Novel Design of a Joint with Translational Freedom for Propellant Tank Anti-Slosh Baffle System

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ABSTRACT

Launch vehicles stages are fuelled by different propellants at ambient and cryogenic temperatures, which are stored in propellant tanks. The propellant tanks are subjected to pressure, structural, aerodynamic, thermal and slosh loads. Anti-slosh baffles are provided inside propellant tanks to provide adequate slosh suppression. These baffles are mounted inside tanks within limited space, along with other internal systems. The pressure and structural loads induce axial dilation of propellant tanks and rotation of end rings. Axial freedom at baffle mounting joints is required to reduce the mechanical reaction loads at the joints caused by tank deformation, under pressure and axial loads. The anti-slosh baffle joints should have sufficient freedom to manage such tank dilations, which is achieved by providing translational freedom to the joints, without compromising the structural integrity. Conventional joints with translational freedom have complex joining procedure, demand space and access requirement. Only limited joining techniques like bolting and riveting can be employed in anti-slosh baffle systems to meet the assembly and integration requirements. The assembly of anti-slosh baffle sub-assemblies inside propellant tanks is easier with joints comprising of rivets. This paper presents a novel design of a riveted joint with translational freedom for anti-slosh baffles inside a propellant tank. The design is validated with a quasi-static tensile test and the qualified joint is adopted in engineering of the system for flight.

Keywords: *Propellant tank; Anti-slosh baffle; Tensile test; Translational freedom*

1. INTRODUCTION

Launch vehicles have multiple stages designed to achieve diversified mission requirements. The liquid propulsion stages of a launch vehicle are powered using different propellants. Depending on the requirements, these are stored at ambient or cryogenic temperatures in propellant tanks. Apart from storing propellants, the tanks also act as primary structures of the vehicle [1], because of which these are subjected to various loads namely pressure, structural, aerodynamic, thermal and slosh.

Propellant tanks deform under the structural and pressure loads. The systems mounted to propellant tanks should have sufficient stiffness and freedom of movement to withstand these loads. Also, the inert mass of the vehicle is to be minimized to increase its payload capacity. Hence, designing joints in propellant tanks with sufficient joint strength, stiffness, degrees of freedom, in addition to optimised mass and limited space is challenging.

The slosh, which is the oscillations of the free surface of the propellant tank induce additional load on the tank. The slosh load, if not controlled, can lead to vehicle instability and huge load on tank structure, leading to failure of the mission. Slosh suppression devices are used to dampen the free surface motion of propellants, and prevent vehicle instability induced due to slosh [2]. Anti-slosh baffles used in propellant tanks as an effective way of supressing slosh, reducing the duration and amplitude of oscillations [3]. These are mounted inside tanks within limited space, along with other internal systems. The slosh dampening systems are mass optimised structure made of sheet metal primely to dampen the slosh

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effects. The joints of the slosh dampening system to the tank shall have required translational freedom to avoid the structural load transfer taking place through the slosh members.

The particular tank requires a matrix structure of horizontal and vertical baffles. The baffle structure is connected between the fore and aft end rings of the propellant tank. The internal pressure and structural loads induce axial dilation of the fuel tanks and rotation of end rings. This demand for axial freedom at baffle mounting joints to reduce the mechanical reaction loads at the joints, without compromising the structural integrity. Only limited joining techniques like bolting and riveting can be employed in these joints to meet the assembly and integration requirements.

If the bolted joints are used, high preload of the order of 60% of yield strength of the bolt material is to be applied to avoid bolt loosening under vibration. However, this preload will affect the translational freedom of the joint. To overcome this, special designs with steel bushes in the bolted holes are to be employed. The length of the bush is to be designed in such a way that the preload does not act on the anti-slosh baffle plates. The steel bush should not yield and the anti-slosh baffle plate should not be loose. This will result in a higher hole diameter and only a few fasteners can be provided in the available space, which may not be structurally acceptable.

Riveted joints are prepared for the assembly of anti-slosh baffle sub-assemblies inside propellant tanks made of sheet metal, and it requires lesser space than bolted joints. This paper describes about the novel design of a riveted joint with translational freedom in the anti-slosh baffles of a fuel tank. Detailed FE analysis is carried out to finalise the riveted joint. The design is validated with a quasi-static tensile test and the qualified joint is adopted in engineering of the system for flight.

2. Literature Review

Different studies on bolted and riveted structural joints used in aerospace structures are reviewed to evaluate its performance. Li et al. [4] conducted experiments on composite riveted joints used for aircraft applications. They studied the tensile response and failure of seven CFRP riveted joints with circular holes, at quasi-static (0.5 mm/min) and dynamic (4 m/s and 8 m/s) loading rates.

The variation of tensile strength with loading rate was found to have no effect on the joints. It was also shown that the average total energy absorption of the joints increased and the modes of failure of the joints changed with increase in the loading rate.

Szolwinski and Farris [5] analysed a single-lap joint structure to obtain the residual stress field in the joint. They included the results of rivet installation process parameters viz. quasi-static loading and squeeze force-controlled riveting, obtained from the FE model, to the final residual stress field. A coefficient of friction of 0.2 was used between all contact surfaces, during the FE analysis. The model was validated experimentally. The analysis showed that the residual stress field depended on the through-thickness restraint because of the rivet. It also showed that near the rivet hole periphery, a zone of compressive residual hoop stress is dominant, whereas a zone of tensile residual hoop stress is dominant away from the hole. The residual hoop stress zones can impact the propagation of fatigue cracks.

The quality of rivet formation was studied by Hossein Cheraghi [6] with reference to process parameters like squeeze force, rivet length, tolerances on rivets and hole diameters. The effect of the above-mentioned parameters was studied numerically and statistically. He used a frictional value of 0.2 for modelling all contacts in the numerical model. It was concluded that if the rivet hole diameter decreases, or if the rivet diameter increases, the range of squeeze force increases. It was found that if the rivet length and diameter increase, the amount of squeeze force needed to drive a rivet increases.

Honglun Huan and Min Liu [7] experimentally evaluated the structural behaviour of riveted lap joints with squeeze force. The study comprised of static strength experiments and FE analysis. Three levels of squeeze forces: 28 kN, 31 kN and 34 kN were used for the study. The coefficient of friction was specified as 0.18 for the FE analysis. It was shown that the joint stiffness is not affected by increasing the squeeze force. But the joint strength decreases with an increase in squeeze force. The joint stiffness gradually decreases with an increase in the tensile load. It was also proven that under static load, the first hole of the outer sheet fails first, due to the highest amount of load transmission and secondary bending.

Joints with translational freedom are commonly used in the railway industry, on joints in railway tracks to accommodate axial displacements due to temperature differences in long welded rails. In a switch expansion joint, two running rails viz; a stock rail and a tongue rail, slide against each other [8]. The expansion joints limit the degrees of freedom in lateral direction on Continuously Welded Rail (CWR) tracks.

Wald et al. [9] predicted the stiffness of bolted joints in steel plates with slotted holes, using component method. The slotted holes are perpendicular to the applied force. As compared to circular holes, it was observed that bolted joints with slotted holes perpendicular to the applied force had lower stiffness and higher deformation capacity.

It can be observed that riveted and bolted joints are widely used in aircraft and aerospace industry. Many researchers studied the behaviour of these joints under different influencing parameters. Finite element analysis has been used in predicting the influence of squeeze force on rivet head formation, fatigue life, crack formation, residual stress analysis of riveted joints. The FE models are validated using tensile tests, mainly quasi-static, carried out at a very low loading rate. Though study has been conducted on the behaviour of bolted joints with translational freedom, study of such riveted joints is not conducted. The joint under present study has translational freedom when the propellant tank deforms under loads. Being a new design, a quasi-static tensile test is done on a test specimen simulating the joint, to experimentally validate the design.

3. Design of Riveted Joint with Translational Freedom

The baffle in this fuel tank is a matrix structure consisting of 8 vertical and 5 horizontal members. The location of baffles is finalised after detailed slosh analysis. Figure 1 shows the tank internal view with anti-slosh baffles. The vertical baffles are classified as top (in the fore end dome), middle (in the cylindrical shell) and bottom (in the aft end dome) vertical baffles. The middle vertical baffles are attached to the projections on end rings at the fore and aft ends. The middle vertical baffles are connected to brackets fastened to tank end rings by Ø5 mm universal head V65 rivets (ultimate shear strength = 265 N/mm^2). Constrained joint will result in a stress level beyond the ultimate shear strength of the rivet material. The size and arrangement of the rivets are a trade-off between the margin w.r.t the ultimate shear strength and the space constraints in the ant-slosh baffles. The rivets are arranged in such a way that the edge distance between the rivet hole and anti-slosh baffle plates are ensured. In addition to the axial load condition, rotation of the tank end ring flanges is also noticed due to the inward movement of the ring region under internal pressure. For this, extensive Finite

Element Analysis (FEA) was carried out using ANSYS 18.1 considering the rotation and translational freedom at the joint.

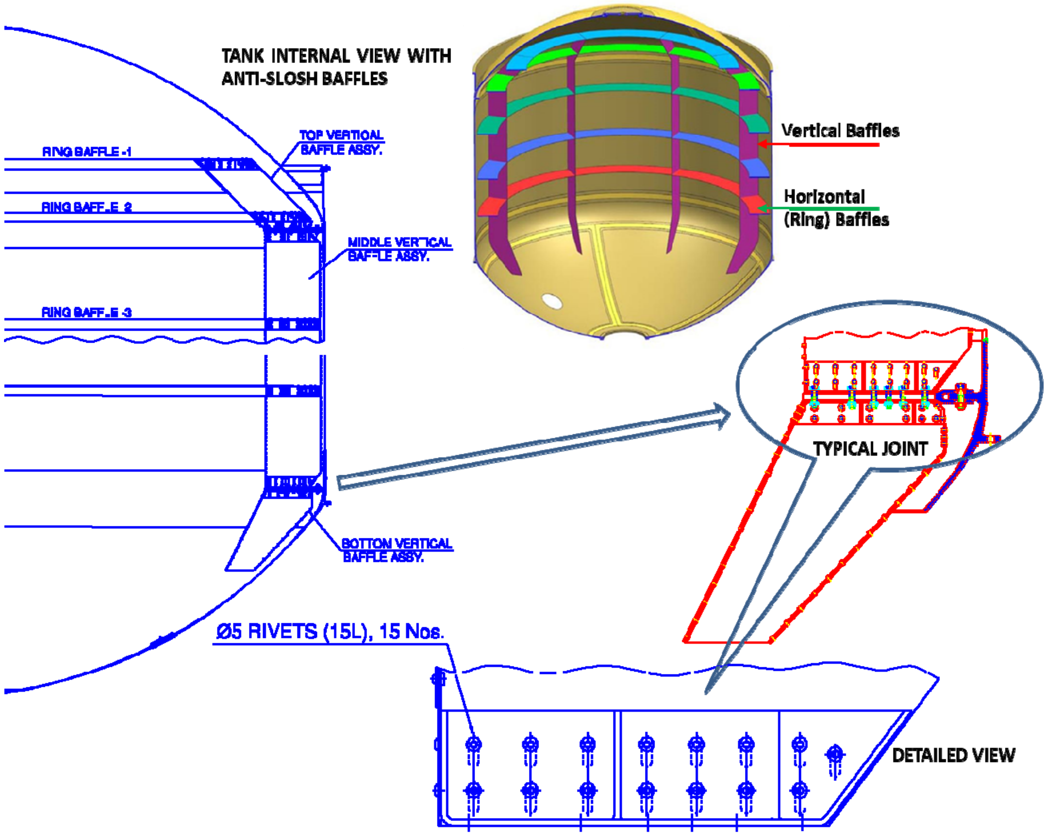

Figure 1: Tank internal view with anti-slosh baffles

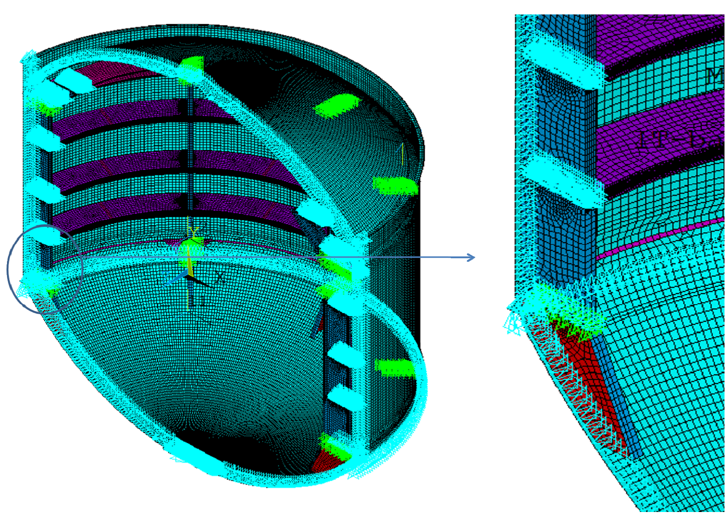

Figure 2: FE model with boundary conditions

Figure 2 shows the FE model with boundary conditions. The tank is modelled by 2D shell elements and is assumed to be simply supported at the AE ring flange. The symmetric boundary conditions at the ends of the tank are modelled. The symmetric boundary conditions at the end of the horizontal baffles are also modelled to simulate the hoop continuity. The elongated axial slots at the middle vertical baffle with AE bracket interface is simulated by coupling all degrees of freedom except axial translation at the AE side. There are elongated radial slots at the tank support bracket interface simulated by coupling all degrees of freedom except radial translation. Fasteners are modelled using beam elements.

Three load cases are considered as specified in Table 1.

Table 1: Load cases considered for FE analysis

Load Step	**Loads considered**
Load step 1	Internal pressure
Load step 2	Structural tensile load
Load step 3	Slosh pressure load acting on the horizontal and vertical baffles

The maximum ultimate slosh pressure load applied at the anti-slosh baffles was 11.25 kPa, and the value varied at different baffles. The total baffle deformation under ultimate tank and slosh pressures is shown in Figure 3. It is noticed that deformation under ultimate tank and slosh pressures, a maximum baffle deformation of 36 mm is locally observed at the horizontal baffle in the second horizontal baffle in the FE dome. This is acceptable from the structural and slosh point of view. The von-mises stress distribution in vertical baffles under ultimate tank and slosh pressures is shown in Figure 4. A maximum stress of 280 N/mm^2 is observed locally at the FE of one of the vertical baffles, which is acceptable.

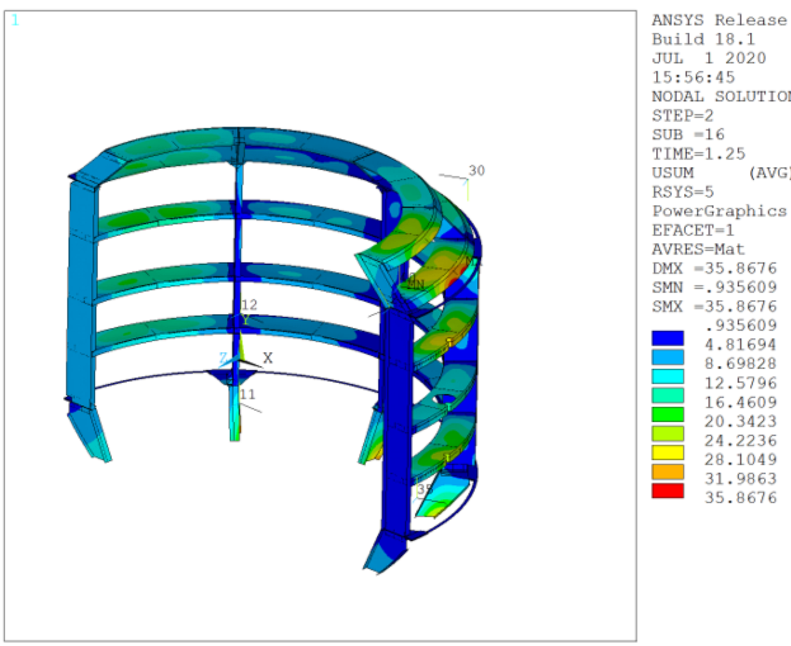

Figure 3: Total baffle deformation under ultimate tank and slosh pressures

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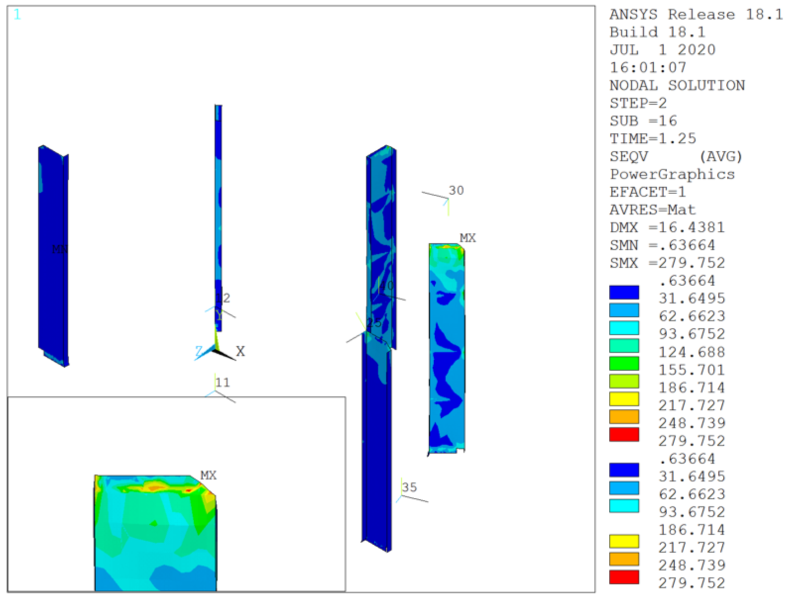

Figure 4: Von mises stress distribution in vertical baffles under ultimate tank and slosh pressures

Modal and harmonic analysis were also done for all critical modes, considering 3% damping. Based on the FE analysis, the maximum shear stress noticed at Ø5 mm rivets connecting middle vertical baffle to brackets is 170 N/mm^2. A margin of 0.55 exists w.r.t the ultimate shear strength of 265 N/mm^2 for V65 rivets.

4. Experimental Validation of Design

The design of riveted joint with translational freedom for anti-slosh baffles inside propellant tank is experimentally validated. To simulate the actual assembly inside the tank, a test specimen from AA2219-T6 sheets is fabricated, considering four riveted joints in slotted holes.

One AA2219-T6 sheet of 1.5 mm thickness is sandwiched between two AA2219-T6 sheets of 3 mm thickness by Ø5 V65 rivets, to simulate the actual baffle joint in the fuel tank. According to rivet selection procedure by Li et al. [10] rivet length of 15 mm is selected. The configurations of the sheets are shown in Figure 5. Two rows and two columns of the actual riveted joint are simulated in the experiment. Four slotted holes are provided on the 1.5 mm sheet. The length of each slotted hole is 12 mm, and the width is 5.5 mm.

Ø2 mm pilot holes are initially drilled on each of the 3 mm sheets, which are then match drilled and enlarged after suiting with the other sheets. Sheet dimensions of 73 mm x 58 mm simulate the actual joint in the tank. An additional length of 90 mm is provided on each sample for adequate grip length. Hence, sheet samples of dimension 73 mm x 148 mm are used for testing.

The rivet joint is formed using a pneumatic rivet gun with a squeeze force of 9 kN applied on the driven head of the rivet at a pressure of 6 bar. After riveting, it was ensured that the manufactured rivet heads were free of tool marks and no gap exists between the head and the

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3 mm sheet. It was also ensured that the material around the rivet head on the 3 mm sheet was free of cracks.

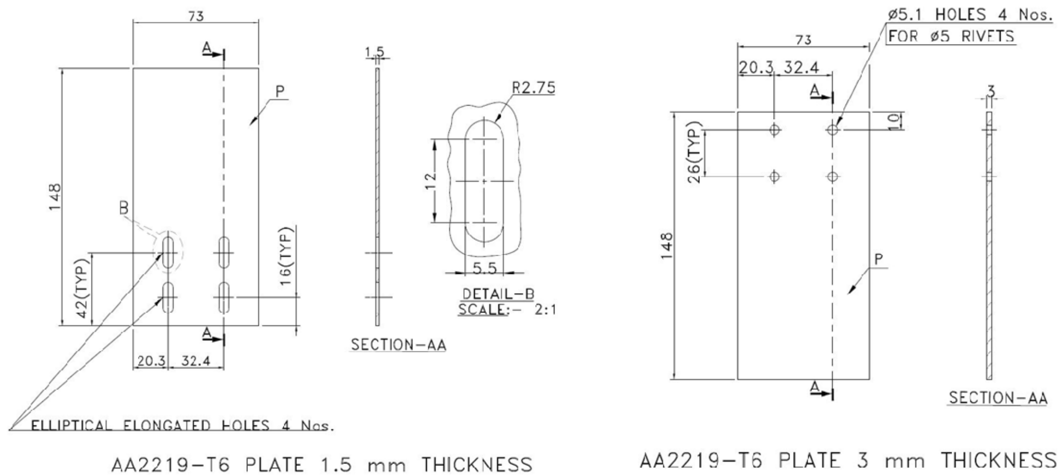

Figure 5: AA2219-T6 sheets for experimental validation of design

The riveted joints of the test specimen were subjected to axial loading. The test specimen is shown in Figure 6. The experiment was carried out on a Zwick Roell- 250 kN universal testing machine. The frictional resistance, F_f, in the riveted joint is obtained by the equation:

$$F_f = \mu R \tag{1}$$

where, μ = coefficient of friction and R = reaction load.

The residual load acting at the rivet joints is estimated to be of the order of 1 kN. Hence, for a joint with four rivets, the expected frictional resistance is of the order of 4000 x 0.2 = 800 N.

The riveted joint is subjected to pure shear by applying a tensile load. A dummy sheet of 1.5 mm thickness is placed between the two 3 mm sheets to ensure proper grip in the UTM. The test setup is shown in Figure 7.

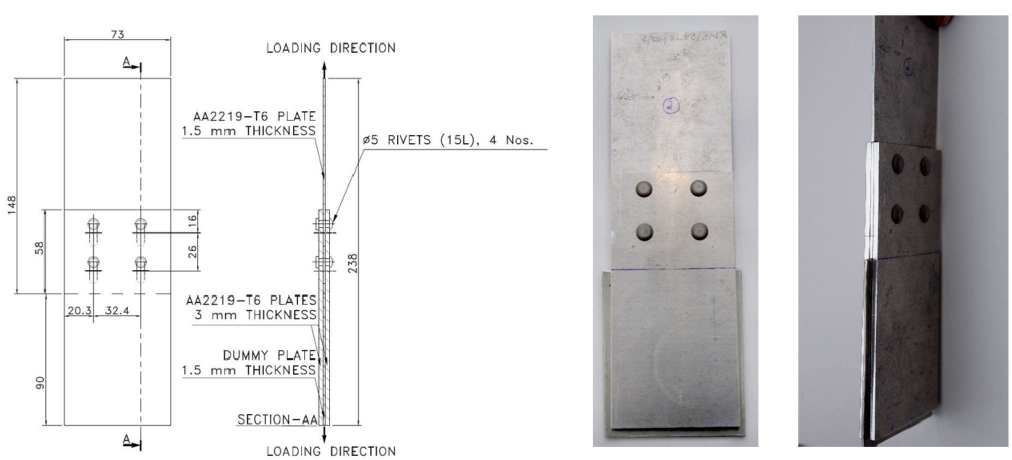

Figure 6: Test specimen for loading rivet joints

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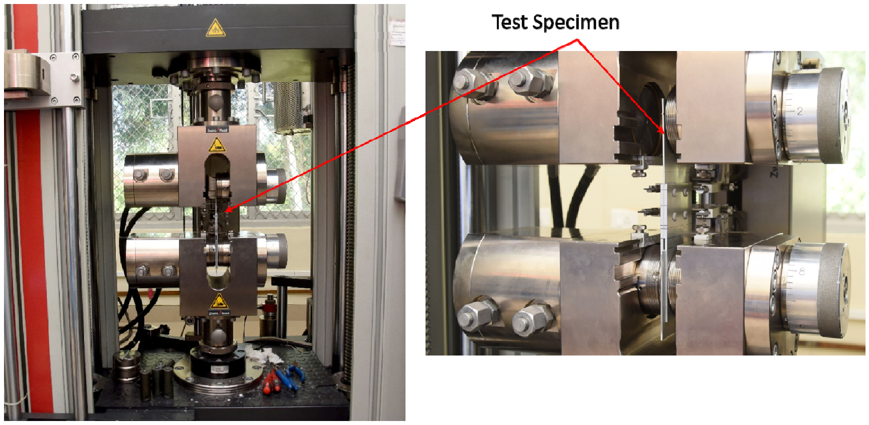

Figure 7: Test setup for loading riveted joints

A quasi-static tensile load was applied to the setup, with a loading speed of 2 mm/min. Step loading of 50 N was applied with a hold time of 5 sec between the load steps. The test continued till 10 mm displacement of the rivets in the slots, or till a maximum load of 500 N, whichever is the earliest.

After testing, the rivet joint was cut as shown in Figure 8. This is to verify that the joint is intact after the movement of the sheets. This ensures the rivet movement due to specimen loading.

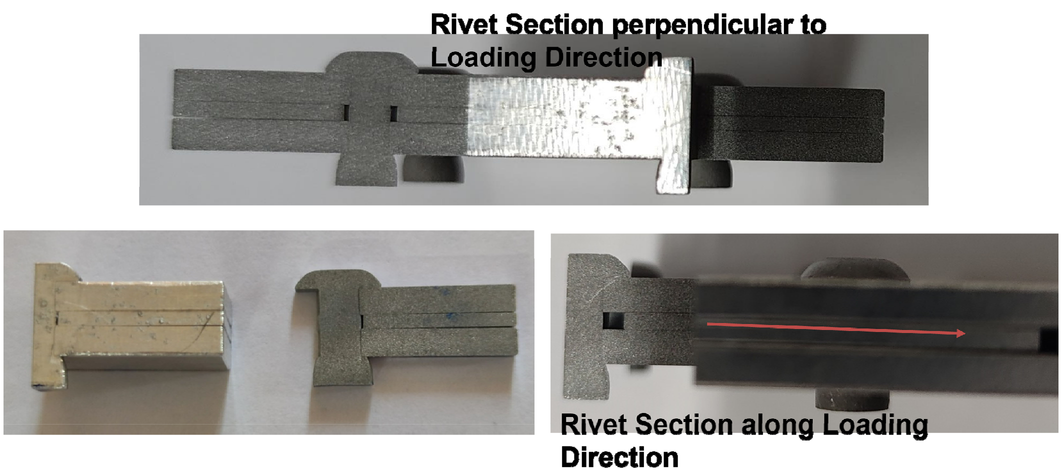

Figure 8: Cross sectional views of rivets after testing

5. Results and Discussions

With the application of tensile load, the joint remained intact till 485 N, before the displacement of the joint commenced, and continued till 10 mm in the slot. Force vs. displacement, displacement vs. test time and force vs. test time were plotted. The results are shown in Figure 9.

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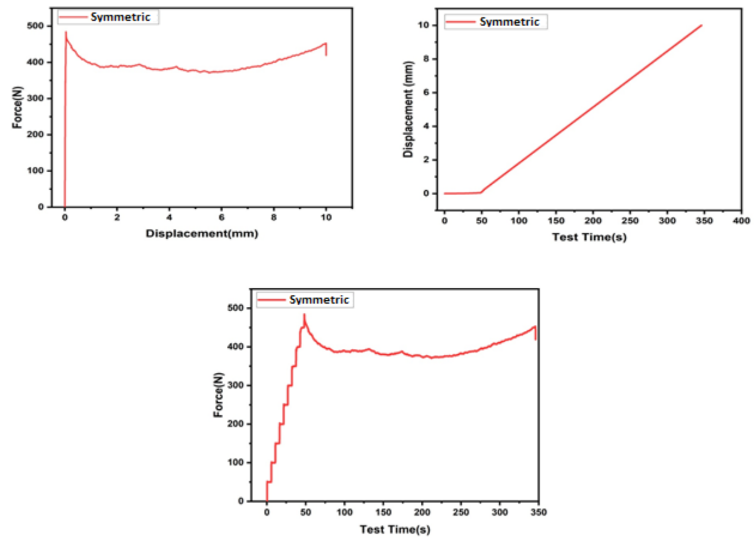

Figure 9: Plots of loading riveted joints

During the riveting process, both ends of the rivets get mechanically deformed and closely butt with the mating surfaces of the adjacent sheets, closing all the nearby gaps and crevices. Cut sections of the rivets after testing shows that the initial gaps between rivet and the mating sheets gets closed due to material flow of the rivet. In this condition, the joint is fully intact till the applied load overcomes the frictional resistance offered by the riveted joint. The load is observed to be 485 N, which is analogous to static frictional force to disturb the initial configuration of the riveted joint. Once the frictional resistance is overcome, slight reduction in the applied load is observed. This corresponds to the force required to move the rivet within the slotted hole, with close contact between the rivet heads and the adjacent sheets. This is analogous to kinetic friction. Once it is about to reach the other end of the slotted hole, slight increase in the load is observed due to the constraints offered by the edges of the plates.

The location of rivets in the slots after testing is shown in Figure 10.

Figure 10: Rivet movement in slots after testing

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Since the rivet movement in the joints commenced at 485 N, which is within the expected value of 800 N, the adequacy of the design is validated. The uncertainties of the frictional resistance in the riveted joint are a function of parameters like coefficient of friction, squeeze force, rivet length, tolerances on rivet and rivet hole diameter.

6. Conclusions

A non-conventional riveted joint with translational freedom is designed for anti-slosh baffles in a fuel tank. The joint is design account the limited space inside and ability to maintain integrity in variety of loads acting on the joints. The integration aspects of the joint also taken incorporated considering the accesses and tooling.

Detailed FE analysis was carried out considering the rotation and translational freedom at the joint. Axial elongation considering both tank ultimate internal pressure and structural tensile loads is 8 mm. The designed joint is able to accommodate a ring movement of 12 mm. The size and arrangement of rivets are arrived at after several iterations. Modal and harmonic analysis were also done for all critical modes, considering 3% damping. Based on the FE analysis, the maximum shear stress noticed at Ø5 mm rivets connecting vertical baffle to brackets is 170 N/mm^2. A margin of 0.55 exists w.r.t the ultimate shear strength of 265 N/mm^2 for V65 rivets.

The design is validated with a quasi-static tensile test by loading the riveted joints.

When the riveted joints are axially loaded, these are subjected to pure shear. The rivet movement commenced at 485 N. This is within the expected value of 800 N. Such joints can be adopted in engineering of the system for flight. More tests are planned with different loading conditions of riveted joints, to predict the range of load to initiate the joint movement.

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