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DESIGN OF ULTRA HIGH VACUUM (UHV) SYSTEM TO EVALUATE OUTGASSING RATE OF THE MATERIALS TO ASSES THEIR UHV COMPATIBILITY

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ABSTRACT

Stainless steel and copper are very suitable structural metals for fabrication of UHV vessels. Indus-1 (450MeV) and Indus-2(2.5GeV) are electron storage rings dedicated for their application as a synchrotron radiation sources. In accelerators the residual gas particles limit the e-beam performance and its life time. The main factor deciding the UHV compatibility of any vacuum component is the specific outgassing rate, at room temperature and specifically at the operating temperature. To evaluate the outgassing rate of the various materials to be used in UHV, an experimental set-up made up of stainless steel has been designed, developed and tested. A thin cylindrical shell under external pressure, like a column, may collapse either by yield or by buckling. Collapse by yield is caused by stresses in the shell reaching the yield point of the material. Collapse by buckling is caused by buckling of the shell at stresses which may be considerably below the yield point. The shell strength i.e. the thickness of the considered experimental set – up chambers was designed on the basis of ASME code section VIII/Div-1, II/part-D. Orifice was designed as per AVS standard. The buckling analysis, deflection analysis and stress analysis were also performed on the experimental set-up chambers using ANSYS software and results were found up to the mark. This paper elaborates design calculations for chamber and orifice, outgassing- rate measurements and the ANSYS results.

KEYWORDS: Ultra High Vacuum, Sputter Ion Pump, Bayard Alpert Gauge, Turbo Molecular Pump.

INTRODUCTION

Sources of gas load in a vacuum system are, gas penetrates into the system as a result of leakage Q_L, gas resulting from the outgassing of the materials in the system Q_D, gas resulting from the vapor pressure of the material Q_V, gas entering the system by permeation through the walls Q_P.

Total gas load $Q = Q_L + Q_D + Q_V + Q_P$ [1]

In any leak-proof UHV system, the main factor controlling the ultimate pressure is the outgassing rate Q_D from the vacuum chamber.

The achievement of an ultimate pressure in the UHV range depends on reducing the outgassing load substantially by bake-out.

$Q = P \times S$

Where Q = Total outgassing rate (mbar – l/s),
S = Pumping Speed (l/s)
P = Ultimate pressure (mbar)

To measure the outgassing rate from the chamber / sample to be tested, known conductance method is used.

The formula is

Total outgassing rate $Q = C\,(P_1 - P_2)$

Where **Q** = Total outgassing rate (mbar – l/s)
C = known orifice conductance (l/s)
P_1 & P_2 = pressure on both sides of the orifice (mbar).

Specific outgassing rate of the chamber /sample ($q_{specific}$)

$q_{specific} = Q / A$ => $q_{specific}$ = specific outgassing rate of the chamber (mbar – l/s / cm ^{2})

A =total surface area of the chamber/ sample to be tested (cm^2)
Sources of gas load in a vacuum system depend strongly on the construction material of the system [1]**.**Thus the choice of materials with correct properties for the design of UHV system becomes an important criteria and for this ,evaluation of out-gassing rate of the materials , is a prerequisite .

To evaluate the outgassing rate of various materials to be used in UHV, an experimental set-up made up of stainless steel 304L has been designed, developed and tested so that suitable material could be used in the electron storage rings.

DESIGN OF EXPERIMENTAL SET-UP

Experimental set-up
The set-up (Fig1) is designed on the basis of the through-put method for the outgassing rate measurement, based on AVS standard and ASME code [2], [4], [5], [6] .Fig 1 shows the experimental set-up based on AVS standard.

Set-up consists of two cylindrical chambers made of stainless steel 304L. Both the chambers are separated by an orifice of known conductance **C**. Both the chambers are provided with

Bayard-Alpert gauges (B. A. Gauges) for pressure measurement i.e. $\mathbf{P}_1$ and $\mathbf{P}_2$ readings.

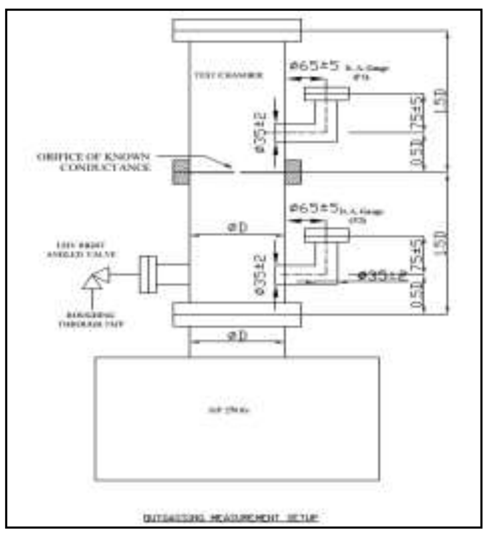

Fig 1 Experimental Set-Up

The lower chamber is provided with another port for connecting a Turbo molecular Pump (TMP) for roughing and pumping during baking.TMP is connected through a UHV isolation valve. The main pumping is carried out using a Sputter-ion pump (SIP) which is connected directly to the lower chamber.

DESIGN OF THICKNESS FOR THE VACUUM CHAMBERS (TEST CHAMBER AND AUXILIARY CHAMBER)

The failures in a chamber may have two types, namely, (i) Material failure and (ii) Form failure. In material failure stresses in the chamber exceed the specified safe limit, resulting in the formation of cracks which can cause failure. In Form failure, though Stresses may not exceed the safe value, chamber may not be able to maintain its original form. The chamber does not fail physically but may deform to some other shape due to intolerable external disturbance. Form failure depends on geometry and loading of the chamber. It occurs when the conditions of loading are such that compressive stresses get introduced [8].

When the magnitude of the load on a structure is such that equilibrium of the structure changes from stable to neutral, the load is called the critical load. This phenomenon of change of equilibrium is called the buckling of a chamber [8]. Vacuum chambers/vessels are external pressure vessels and failed in buckling.

Selected UHV pump for test set-up is Sputter-ion pump (SIP-270l/s).To get UHV in test set-up, it is required to bake the whole set-up at 250°C.

Detail of inlet port of SIP -270 l/s is

D = inner diameter =146.46 mm

Selected chamber dimensions are

D_o = Outer diameter = 152 mm (assumed as per ASME procedure)

Total Length of each chamber with flanges = 1.5 D = 1.5 x 146.46 = 219.69 mm

L = Length of cylindrical pipe used in test chamber

= 219.69-2x8 [8]

= 203.69 mm

$L / D_o = 1.34$

Designed external pressure on the outside of the chamber is 1atmosphere and vacuum exists inside it

And

Thickness of chamber t =? => To find out

PROCEDURE TO FIND OUT THICKNESS OF THE CYLINDRICAL VESSEL AS PER ASME CODE FOR THE EXTERNAL PRESSURE VESSELS [6]

Step 1- t = thickness = 0.5 mm (assume as per ASME procedure)

Step 2- L= length of shell = 203.69 mm

Step 3- $L / D_o = 1.34$, $D_o / t = 304$,

D_o = Outer diameter of shell = 152 mm

Step 4- Determine Factor "A" from ASME Code, section II, Part D, Subpart 3, using FIG. G: Geometric Chart for Components under External or Compressive Loadings (for All Materials)

Factor "A" = 1.87×10^{-4} (using Interpolation)

Step 5- Using Factor "A" determined in step 4 , enter the applicable material chart from ASME Code ,section II, Part D , Subpart 3,using FIG.HA-3 Chart For Determining shell Thickness of Components under External pressure Developed for Austenitic Steel 18Cr- 8Ni-0.037 Max. Carbon, Type 304L at the appropriate temperature 315 °C (>250°C) and get

Factor B = 14.66 MPa and E = 169×10^3 MPa

Step 6- As per ASME Code, section VIII, Division 1-Rules for construction of Pressure vessels-UG-28 / Thickness of Shells and Tubes under

External Pressure - subsection (c) – Cylinders having $D_o / t \geq 10$ as follows-

a) For values of Factor A falling to the right of the end of the material / Temperature line assume an intersection with the horizontal projection of the upper end of the material/temp. Line. Move horizontally to the right and read the value of Factor "B" and allowable external pressure Pa should be computed as follows.

$$P_a = 4B / 3 (D_o / t)$$

b) For values of Factor A falling to the left of the applicable material / Temperature line, The value of allowable external pressure P_a should be computed as follows.

$$P_a = 2A E / 3 (D_o / t)$$

Step 7- In our case Factor A falls on the applicable material / Temperature line 315 °C and Step 6- a) is required to evaluate the allowable external pressure P_a .

Step 8 -

$$P_a = 4B / 3 (D_o/ t)$$
$$= 4x\ 14.66 / (3 \times 304)$$
$$= 0.06429 \text{ MPa}$$

Step 9- Comparing P_a with P,

(Design Ext. Pressure P=1 atm =1.01325×10^5 Pa =0.101325 MPa)

$P_a < P$

Selected "t" is too small. Since this pressure (P_a) is considerably lower than the desired external pressure of 0.101325 MPa for full vacuum, the calculation is repeated with a greater thickness.

Let new thickness be t =1.0mm, L = 203.69 mm, D_o (outer diameter of shell) = 152 mm, D_i = inner diameter of shell = 150 mm,

D (mean diameter of shell) = 151 mm

$L / D_o = 1.34$, $D_o / t = 152$, T = 315 °C

Critical length of chamber

$L_c = 1.11\, D \sqrt{(D/t)}$

$= 2059.626 \quad >> \quad L$ => Intermediate cylinder

After following the previous step 1 to 9 as described earlier, we get

Factor A = 5.3×10^{-4}

Factor B = 29 MPa

$P_a = 4B / 3 (D_o / t)$

$= 4 \times 29 / (3 \times 152) = 0.254$ MPa ($> P$)

Thus Selected t = 1.0 mm is safe thickness as $P_a > P$. According to ASME CODE SECTION VIII- DIVISION – I - UG-16 (b)- minimum thickness permitted for shells and heads holding components shall be 1.6mm exclusive of any corrosion. Therefore, minimum thickness of cylindrical shell of outgassing set-up = 1.6mm.

As per ASTM A 312 /312M [3]for a particular outside diameter nominal wall thickness of the pipe increases as schedule number increases from 5 S to 80 S. Schedule no. 5 S specifies the 2.77mm thickness for the pipe having O.D. 141.3mm to168.28mm. As 152mm O.D. pipe does not exist in any schedule specify in it. This size (available commercially) exist in ASTM A 511 /A 511M –Standard specification for Seamless Stainless Steel Mechanical Tubing. Thus selected thickness of 152 O.D. pipe for the test chamber as well as auxiliary chamber (considering the reinforcement required for opening in cylindrical shell) is 2.77mm as both the chambers are identical.

Nozzle neck thickness [6]

As per UG-45 (b)(2) Nozzle neck thickness is obtained by using the external design pressure as an equivalent internal design pressure (assuming joint efficiency E=1.0) in the formula for the shell at the location where the nozzle neck attaches to the vessel but in no case less than the minimum thickness 1.6 mm specified in UG-16 (b).

As per UG-27 (c) (1) (thickness of shell under internal pressure)

For circumferential stress,

Nozzle thickness. $t_n = P.R_n / (S.E - 0.6P)$

Where P =100kPa (atm) R_n = Nozzle radius = 17.5mm, E = Joint efficiency = 1.0

S = max. Allowable stress value at design temp. 300°C =97.7 MPa (for S.S.304L)

t_n = 0.018mm <1.6mm

So thickness of nozzle as per ASME code = 1.6 mm

As per ASTM A 312 /312M and ASTM A511/A511M, pipe size 38.3mm O.D./thickness 1.65mm exists commercially. So selected thickness of nozzle is 1.65 mm.

Limits of reinforcement required for opening in cylindrical shell

Fig 2 shows the limits of reinforcement required for opening in cylindrical shell [6].

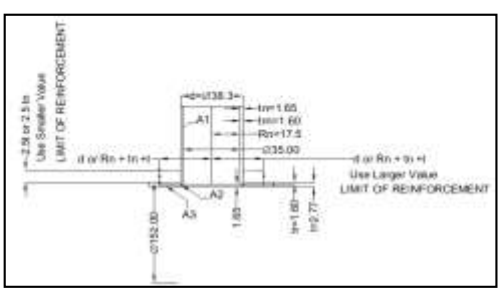

Fig 2 limits of reinforcement for opening in cylindrical shell [6]

In fig 6.5, t_n= nozzle thickness =1.65mm,
t_{rn}= required nozzle thickness as per ASME code =1.60 mm
t = Shell thickness = 2.77mm
t_r = required shell thickness as per ASME code =1.60mm
d =finished diameter of circular opening =38.3mm
Selection of limit of reinforcement as per ASME code section VIII/Division 1/UG-40

a. $2.5t = 2.5x2.77 = 6.925mm$
$2.5t_n = 2.5x1.65 = 4.125mm$ (selected as it is smaller value)
b. $R_n + t_n + t = 17.5 + 1.65 + 2.77 = 21.92mm$
d = 38.3mm (selected as it is larger value)

As per ASME code section VIII/Division 1/UG-37(d),the total cross-sectional area of reinforcement (shall not be less than) $A = d.t_r/2$

$$A = d.t_r/2 = 38.3x1.6/2 = 30.64mm^2$$

Area available for reinforcement without a pad includes:

$\mathbf{A_1}$ =the area of excess thickness in the nozzle wall
$= 2x\,(2.5t_n + t- 1.65)\,(t_n - t_{rn})$
$= 2x\,(4.125+2.77-1.65)\,(1.65-1.60)$
$= 0.53mm^2$

$\mathbf{A_2}$ = the cross-sectional area of the weld
$= 2x\,(1.65x1.65/2)$
$= 2.72mm^2$

$\mathbf{A_3}$ = the area of excess thickness in the cylindrical shell
$= 2x\,(d-R_n-t_n)\,(t-t_r)$
$= 2x\,(38.3-17.5-1.65)\,(2.77-1.6)$
$= 44.81mm^2$

$A_1+A_2+A_3$ (=48mm^2) > A (=30.64mm^2) => the opening is adequately reinforced and no reinforcing pad is required.

Imperfections and their effect on the buckling strength of cylinders

In flexible structures thin walled vessels, imperfections such as out of roundness can dramatically reduce the buckling load .It is mandatory that an applicable tolerance or standard be first established for design and then rigidly adhered to throughout construction. As per ASME Boiler and pressure vessel code UG-80 / permissible out of roundness of cylindrical shell / (b) External pressure, the shell of a completed vessel shall be substantially round and shall meet the following requirements-

a. The difference between the maximum and minimum diameter at any cross section shall not exceed 1% of the nominal diameter for that cross section. This is for the purpose of eliminating general out of roundness.
b. When the cross section passes through an opening, the permissible difference in diameters given above may be increased by 2% of the inside diameter of the opening.

Orifice design as per AVS standard

As per AVS standard, the orifice should be large enough so that net speed is at least ten times the pumping speed of the gauge if the latter has an appreciable pumping action for the gases liberated by the sample (e.g. ionization gauge) [4].
Pumping speed for a nude ion gauge (Varian make Bayard-Alpert gauge) is typically around 0.5 l/s
i.e. $S_{ORIFICE} \geq .5x10$ l/s > 5 l/s
B.A.Gauge pumping effect can be explained as follows-
a) After gauge has been outgassed, gas entering the B.A. Gauge tube is readily adsorbed especially on the tube walls.
b) Chemical reactions induced by hot filaments produce further sorption.
These processes are responsible for the pumping action of gauges, which causes the pressure at the gauge to be lower than that in the system.
When a high vacuum test chamber is connected to a pumping system by an orifice or tube whose conductance is less than 5% of the speed of the pumping system, the net pumping speed in the test chamber is independent of pressure and variations in the speed of the pumps.

Selected UHV pump for the test set-up is SIP*-270 l/s.

Speed of SIP in the range of 10^{-10} mbar $S_P = 5\ \% \times 270$ l/s =13.5 l/s

Thus, 5 l/s < $C_{ORIFICE}$ < 13.5 l/s

=> 7.4mm < $D_{orifice\ dia.}$ < 12.1mm

Let selected $C_{ORIFICE}$ = 9 l/s

For Air, at 295K i.e. at room temperature,

C_o= conductance of an orifice [9] = 11.6 A l/s Where A in cm^2

C_o= 9.1 D^2 l /s (for a circular orifice of diameter D cm)

C_o= 9.1 D^2 l /s = 9 l/s => D =0.994 cm = 9.94mm

Selected

$C_{ORIFICE}$=9l/s=C_o

= > D = Orifice diameter = 9.94 mm

Relation between orifice diameter and its length as per AVS standard [3]

$L \leq 0.02D$

$L \leq 0.02 \times 9.94$

$L \leq 0.198$ mm

Fig 3 represents the details of the orifice designed in 160 CF ConFlat flange welded in auxiliary chamber to keep the conductance error less than 1.5%.

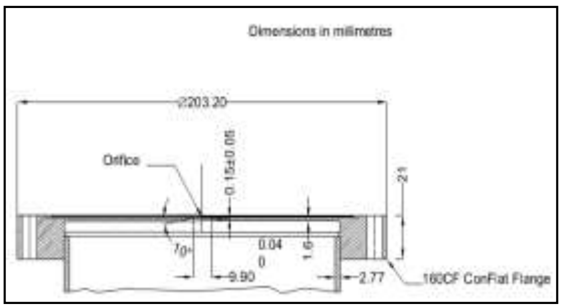

Fig 3 Orifice designed in 160CF ConFlat Flange

For making demountable joints for the UHV test set-up, ConFlat flanges (SS304L) have been used [7].

Fig 4 represents the auxiliary chamber designed as per ASME code and AVS standard.

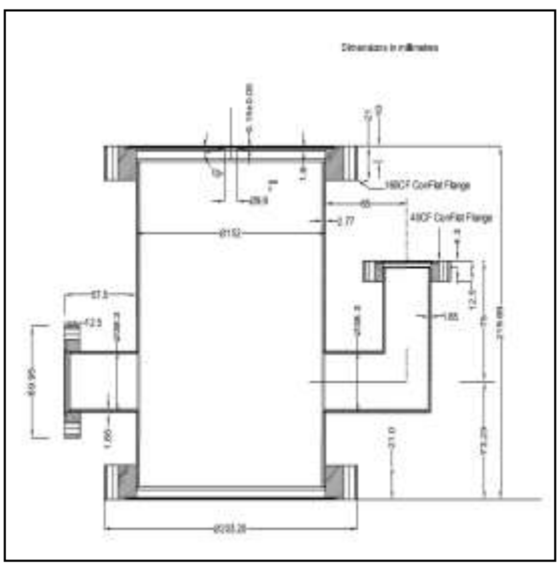

Fig 4 Designed auxiliary chamber

Fig 5 represents the test chamber designed as per ASME code and AVS standard.

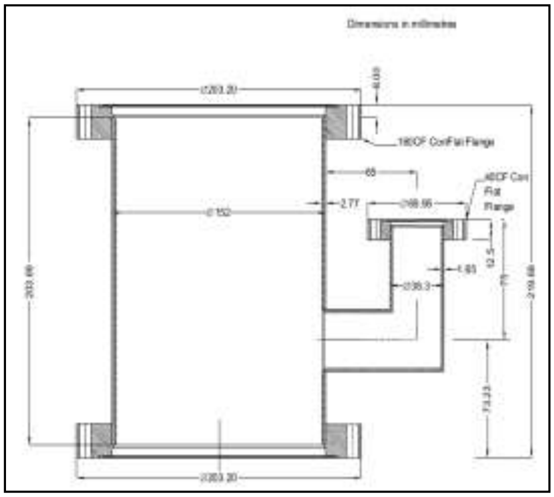

Fig 5 Designed Test Chamber

QUALIFICATION OF FLANGED WELDING DONE IN THE TEST CHAMBERS AND AUXILIARY CHAMBER

All the flanges were welded from the inside to the respective chambers through TIG welding. For weld preparations, cleanliness is a must. All the welded joints were leak tested through helium leak detector and found leak proof 2x 10^{10}mbar .l/s.

UHV TESTING AND OUTGASSING RATE MEASUREMENT OF TEST SET-UP

To qualify the UHV testing of test set-up, base pressure measurement test was performed. After roughing through TMP or 24hrs, whole set-up was baked at 250°C for 24 hrs. Using flexible heating tapes (3kW). Ultimate vacuum observed in B.A.gauges P_1 and P_2 was 3.8×10^{-10} mbar and 2.5×10^{-10} mbar respectively. Total inside surface area of the test chamber'A'is 1370 cm^2 .Orifice conductance is 9 l/s. Total outgassing rate of the test chamber [calculated using $Q = C.(P_1 - P_2)$] was 1.17×10^{-9} mbar.l/s and specific outgassing rate of the chamber (using $q_{specific} = Q/A$) was 8.5x 10^{13}mbar.l/s/cm^2, after baking followed by 24 hrs. of pumping.

STATIC AND BUCKLING ANALYSIS FOR DESIGNED TEST CHAMBER AND AUXILIARY CHAMBER THROUGH ANSYS

CAD MODEL AND INPUTS

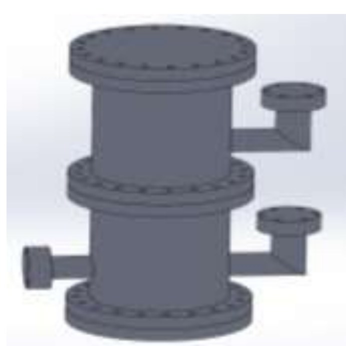

Fig 6 CAD Model of test camber and auxiliary chamber

Material:

- Austenitic stainless steel AISI 304L
- Modulus of Elasticity E = 193 GPa
- Poisson's Ratio = 0.3

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- Yield Strength = 170 MPa

Loads: 1 bar pressure on all surfaces exposed to atmosphere
Constraints: Bottom flange fixed.
Analysis carried out:
Static Analysis: To evaluate stresses and deformations for designed wall thickness.
Buckling Analysis: To evaluate Buckling Load Factor and Mode Shape.

Stress Distribution in chambers

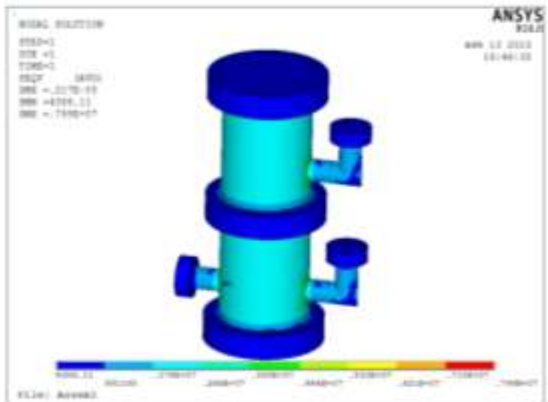

Fig 7

Fig 7 shows the Stress Distribution in chambers. The maximum Stress developed 8 MPa which is well below the yield stress σ_y (170Mpa) of the material. Maximum stresses developed at shell port junction

Deformation Pattern in chambers

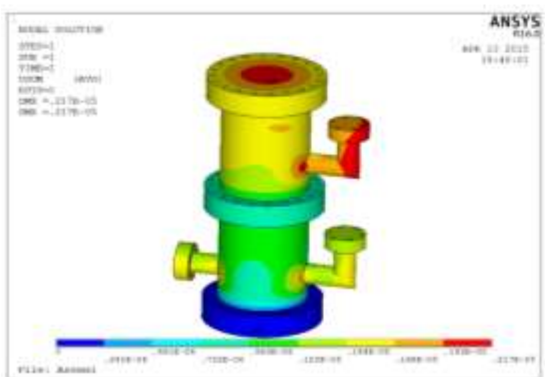

Fig 8

Fig 8 shows that maximum deformation is 2.2 microns. Maximum deformation are developed at shell port junction.

Buckling Analysis of Chambers

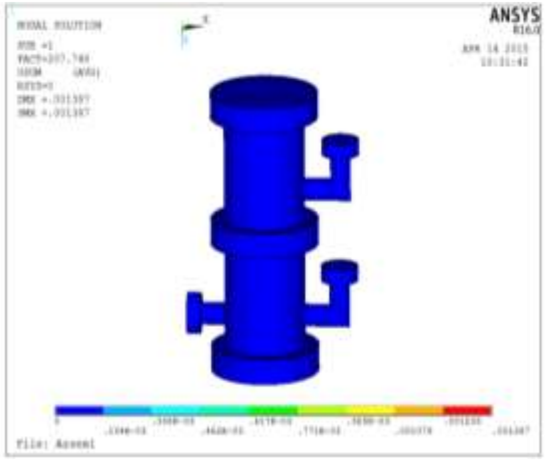

Fig 9

Fig 9 shows the buckling Analysis of Chambers Buckling load factor is found to be 207

RESULTS AND CONCLUSION

- During design of vessel, the vessel should withstand with the stresses developed due to 1atmospheric pressure as well as it should not buckle due to external pressure.
- During design we have considered standard wall thickness 2.77mm for cylindrical shell as per ASTM standard i.e. ASTM A 312 /312M, A 511 /A 511M –Standard specification for Seamless Stainless Steel Mechanical Tubing.
- The maximum Stress developed 8 MPa which is well below the yield stress σ_y (170MPa) of the material.
- Buckling load factor is found to be 207.
- Finite element analysis results show that design is safe and acceptable.
- To qualify the UHV testing of test set-up, base pressure measurement test was performed. Specific outgassing rate of the test chamber was 8.5x 10^{-13}mbar.l/s/cm^2, after baking (at 250°C) followed by 24 hrs of pumping.

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