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Research Paper

CONTACT GAP ANALYSIS OF CANISTER TESTING CHAMBER BY USING FINITE ELEMENT ANALYSIS

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Canister testing Chamber is one of the most critical components in Defence Organization. Unavailability of data and literature regarding Missile Canister, are considered to be one of the main contributors for the failure of manufacturing Canister Chambers in a local industry. Canister is used for carrying, storing and launching of missile. During storage and launching, the canister is subjected to an internal pressure of 45 kg/cm^2 and external pressure of 9 kg/cm^2. So it is very important to test the canister for these pressures. A Canister testing chamber is used to test the canister for both internal and external pressures. For testing the canister it's both ends shall be closed by dummy dished ends. The test chamber will be used for testing of the integrated canister assembly for an external pressure of 9 kg/cm^2 and internal pressure of 45 kg/cm^2. As the testing chamber is subjected to these high pressures it shall be tested for any leakages through the hinged dished end. So the analysis of contact gap between this dished end and the shell of the chamber is to be done. To estimate the contact gap analysis a three-dimensional model of a canister testing chamber was made by finite element method using Uni Graphics.

Keywords: Canister testing chamber, Missile, Contact gap analysis, Finite element method

INTRODUCTION

Canister is a cylindrical container for holding, carrying, storing and launching of missile, usually a specified object or substance. The Canister Testing Chamber is used for testing the canister. This chamber will be used for the testing of the integrated chamber assembly for the internal pressure of 45 bar and external pressure of 9 bar. The testing chamber will be dedicated especially to perform the internal and external pressure testing.

The test shell setup is made from IS: 2062 plates welded to get a shell of 11 meters length and diameter of 1.5 meters. One end of the test setup will have dished end welded integrally to the shell. The other end of the shell will be a hinged door which is bolted to the shell with proper sealing between the contact

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gap to prevent the leakage. On the side of the dished end a screw rod is provided to press the dished end for leak proof joint which shall withstand the internal pressure during testing. The screw is actuated by a hand wheel provided through nut. The nut is fixed in a welded housing on the dished end. The screw front portion will have good surface finish with proper sealing arrangement to withstand 45 bar pressure without any leak. This screw is used to press the dummy dish end of the canister. Between both the mating faces rubber/gasket will be provided to avoid any leak of water during pressure testing.

The chamber will have inlet, outlet and air removal ports with suitable ball valves. 2 no's of pressure indicators, one analog and one digital will be provided. A storage tank and pumping system is also provided for pressurization of the chamber.

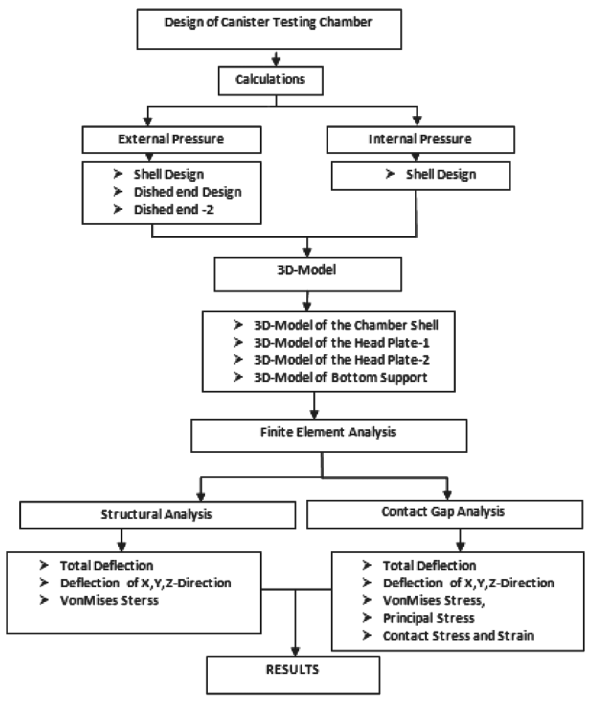
In the present study, the analysis of the contact gap between the hinged dished end and the shell of the Testing Chamber used for canister testing will be performed. The most important factors that are concentrated are stress distribution, deformation and selection of best size of Bolt.

OBJECTIVE AND METHODOLOGY

The methodology adopted in the project has listed below and this Procedure is shown in Figure 1:

- Design calculations of canister testing chamber are decided based up on canister dimensions and loads.
- From the calculations, a 3-D model of chamber is developed using Unigraphics.
- After developing the model, Finite Element Analysis is carried out for the internal pressure, deflection and stresses on the Chamber. And also, factor of safety is calculated and this should be above 3.
- The best bolt size will be selected based on the results of the finite element analysis.

Figure 1: Flow Chart Showing Methodology



DESIGN CONSTRAINTS AND MODELLING

The design formulae used in the "design by rule" method are based on the principal stress theory for calculating the average hoop stress. The principal stress theory of failure states that failure occurs when one of the three principal stresses reaches the yield strength of the material. Assuming that the radial stress is negligible, the other two principal stresses can be determined by simple formulas based on engineering mechanics. The Code recognizes that the shell thickness may be such that the radial stress may not be negligible, and adjustments have been made in the appropriate formulas. Various formulae used to calculate the size of the Bolt to avoid any leak of water during pressure testing are discussed below.

BOLT PRELOAD CALCULATION FOR M36 BOLTS

Bolt pretension, also called preload or pre stress, comes from the installation torque T applied while installing the bolt.

The inclined plane of the bolt thread helix converts torque to bolt pretension. Bolt preload is computed as follows:

$Pi = T/(K * D)$

where Pi = Bolt preload

T = Bolt installation torque = 10858 N-mm

K = Torque coefficient = 0.2

D = Bolt nominal diameter (i.e., bolt nominal size) = 36 mm

Pi = Bolt preload = 1508 N

Torque coefficient K is a function of thread geometry, thread coefficient of friction mc, and collar coefficient of friction mc.

Delta(δ) = $PL/(EA)$

Pi = Bolt preload = 1508 N

L = Bolt length = 50 mm

E = Bolt modulus of elasticity = 2e5 N/mm^2

A = Bolt cross-sectional area = $(\pi/4)*D^2$

D = Bolt nominal shank diameter = 36 mm

Delta(δ) = Measured bolt elongation in units of length = 0.00037 mm

The calculated Delta(δ), 0.00037 mm, is applied as a pretension (initial strain) for the bolts modeled as beams.

Table 1: Pre Load and Strain of the M36 Bolts

S. No.	Name of the Bolt	Pre Load (Pi) in N	Strain (δ) in MM
1.	M36	1508	0.00037

MATERIAL PROPERTIES USED

Table: 2 Material Properties Used

S. No.	Properties	Value
1.	Young's Modulus (Ex)	2e5 N/mm^2
2.	Poisson's Ratio	0.3
3.	Density	7850 Kg/m^3

BOUNDARY CONDITIONS

- Base plates are constrained in all degrees of freedom.
- Head closure is bolted to chamber using Constraint equations – Simulating bolts.
- Internal pressure of 9 bar is applied.
- Gravity – 9810 mm/sec^2 is applied to simulate self weight.

3D MODEL OF THE CHAMBER USED FOR CONTACT ANALYSIS

Figure 2: 3D Model of the Testing Chamber Used for Contact

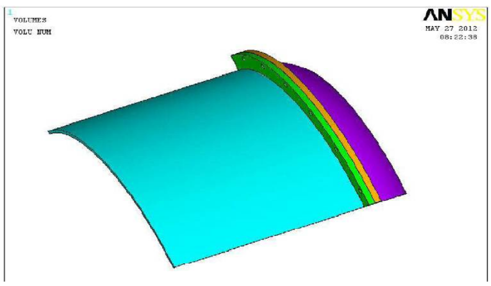


Figure 3: Mesh Model of the Chamber Used for Contact Analysis

Figure 5: Mesh Model of the Bolt Used for Contact Analysis

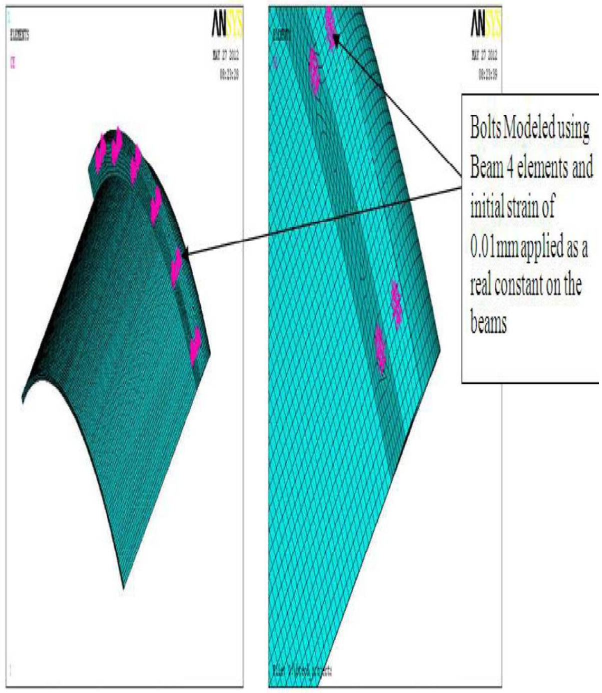


Figure 4: Bolts Created Using Beam 4 Elements

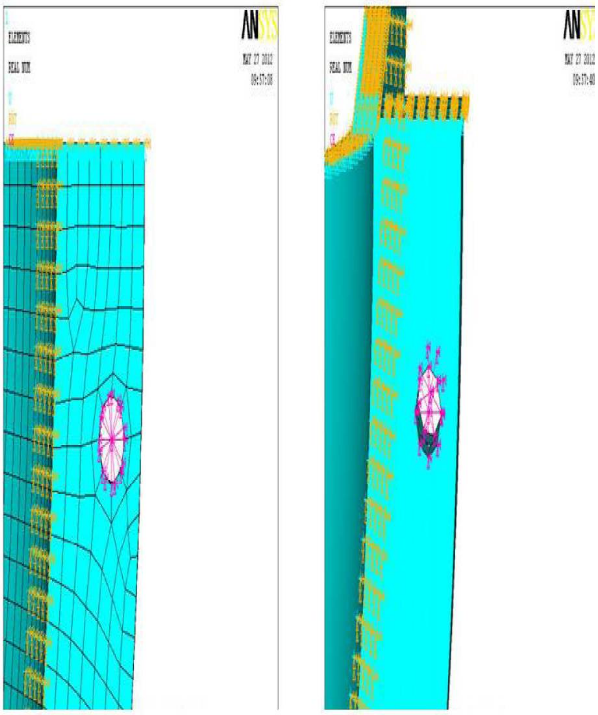


Figure 6: Boundary Conditions Used for Contact Analysis

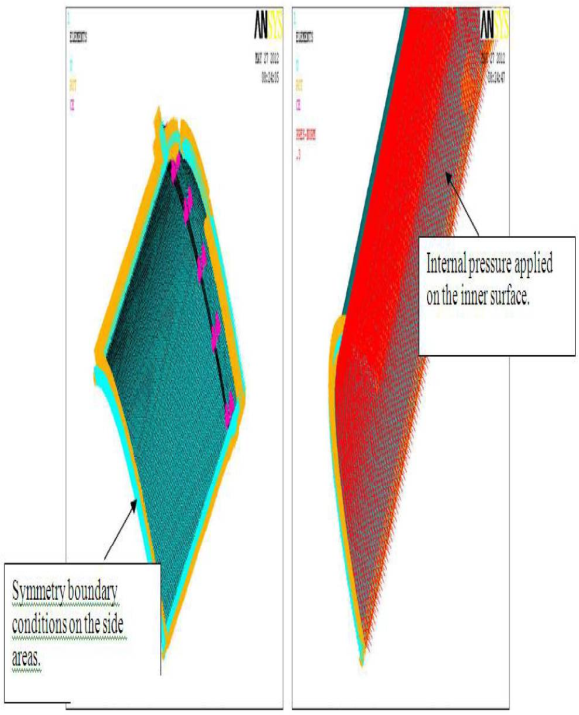
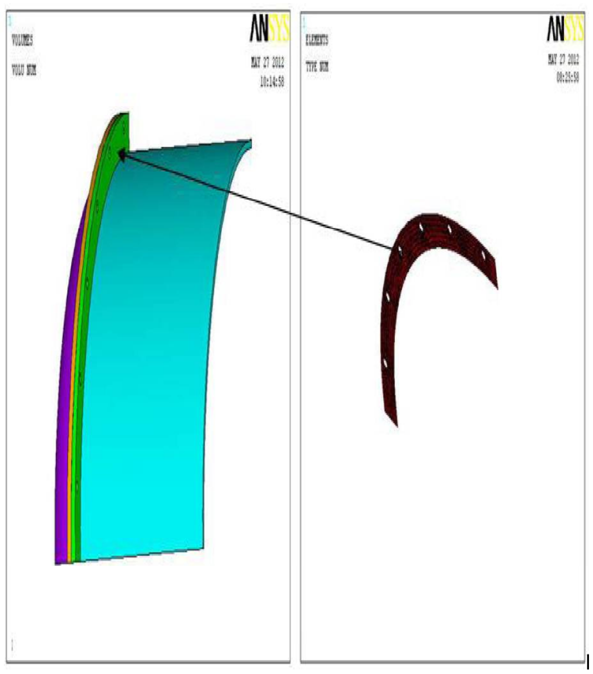


Figure 7: 3D Model of Contact Created Between the Mating Surfaces Near Bolted Region



RESULTS OF THE CONTACT GAP ANALYSIS FOR M36 BOLTS

Figure 8: Total Deformation for M36 Bolts

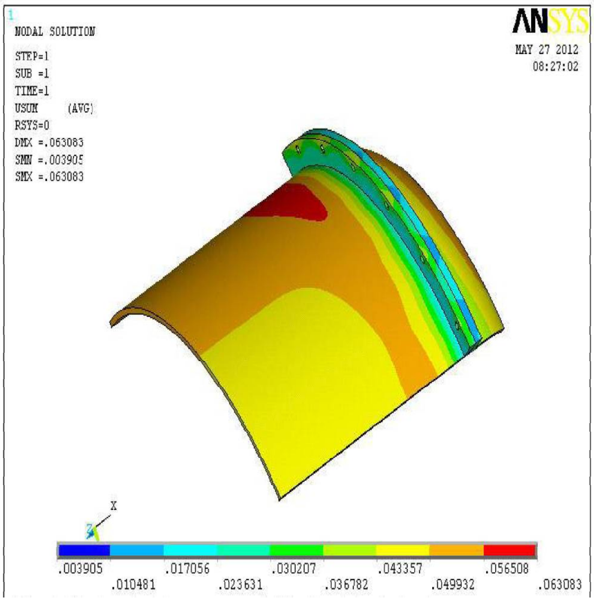


Figure 9: VonMises Stress for M36 Bolts

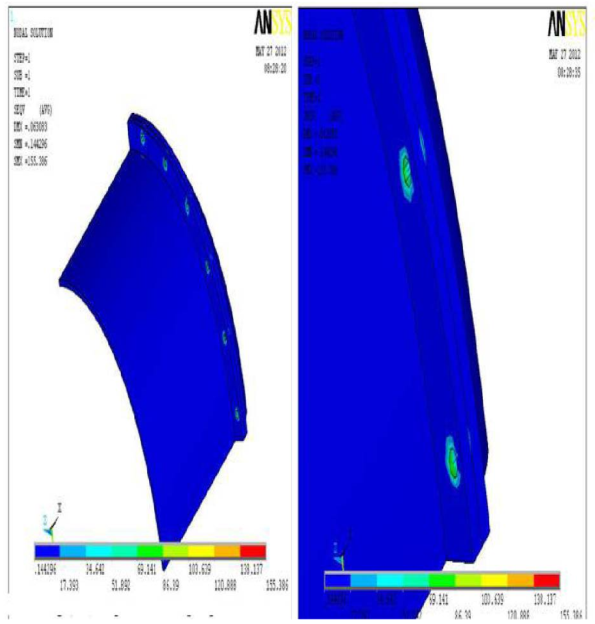
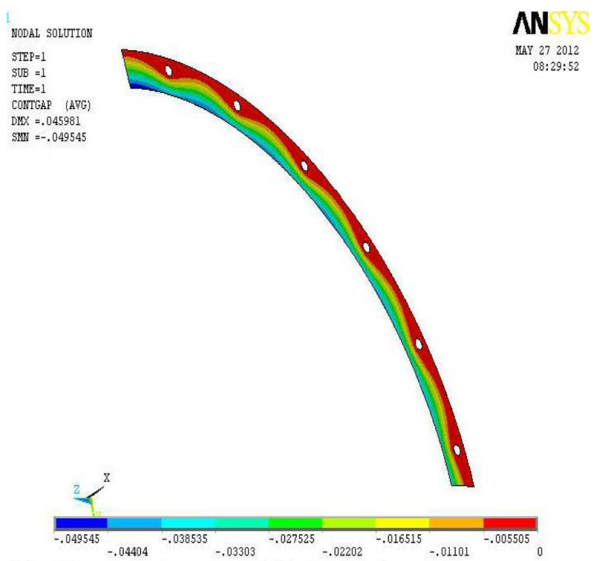


Figure 10: Contact Gap for M36 Bolts



From the above Figure 10, It can be observed that the Contact Gap is within the limit at the bolt region and there is no opening at the bolt locations. The contact status is also observed as sticking at the bolting regions. So, M36 bolt is the suitable bolt diameter to prevent the leakage of pressure to the atmosphere.

CONCLUSION

- Contact gap analysis is done for M36 bolts and observed the gap opening as Zero mm.
- Total deformation for M36 Bolts is 0.063 mm.
- VonMises Stress for M36 Bolts is 155.35 N/mm^2.
- Pre load of The M36 Bolts is 1508 N.
- Strain of The M36 Bolts is 0.00037 mm.
- From the analysis it is concluded that M36 bolts are recommended for canister testing chamber to avoid the pressure leakage to the atmosphere.

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