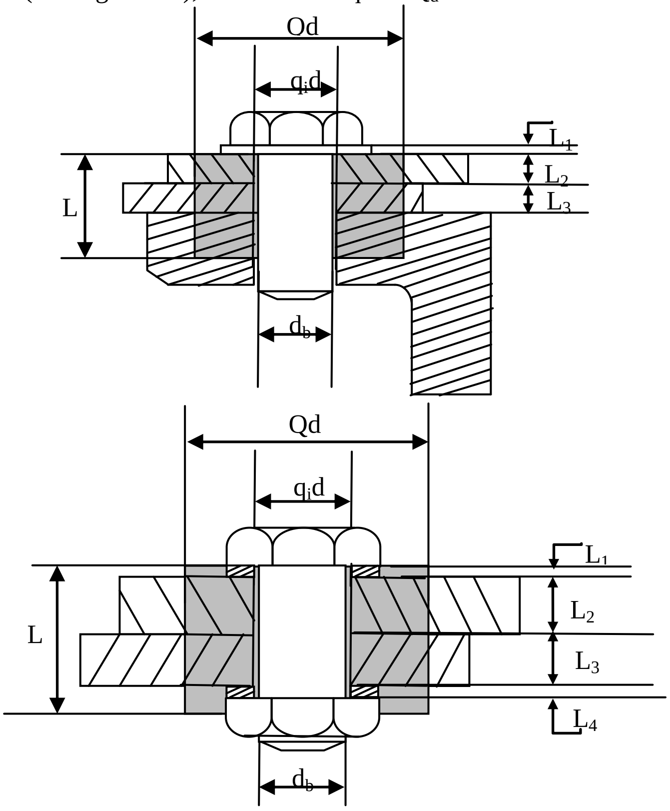


Fig. 2.1 Cylindrical stress field in the clamped flange

A factor, Q, is defined as the ratio between the actual bolt diameter and the idealized cylindrical stress field as

$$Q = \frac{d_c}{d_b} \quad (6)$$

The accuracy of this method is highly dependent on the choice of Q and generally, a value of 3 has been suggested for Q in literature. By considering the layer as a one dimensional spring, the stiffness of the i^{th} layer can be computed as,

$$k_i = \frac{A_i E_i}{L_i} \quad (7)$$

The area of i^{th} layer can be computed, assuming the inner diameter is $q_i d_b$ (where $q_i \geq 1$, and is used to allow for clearance between the clamped material and the bolt) and the outer diameter is Qd_b, as

$$A_i = \frac{\pi((Qd_b)^2 - (q_i d_b)^2)}{4} \quad (8)$$

2.3. Shigley's Frustum approach

[6]Shigley used a similar methodology but made a different assumption about the shape of the stress field to have better correlation with experimental results. In this method, the stiffness in a layer is obtained by assuming the stress field looks like a frustum of a hollow cone as shown in figure 2.2

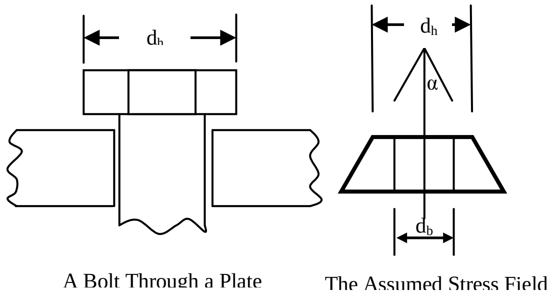


Fig. 2.2 Compressive stress distribution in the clamped flange

. By assuming an axial compression, the Stiffness of a layer can be computed as

$$k_m = \frac{\pi E_m d_b \tan\alpha}{\ln\left[\frac{(2L_j \tan\alpha + d_h - d_b)(d_h + d_b)}{(2L_j \tan\alpha + d_h + d_b)(d_h - d_b)}\right]} \quad (9)$$

Various angles, α, have been used. Usually 45 degrees is used but this often over estimates the clamping stiffness. He states that the angle typically used should be between 25 and 33 degrees and in general recommends 30 degrees (this is assuming a washer is used). There are two obvious examples when this happens. The first case is when there is not enough material for the frustum to exist (e.g., a bolt hole very near an edge of a plate). The second case is for very thick clamping areas. For this criterion, the shape of the actual stress distribution looks more like a barrel and the shape assumed by Shigley is inappropriate.

2.4. Juvinall's Approach

[5]This approach is also based on the conical compression frustum idea. However, this method involves an approximation to the volume of the frustum and assumes a half-angle of 30°. It also includes the assumption $d_h = 1.5d_b$. The formula for calculating the effective flange area in compression is given by,

$$A_f = \frac{\pi}{4}\left[\left(\frac{d_h + L_f \tan 30^\circ}{4}\right)^2 - d_b^{\,2}\right] \quad (10)$$

2.5. European Space Agency Approach

[5]This approach is from a draft document of the European Space Agency, and is considerably more complicated than above methods. It accounts for the fact that there may not be enough flange area for the compression frustum to develop fully. It also separates the compressed material into a sleeve (directly beneath the bolt head or washer) and a cone (which is truncated if the flange does not extend far enough to allow it to fully develop).

To check whether the frustum has room to fully develop, the maximum diameter of the frustum is given by,

$$d_{lim} = d_w + wL_f \tan\theta \quad (11)$$

The obtained d_{lim} is compared to the available flange diameter d_{avail}. For a nut joint, w = 1. In the case davail ≥ d_{lim}, the frustum has room to fully develop, and the flange stiffness is given by a formula identical to that of Shigley's approach except that the half-angle is a function of the joint geometry instead of a constant:

$$\tan\theta = 0.265 - 0.032\ln r + 0.153\ln s \quad (12)$$

$$A_f = \frac{\pi r L_f^2 \tan\theta}{2\ln\left(5\frac{\tan\theta + 0.5r}{\tan\theta + 2.5r}\right)} \quad (13)$$

In the above equation, the formula tan θ is a function of $r=d_b/L_f$ and $s=d_{avail}/d_w$.

If the frustum does not have room to fully develop (i.e $d_{avail}<d_{lim}$), a modified flange stiffness formula is used, as follows:

$$A_f = \frac{\pi r L_f^2 \tan\theta}{2\ln\left(5\frac{s-1}{s+1}\right) + \frac{4(\tan\theta - rs + 1.5r)}{r(s^2-1)}} \quad (14)$$

Also note that when the frustum becomes limited by the available flange area ($d_{avail}=d_{lim}$),

$$s = 1.5 + \frac{\tan\theta}{r} \quad (15)$$

and the correction term (the second term in the denominator of Equation (14) becomes zero. Also note that when there is no flange material except that directly under the washer, s = 1 and Equation (14) blows up. In this case, all of the available flange material is under compression, so the flange effective area is equal to its actual area.

2.6. Effective length of the bolt

[7]In the case of through drilled joints clamped by a nut and bolt, the clamped length of the joint and the grip length of the bolt are clearly identical. However, since the tensile stress in the bolt does not fall immediately from its full value to zero at the loaded faces of the bolt head and nut, it is usual to consider the effective length of the bolt to be slightly greater than the actual grip length. This has the beneficial effect of slightly reducing the bolt stiffness. An addition of 0.4 times the local shank diameter at each end is suitable as shown in figure 2.3

$$L_b = L_j + 0.4d_b + 0.4d_b \quad (16)$$

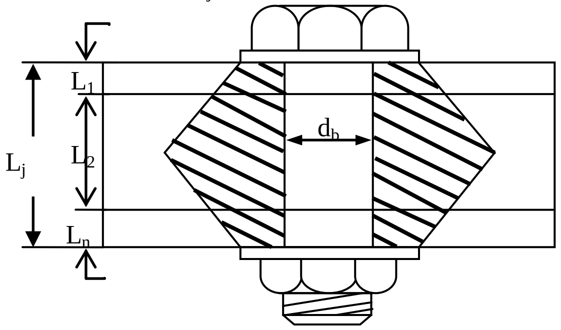


Fig. 2.3 Effective length of bolt and joint for a bolt in a through hole

ESA's approach unlike the other approaches, corrects the bolt effective length for the compliance of the parts of the fasteners outside the grip. For the design of the gripped shank plus substitution lengths of $0.4d_b$ for the head, the nut, and some portion of the shank inside the nut. Thus,

$$L_b = g + 0.4d_b + 0.4d_b + 0.4d_b\left(\frac{d_b}{d_{min}}\right)^2 \quad (17)$$

Where the bolt is screwed into a tapped hole in one of the joint members, the latter is clearly not wholly in compression. This tends to reduce the stiffness of the joint compared to the bolts. It can be observed that the effective joint length is greater than the effective bolt length, resulting in a less favourable stiffness ratio than for the case of a bolt in a through drilled hole. So, the effective lengths are

$$L_j = L_1 + d_b \quad (18)$$

$$L_b = L_1 + 0.4d_b \quad (19)$$

In case of the bolt screwed into an insert as shown in fig 2.4, only half of the insert and the flange around it, is considered to be under compression.

$$L_j = L_1 + L_2 + \left(L_n - \frac{L_i}{2}\right) \quad (20)$$

$$L_b = L_j \quad (21)$$

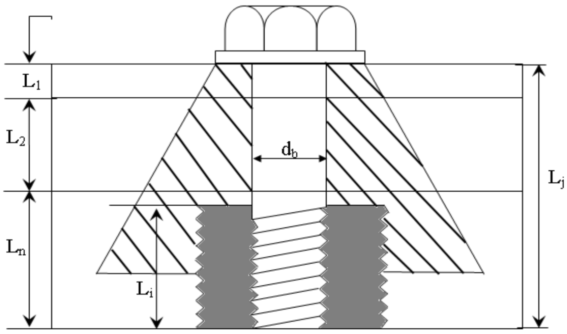


Fig. 2.4 Effective length of bolt and joint for a bolt screwed into an insert

2.7. Partitioning axial tensile load between the Joint and the Bolt

[6]Now that an estimate for the bolt stiffness, K_b, and the joint material stiffness, K_m, has been obtained, it can be examined an externally applied tensile load is partitioned between them. An applied axial load, F, will produce a displacement, δ. Part of the load will be taken up by the bolt, F_b, and part will be taken up by the clamped material, Fm. It is known that the bolt and the clamped material act as springs in parallel so we can solve for the total displacement (assuming the joint is not loaded to the point where the material is no longer clamped which is complete failure of the joint) as,

$$\delta = \frac{F}{k_b + k_m} \quad (22)$$

The stiffness constant C, of the joint is defined to be the ratio of the load taken by the bolt to that of the joint as a whole and can be computed as,

$$C = \frac{k_b}{k_b + k_m} \quad (23)$$

The part of external load that is taken up by the bolt can be computed as,

$$F_b = CF = k_b\delta \quad (24)$$

Similarly the load in the clamped material can be computed as,

$$F_m = (1-C)F = k_m\delta \quad (25)$$

3. MATERIAL PROPERTIES OF BOLTED JOINTS

Stiffness and strength are the two main criteria for selecting a material. Material with high stiffness-weight ratio and high strength to weight ratio reduce the mass of the components.

Honeycomb Sandwich construction is the most suitable for aerospace application. However, these composites have to be made from base materials. The base materials considered to make composites are either isotropic or orthotropic.

Isotropic material: Properties are uniform in all directions; the elastic constants are the same irrespective of direction and orientation. Isotropic materials are characterized by properties which are independent of direction in space.

Orthotropic material: it is a material with varied properties in mutually perpendicular directions.

In aerospace field material selection highly depends on properties to density ratios for e.g. E/ρ [specific stiffness] and σ/ρ [specific strength]. Materials should have low mass and corrosion resistant. Aluminium and Titanium Alloys are traditionally used metallic materials in aerospace field. Compared to steel aluminium alloy has low density, high bending strength and high specific stiffness. Titanium is commonly used for bolt materials.

To satisfy the mass and launch vehicle requirements, the structure must be stiff and strong yet to have a very low mass. These requirements are very difficult to achieve using solid aluminium structural members with bolted joints. Sandwich construction, however, is ideal for the structure due to its excellent stiffness-to-weight and strength-to-weight ratios. Sandwich structures are composed of two face sheets bonded to a core (Figure 3.1).

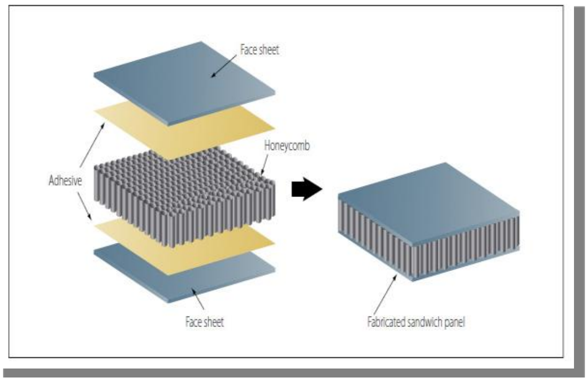


Fig. 3.1 Sandwich Constructions with Honeycomb Core

The most common aerospace core type, aluminium honeycomb, is constructed of bonded strips of aluminium foil which are then expanded to create hexagonal cells. When loaded, the face sheets of a honeycomb structure act like the flanges of an I-beam, and the core acts as the web. Face sheets are made out of aluminium alloy and core is made of sandwich construction with honeycomb core. The alloys of aluminium represent the majority of spacecraft structural materials. A combination of high stiffness to density ratio, excellent workability, non-magnetism, moderate cost, high ductility, good corrosion-resistance, and availability in numerous forms make it the best choice for most uses. Its low yield strength is the only appreciable disadvantage. It is an isotropic and honeycomb construction is orthotropic material. Sandwich panels are drilled and in these holes threaded inserts are placed. Threaded inserts are again made out of aluminium alloy. They are bonded to the sandwich deck using adhesives, which are injected through the holes present at the insert top surface. Adhesive bond is used to maximize efficiency and minimize mass; the material used for adhesive bond is araldite. Bolts are made of titanium alloy. The properties of all these materials are listed in below tables.

Table 3.1 Materials used for analysis

NAME OF THE COMPONENT	MATERIAL
Face skin	Aluminium alloy
Core	Aluminium alloy
Adhesive	Araldite
Bolts	Titanium alloy

Table 3.2 Properties of aluminum alloy (2024-T3)

Properties	Value
Young's modulus (E)	70000 MPa
Density	2.8 e-6 Kg/mm^3
Tensile yield strength	384 N/mm^2
Poisson's ratio (µ)	0.3
Co efficient of thermal expansion	9.1 e-6 /°c

Table 3.3 Properties of Titanium Alloy (Ti-6al-4v)

Properties	Value
Young's modulus	115000 N/mm^2
Density	4.63 e-6 kg/mm^3
Tensile Yield strength	880 N/mm^2
Compressive Yield strength	970 N/mm^2
Poisson's ratio	0.3
Co efficient of thermal expansion	12.6 e-6 /°c

Table 3.4 Properties of Araldite

Properties	Value
Young's modulus (E)	4000 N/mm^2
Shear Modulus	1200 N/mm^2
Density	1.6 e-6 Kg/mm^3
Tensile yield strength	26 N/mm^2
Shear strength	1 N/ mm^2
Co efficient of thermal expansion	32e-6 /°c

Table 3.5 Properties of Core

Properties	Value
Thickness	25mm
Young's Modulus in X-direction	10 N/mm^2
Young's Modulus in Y-direction	10 N/mm^2
Young's Modulus in Z-direction	310 N/mm^2
Shear Modulus in X-Y direction	10 N/mm^2
Shear Modulus in Y-Z direction	89 N/mm^2
Shear Modulus in X-Z direction	185 N/mm^2
Poisson's ratio in X-Y direction	0.3
Co efficient of thermal expansion-1,2,3	23e-6 /°c

3.1. CAD MODELING OF BOLTED JOINTS

CAD models of the different joint are built using Unigraphics(UG).

3.1.1. CONFIGURATION-1

CAD model of configuration-1 (Partial insert) is as shown below in Fig. 3.2

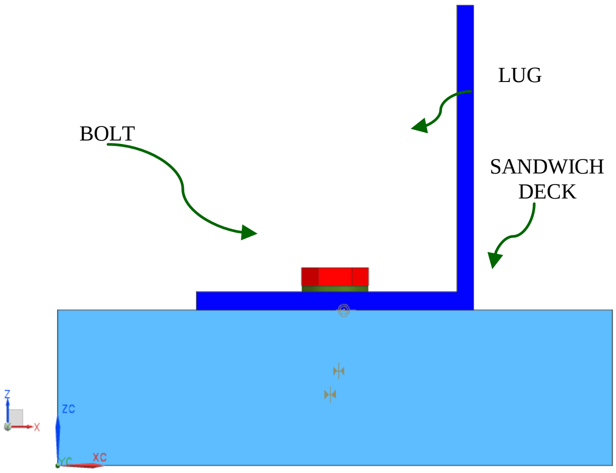


Fig. 3.2 Front view of configuration-1

Geometry:
M8 bolt
Lug -50x50x3 mm
Sandwich deck-100x100x25.5 mm

3.1.2. CONFIGURATION-2

CAD model of configuration-2 (Stepped partial insert) is given as shown below in Fig. 3.3

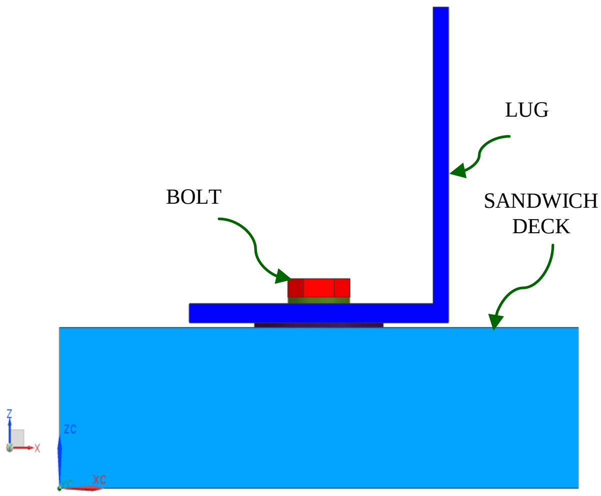


Fig. 3.3 Front view of configuration-2

Geometry:
M8 bolt
Lug -50x50x3 mm
Sandwich deck-100x100x25.5 mm

3.1.3. CONFIGURATION-3

CAD model of configuration-3 (through insert) is given below in Fig. 3.4

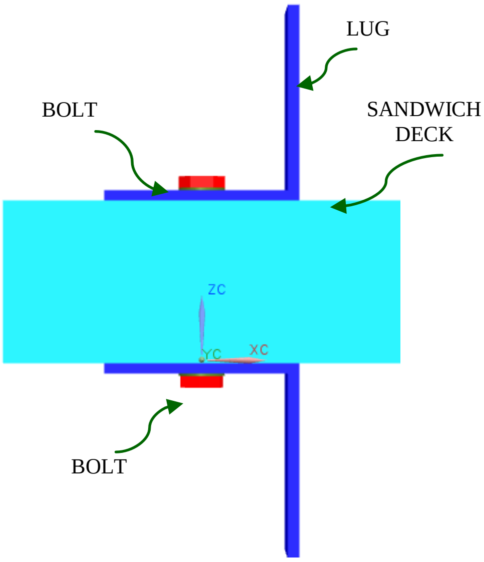


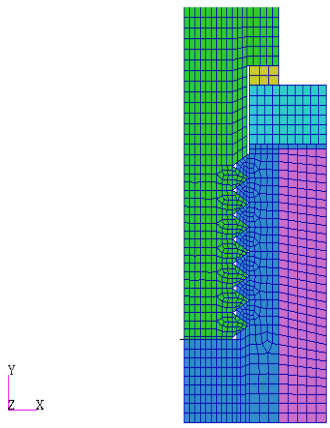
Fig. 3.4 Front view of configuration-3

Geometry:
Two M8 bolt
Two Lug -50x50x3 mm
Sandwich deck-100x100x40.5 mm

4 FEA MODEL DESCRIPTION

Bolted joints analysis is carried out by using Finite Element Method. As it is most popular and accurate method to determine the behaviour of various components in a spacecraft structure. This analysis needs a construction of a finite element model. The finite element model is built by using MSC/Patran (pre-processor and post-processor) and MSC/Nastran and MSC Marc (solver). Two types of modeling approaches are considered. The axisymmetric modeling is used to estimate the change in the bolt forces due to the temperature variations and second is 3D finite model built with 3D, HEX8 elements used to study the behavoiur of joints due to the external forces.

Finite element modelling of the components are given as below,



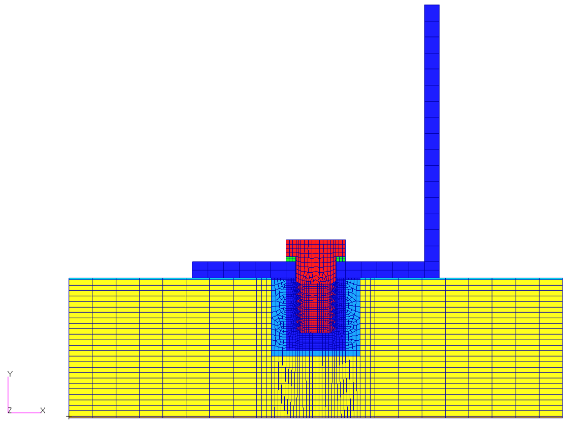
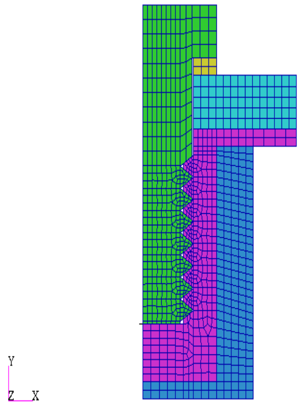


Fig. 4.1 Finite element axisymmetric and 3D models of Configuraion-1



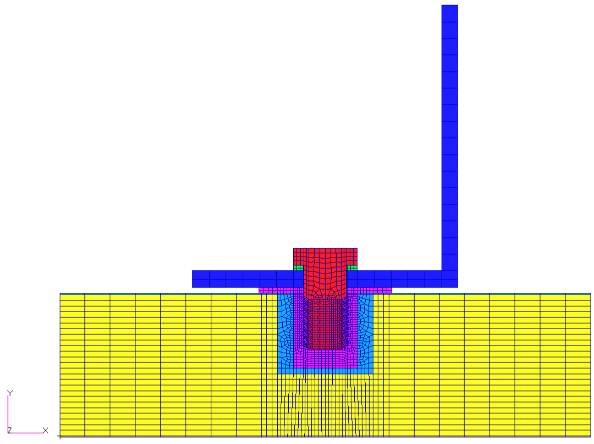
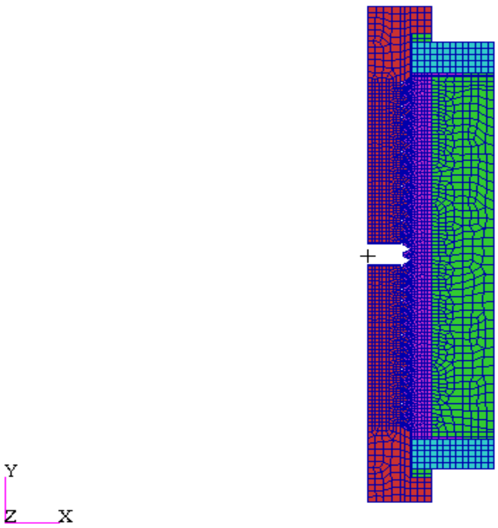


Fig. 4.2 Finite element axisymmetric and 3D models of Configuraion-2



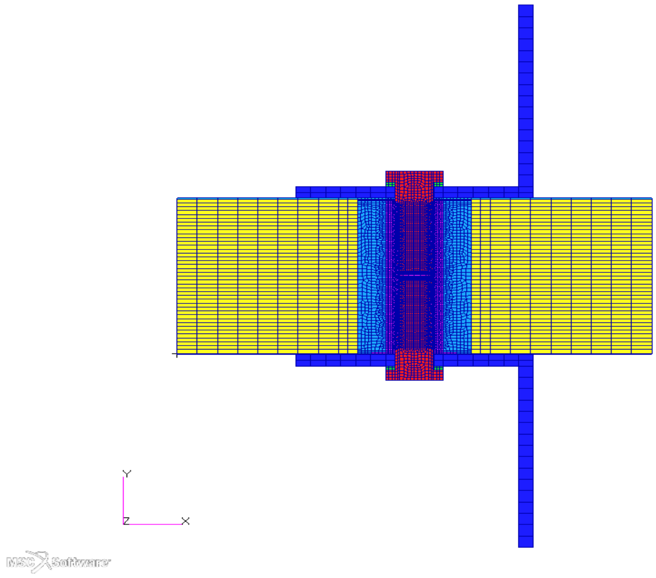
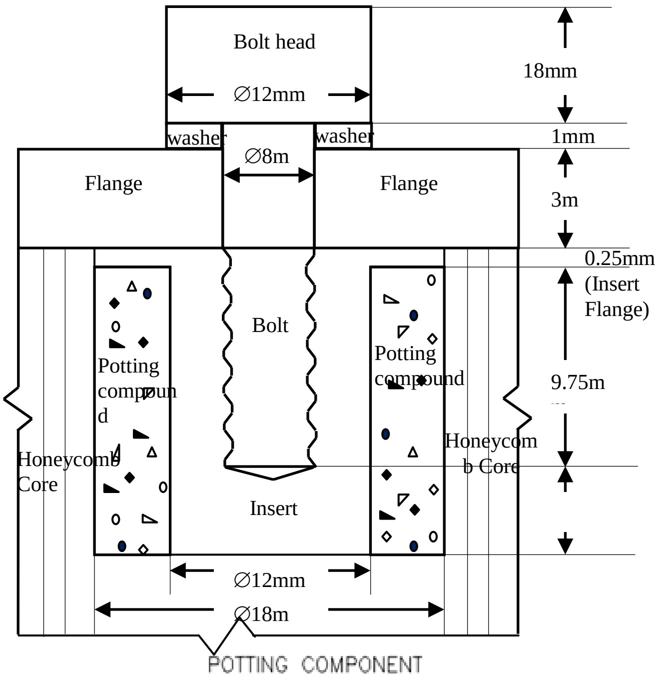


Fig. 4.3 Finite element axisymmetric and 3D models of Configuraion-3z

5. RESULTS

5.1. CONFIGURATION-1 (PARTIAL INSERT)

5.1.1 HAND CALCULATION RESULTS:



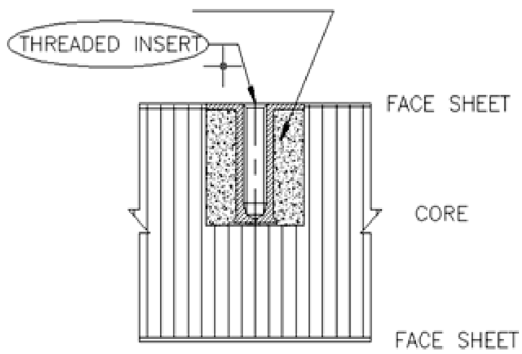


Fig. 5.1 Typical partial insert used in sandwich panel and its AutoCAD sketch

Effective length calculation:
Length of flange, $L_f = 3mm$
Insert L_i = length gripping the bolt/2 = 10/2 =5mm
Joint length, L_j = 3+5=8mm
Bolt length, $L_b = L_j$ =8mm
Potting compound, L_{pc}= 5-0.25= 4.75mm
Effective length of core, L_{core}=11mm

Effective area of bolt, $A_b = \frac{\pi}{4}(D-0.938p)^2 = 36.6\ mm^2$

Bolt Stiffness, $k_b = \frac{A_b E_b}{L_b} = 526286.4$ N/mm

5.1.1.1 Calculation of stiffness constant: Shigley's approach

Effective modulus, $E_{ipc} = \frac{E_i A_i + E_{pc} A_{pc}}{A_i + A_{pc}}$

E_{ipc}= 37663.37 N/mm^2

Effective modulus of joint, $E_m = \frac{L_j}{\frac{L_f}{E_f} + \frac{L_{i1}}{E_i} + \frac{L_{pc}}{E_{ipc}} + \frac{L_{core}}{E_{core}}}$

E_m= 46674.5 N/mm^2

Using the formula given in eq. (9), the joint stiffness and in turn the stiffness constant C, of the joint configuration has been calculated and tabulated as in table 5.1

Table 5.1 Stiffness constant using shigley's approach (configuration -1)

parameters	Case 1	Case 2
α	30°	45°
d_h	12mm	12mm
d_b	8mm	8mm
L_j	8mm	8mm
E_j	46674.5MPa	46674.5MPa
K_m	828651.2447 N/mm	1148198.258 N/mm
K_b	526286.4 N/mm	526286.4 N/mm
K_m+K_b	1354937.7 N/mm	1674484.7 N/mm
Φ	0.388	0.3143

5.1.1.2 Calculation of stiffness constant: Juvinall's approach

Case1: α=30°

Effective diameter of the flange,

$$d_f = d_h + 2\times\left(\frac{L_f}{2}\right)\tan 30° = 13.732mm$$

$$A_f = \left(\frac{\pi}{4}\right)\times\left(d_f^2 - d_b^2\right) = 97.84mm^2$$

$$k_f = \frac{A_f E_f}{L_f} = 2282850 N/mm$$

Effective diameter of insert (top portion),

$$d_{i1} = d_h + 2*\left(L_{f+}\frac{L_{i1}}{2}\right)\tan 30° = 15.61mm$$

$$A_{i1} = \left(\frac{\pi}{4}\right)*\left(d_{i1}^2 - d_b^2\right) = 141.08mm^2$$

$$k_{i1} = \frac{A_{i1}E_i}{L_{i1}} = 39501242.9\ N/mm$$

Effective diameter of insert (bottom portion), d_{i2}=12mm

$$A_{i2} = \left(\frac{\pi}{4}\right)*\left(d_{i2}^2 - d_b^2\right) = 62.83mm^2$$

$$k_{i2} = \frac{A_{i2}E_i}{L_{i2}} = 925943.1 N/mm$$

Effective diameter of potting compound, d_{pc}=18mm

$$A_{pc} = \left(\frac{\pi}{4}\right)*\left(d_{pc}^2 - d_b^2\right) = 204.2\ mm^2$$

$$k_{pc} = \frac{A_{pc}E_{pc}}{L_{pc}} = 171960.9 N/mm$$

Effective diameter of core,

$$d_{core} = d_h + 2*\frac{L_f}{2}\tan 30° + 2*\frac{L_{core}}{2}\tan 30° = 20.083\ mm$$

$$A_{core} = \left(\frac{\pi}{4}\right)*\left(d_{core}^2 - d_{pc}^2\right) = 62.3\ mm^2$$

$$k_{core} = \frac{A_{pc}E_{pc}}{L_{pc}} = 1755.73 N/mm$$

Now calculating the effective joint stiffness by considering the components as spring in series,

K_m=671013.5 N/mm

Therefore stiffness constant,

$$\varphi = \frac{526286.4}{526286.4 + 671013.5} = 0.411$$

Case 2: α=45°

Effective diameter of the flange,

$$d_f = d_h + 2\times\left(\frac{L_f}{2}\right)\tan 45° = 15\ mm$$

$$A_f = \left(\frac{\pi}{4}\right)\times\left(d_f^2 - d_b^2\right) = 126.45mm^2$$

$$k_f = \frac{A_f E_f}{L_f} = 2950479 N/mm$$

Effective diameter of insert (top portion),

$$d_{i1} = d_h + 2*\left(L_{f+}\frac{L_{i1}}{2}\right)\tan 45° = 18.25mm$$

$$A_{i1} = \left(\frac{\pi}{4}\right)*\left(d_{i1}^2 - d_b^2\right) = 211.32mm^2$$

$$k_{i1} = \frac{A_{i1}E_i}{L_{i1}} = 59169934.1 N/mm$$

Effective diameter of insert (bottom portion), d_{i2}=12mm

$$A_{i2} = \left(\frac{\pi}{4}\right)*\left(d_{i2}^2 - d_b^2\right) = 62.83mm^2$$

$$k_{i2} = \frac{A_{i2}E_i}{L_{i2}} = 925943.1 N/mm$$

Effective diameter of potting compound, d_{pc}=18mm

$$A_{pc} = \left(\frac{\pi}{4}\right)*\left(d_{pc}^2 - d_b^2\right) = 204.2\ mm^2$$

$$k_{pc} = \frac{A_{pc}E_{pc}}{L_{pc}} = 171960.9 N/mm$$

Effective diameter of core,

$$d_{core} = d_h + 2*\frac{L_f}{2}\tan 45° + 2*\frac{L_{core}}{2}\tan 45° = 26\ mm$$

$$A_{core} = \left(\frac{\pi}{4}\right)*\left(d_{core}^2 - d_{pc}^2\right) = 276.46mm^2$$

$$k_{core} = \frac{A_{pc}E_{pc}}{L_{pc}} = 7791.95\ \text{N/mm}$$

Now calculating the effective joint stiffness by considering the components as spring in series,

K_m=732235.3 N/mm

Therefore stiffness constant,

$$\varphi = \frac{526286.4}{526286.4 + 732235.3} = 0.393$$

$$\text{Prying factor,}\quad f_{pry} = \frac{\frac{t_f}{2} + w_b}{e_b}$$

$$f_{pry} = 1.94$$

Therefore, now the stiffness constant considering prying effect is given by

$C=f_{pry}*\Phi = 0.753$

where, c is taken from the shigley's approach for a frustum angle of 30°

5.1.1.3 Core shear stress calculation:

$$\text{Core shear stress is given by,}\ \tau_c = \frac{\text{Force in the core}}{\text{shear area of the core}}$$

$$\text{Force} = \frac{\text{Load} * 48.5}{25}\ \text{N}$$

$$\text{shear area} = (\text{perimeter of core} * \text{thickness}) - \text{potting area}$$

$$\text{Force} = 1319.2\text{N}$$

$$\text{shear area} = 4745.31\text{mm}^2$$

$$\tau_c = \frac{FORCE}{ShearArea} = 0.277\ \text{MPa}$$

5.1.1.4 Bolt force change in axisymmetric analysis:

$$\Delta P = \frac{(\alpha'_f L_f + 2\alpha_w L_w - \alpha'_b L_b)\Delta T}{\frac{L_b}{E_b A_b} + \frac{L_f}{E_f A_f} + 2\frac{L_w}{E_w A_w}}$$

By considering effective co efficient of thermal expansion, effective area and effective lengths as discussed earlier, substituting all the values in the above equation and the change in bolt force due to temperature difference of 100°c for configuration-1is

ΔP=3146.6 N

5.2. CONFIGURATION-1: AXISYMMETRIC ANALYSIS

FE model as shown in figure 4.1 is developed using axisymmetric elements. Simulation has done by splitting the procedure into two conditions. Firstly, for an M8 bolt, a preload of 10400N has applied at the center of the bolt. In the second condition, temperature difference, ΔT=100°c is applied as an external loading.

The change in the bolt force due to temperature change of 100 °C is calculated and following values are tabulated

Initial preload	1.05E+04	Load change due to thermal expansion	1.31E+04

Now, Percentage change in preload due to the thermal expansion of the joint, from the FE analysis is computed by

Percentage change in preload= (change in load/original load)*100

Percentage change in preload= 24.5%

5.2.1. Stiffness constant evaluation:

Table 5.2 Stiffness constant evaluation using axisymmetric analysis

Parameters	FE Results
δ_b	0.0321mm
δ_m	0.0173mm
δ_w	0.0217mm
F	10400N
K_b	323987.5 N/mm
K_m	601156.07 N/mm
K_w	14930301.33 N/mm
$K_b + K_m$	925143.61 N/mm
$\frac{2K_bK_m}{K_w}$	26090.17
Φ	0.34

Stiffness constant for the partial insert using axisymmetric analysis is found out by calculating the stiffness of the bolt, members and the washer from the FE tool and thereby the stiffness constant, as given in the table 5.2 above.

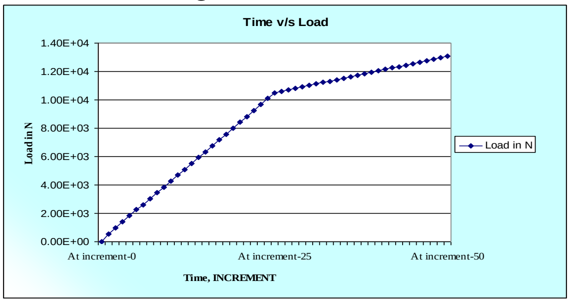


Fig. 5.2 Graph of Time v/s Load

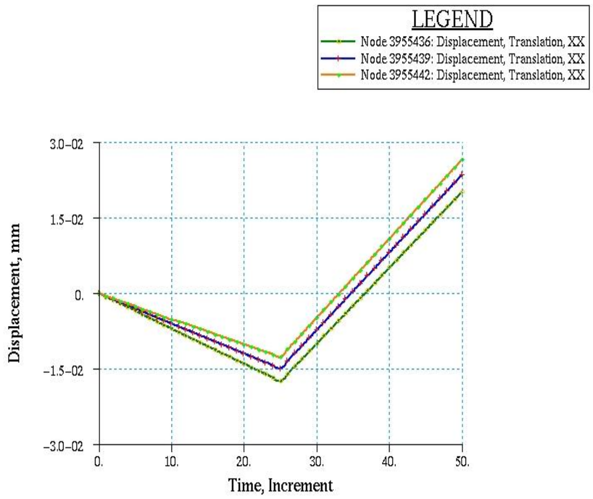


Fig. 5.3 Graph of Increment v/s displacement of bolt with preload and temperature change

The displacement and change in bolt force is plot for nodes at the bolt-washer interface.

FE plots for configuration-1 with axisymmetric analysis is given below

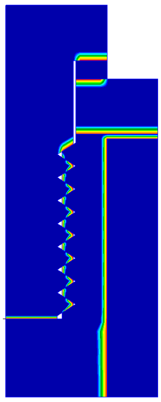


Fig. 5.4 Initial contact status for config-1

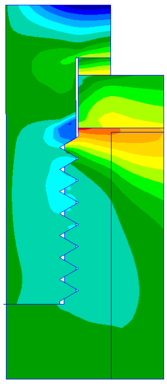


Fig. 5.5 Displacement pattern of joint after preloading

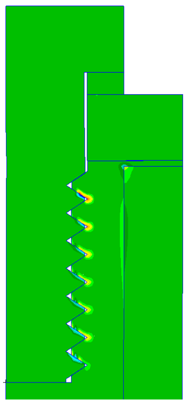


Fig. 5.6 Contact stress after temp loading

Table 5.3 Stiffness constant evaluation using 3-D analysis

Parameters	FE Results
δ_b	0.0658mm
δ_m	0.05mm
δ_w	0.0122mm
F	10400N
K_b	158054.7 N/mm
K_m	208000 N/mm
K_w	12955304.2 N/mm
$K_b + K_m$	366054.71 N/mm
$\frac{2K_b K_m}{K_w}$	5075.2
Φ	0.4258

Stiffness constant for the partial insert at the preloaded state is found out by calculating the stiffness of the bolt, members and the washer thereby the constant, as given in the table 5.3 above.

Table 5.4 Change in force in 3-D analysis

Load	Shear stress in MPa	
	Analytical	FEA (averaged)
680	0.277	0.27
1020	0.417	0.411
1360	0.556	0.551
2040	0.834	0.805
2720	1.112	1.05
3400	1.389	1.53
4080	1.668	1.83

Table 5.5 Shear stress at core for configuration-1

Load	Force in N at time-1	Force in N at time-2
680	7152.23	6579.11
1020	7152.23	6901.54
1360	7152.23	7361.51
2040	7152.23	8742.46
2720	7152.23	10200.82
3400	7152.23	1122838
4080	7152.23	12641.14

5.3 THREE DIMENSIONAL ANALYSIS:

5.3.1 FE PLOTS FOR 3D ANALYSIS:

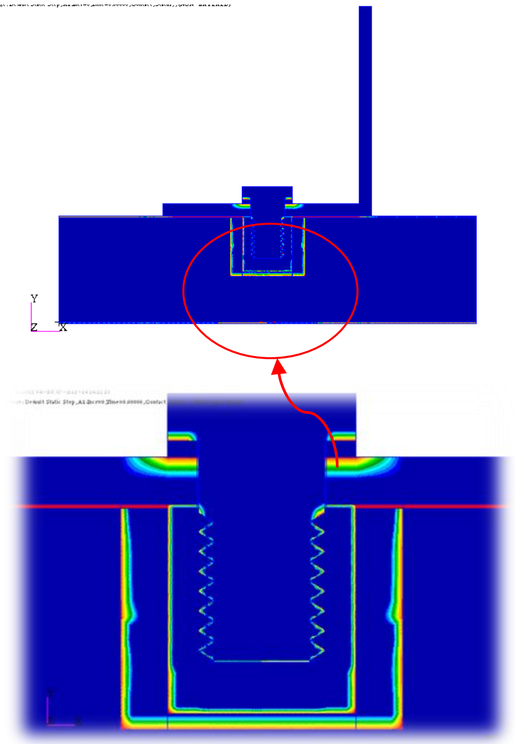


Fig. 5.7 3-D Initial contact status and its enlarged view

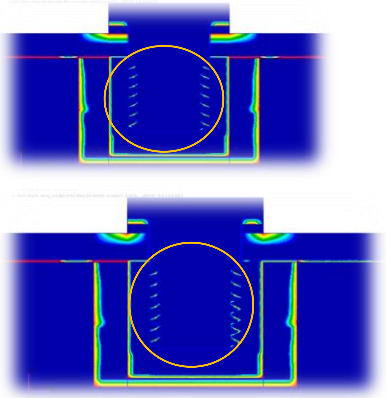


Fig. 5.8 contact status at the preloaded and externally loaded state

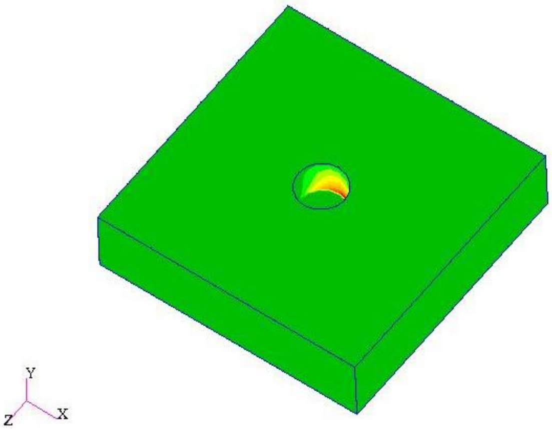


Fig. 5.9 Core shear stress in configuration-1

Similarly, the joint is also studied for various values of external loading and following graphs are obtained.

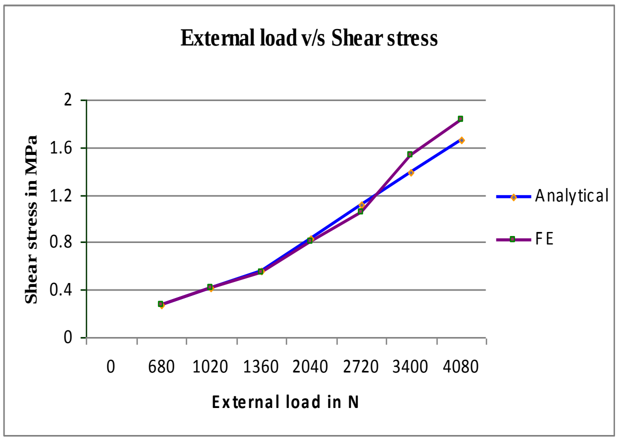


Fig. 5.10 Graph of core shear stress variation for

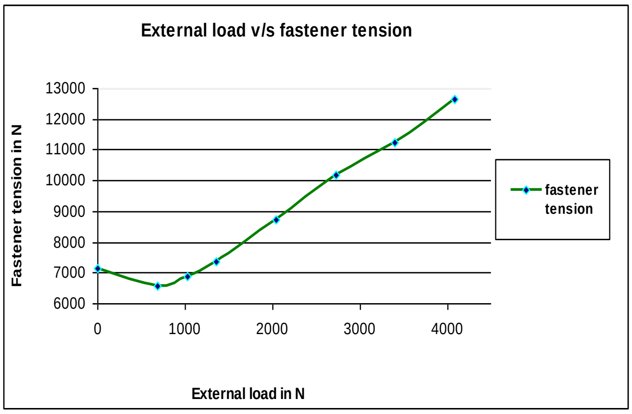


Fig. 5.11 Graph of variation of external load v/s fastener tension.

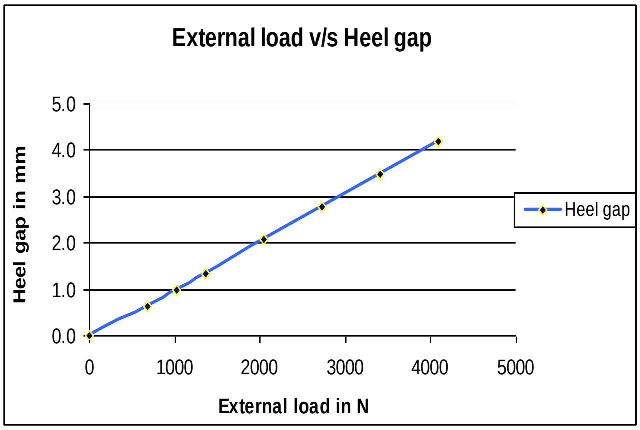


Fig. 5.12 Graph of variation of External load v/s heel gap due to prying effect.

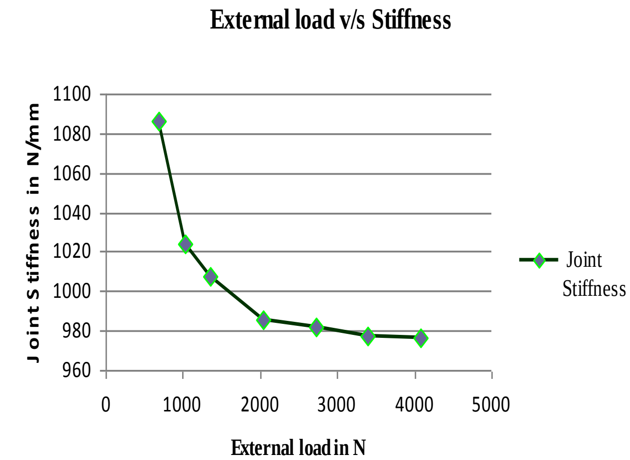


Fig. 5.13 Graph of Load v/s Joint stiffness

Similarly the same analysis is done for other two configurations as shown in Fig. 5.14 and Fig 5.15, i.e., stepped insert and through insert and change in bolt force and the stiffness constant are evaluated.

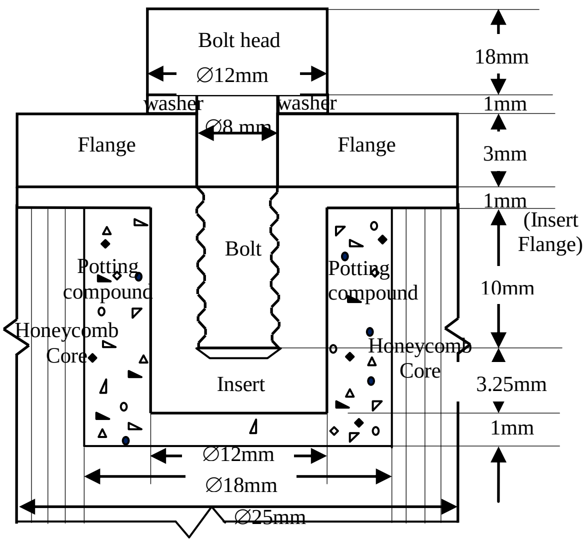


Fig. 5.14 Typical Stepped partial insert used in sandwich panel

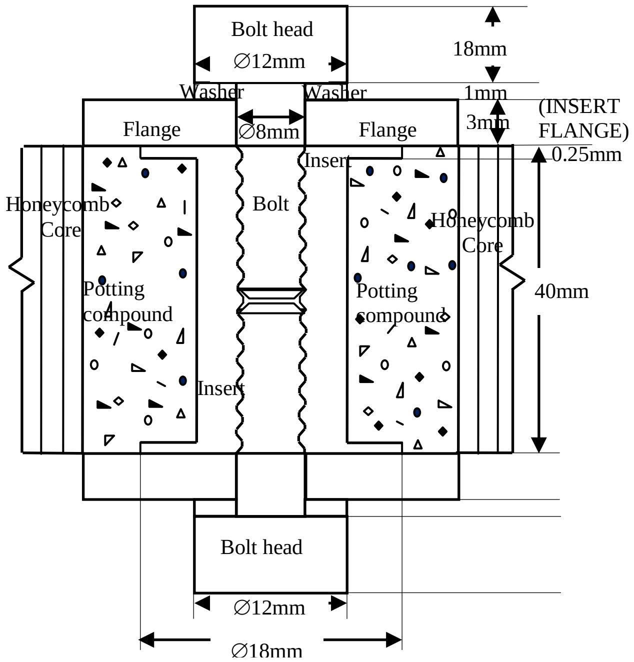


Fig. 5.15 Typical Through insert used in sandwich panel

Table 5.6 Stiffness constant values for various configurations

	Type	Configuration-1	Configuration-2	Configuration-3
Shigley approach	α=30°	0.388	0.371	0.266
	α=45°	0.314	0.298	0.187
Juvinall approach	α=30°	0.411	0.428	0.463
	α=45°	0.393	0.402	0.442
Axisymmetric analysis		0.34	0.271	0.182
3D Analysis		0.425	0.496	

Comparison of stiffness constant values estimated using analytical and FE analysis for three configuration are given in the Table 5.6. We observe that stiffness constant values estimated using the 2D axisymmetric FE models are lesser compared to the theoretical and 3D models. This may be due to the joint material like core and face sheet beyond the potting compound are not considered in the modeling. However axisymmetric models are simple in nature and can be used to estimate the variation in the bolt preload due to the temperature changes

6. CONCLUSION

From these studies, it is observed that the stiffness constant calculated using empirical formulae is more for lower value of the conical frustum angle. Juvinall's approach was found to be conservative; as the stiffness constant obtained using the approach was comparatively more than that obtained using Shigley's approach. 2D axisymmetric FE model is simple and can be used to study the behaviour of joints due to temperature variations at various payload module interfaces. The 3D FE models are used to estimate the variation in the bolt force and the distribution of the stress in the sandwich materials around the insert. Pull-out strength of the insert in the joints is the measure of the load carrying capacity of such joints. Methodology followed in this paper can be used to understand the characteristics of bolt-insert joint in the sandwich panels.

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