Indian Journal of Engineering & Materials Sciences
Vol. 30, August 2023, pp. 542-549
DOI: 10.56042/ijems.v30i4.2230

Sustainable Subsea Pressure Housing for Shallow Water Applications

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Received: 01 June 2023; Accepted: 1 August 2023

The design of subsea pressure housing (SPH) with parameters such as hydrostatic external pressure, temperature and salinity prevailing in the marine environment requires a proper selection of materials. The housing is designed to accommodate the data acquisition electronics that is capable of handling a higher sampling rate and a larger capacity power pack for subsea ambient noise measurement mooring system operating at an ocean depth of 100 m in the shallow waters of the Indian coast. The scope of the housing design includes sustenance and operation of the system for a minimum of six months in an open ocean environment withstanding extreme events like cyclones and storm surges prevalent in the sea. The main aim of this paper is to present the work carried out on the design of SPH with three different materials such as high-strength stainless steel (SS316L), aluminium alloy (Al6061-T6) and titanium alloy (Ti-6Al-4V) for shallow-water applications. The design of SPH in a cylindrical shape with an internal diameter of 0.33m (330 mm) and length of 0.7m(700 mm) to withstand a pressure of 1MPa (10 bar) and hoop, axial and Von Mises stresses has been accomplished as per the American Society of Mechanical Engineers (ASME) code/standards. The buckling of cylindrical pressure housing and principle stresses were calculated through a finite element analysis (FEA) using ANSYS software.Further comparative studies on three materials mentioned above on the criteria of maximum stress, critical buckling pressure, formability, corrosive properties and cost of fabrication were carried out. The pressure housing with SS316L material is found to be an optimal choice for shallow-water applications in the open ocean. The subsea housing thus designed has been fabricated with SS316L and operated in the shallow waters of Goa successfully for a period of six months continuously.

Keywords: Hydrostatic external pressure, Shallow water, Subsea pressure housing

1 Introduction

Subsea pressure housing (SPH) is used extensively in underwater systems such as manned and unmanned underwater remotely operable vehicles (ROV), passive acoustic monitoring systems and other autonomous systems for measurements and operations in the sea. In this work, SPH has been designed for an autonomous ambient noise measurement system (ANMS). The purpose of SPH is to accommodate the electronic subsystems and power pack used for ANMS. In India, under the Ocean Acoustics programme of National Institute of Ocean Technology (NIOT), an autonomous ambient noise measurement system was indigenously developed for time-series measurements of ambient noise in the shallow waters of the Indian seas. The housing was designed to accommodate the data acquisition system with a sampling of 50 kHz and a power pack of 544 Ah to endure a minimum of six months against extreme events like cyclones and low pressures at the sea. The SPH is designed for withstanding an external hydrostatic pressure of 1MPa (10 bar) to carry out operations up to a depth of 100 m in the ocean. The selection of shape, size and material are very important while designing the housing for withstanding 1MPa pressure. Buoyancy and length to diameter ratio (*L*/*D*) also play a major role in the design. If not selected properly, buckling and geometrical instability will occur in the pressure housing. The SPH with a cylindrical shape is mostly chosen for the axial or longitudinal load application. The housing is designed with an internal diameter of 0.33m (330 mm) and a length of 0.7m (700 mm). Material selection becomes an important part of the design for the resulting system to have good corrosion resistance, high strength-to-weight ratio, ability to withstand external hydrostatic pressure and longer durability. Three materials – stainless steel 316L (SS316L), aluminium 6061-T6 and titanium Ti-6Al-4V were considered for the design and development of SPH for subsea applications in shallow-water regions. The non-metallic pressure housing made of a hybrid of PVC and acrylic is often a good alternative to heavier and more expensive metallic materials. But

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the sustainability of non-metallic pressure housing with the required size and shape for long-term measurements is not possible with its gasket design for shallow-water applications. Usually, the pressure housing or pressure vessel for subsea applications or offshore structures are governed by design guidelines, codes like ASME, IS and standard design using thin cylinder or thick cylinder theory. In ASME section VIII, Division 'I' the pressure vessel design is described for external pressure application, especially for subsea purposes[1]. Proper selection of materials, design condition for bolting, sealing, type of flanges, end covers and fasteners for the pressure vessel were studied and applied in the design. Teon Beng Koay et al. developed a self-recording shallow water acoustic logger for an ocean depth of 100 m and a cylindrical pressure housing made of aluminium was used to accommodate acoustic data acquisition electronics equipment having a battery capacity of 230 Wh. The system was operated for 10 days[2]. Andrew P.F. Little et al. observed that the pressure vessel must not only be capable of withstanding the pressure at the required depth but must also have suitable characteristics for factors like resistance to corrosion, high strength-to-density ratio, operating life span of the materials and low cost of fabrication. The pressure vessel with an axisymmetric model with boundary conditions was also described[3]. KhairulIzman Abdul Rahim et al. discussed the design of a pressure hull for an underwater inspection robot for shallow-water applications under an external hydrostatic pressure of 4 MPa. The pressure hull is a circular cylinder shell made of aluminium alloy 6061-T6 and a laboratory experimental study was carried out on this system[4]. Christopher Bassett et al. carried out passive acoustic recordings using a Loggerhead DSG data acquisition system, contained in a pressure case of non-metallic material, at 60 m water depth[5]. KhairulIzman Abdul Rahim et al. discussed the conceptual design of a pressure hull with hybrid composites with metal liner concept for underwater application with an external hydrostatic pressure of 4 MPa and carried out laboratory studies[6]. Joseph H. Haxel et al. designed a passive acoustic mooring system as an acoustic data logging system housed in a titanium pressure case for a rated ocean depth of 55 m[7]. V. Vullo notes that circular cylinders are generally divided into two families according to the equations governing their stress state. Geometrical axisymmetry and the uniform distribution of stresses through the wall thickness mean the radial, circumferential and axial directions are the principal directions of both stress and strain[8]. Michalski Jan presents a unique engineering method for the preliminary design of marine echo-location systems. The design problem that was addressed consists of determining geometrical parameters and selecting structural materials for a vessel of buoyancy necessary to house measuring instruments of a given mass, as well as its maximum operational depth[9]. A.M. Kamal et al. modelled the seawater reverse osmosis (SWRO) pressure vessels using an analytic solution and finite element modelling for stainless steel 316L and fibre-reinforced composite materials to optimise the PV design parameters. For SS 316L pressure vessel, the same wall thickness is obtained for both analytical solution and finite element method (FEM) for each vessel size with an acceptable safety factor error. The pressure vessels were designed on the basis of buckling analysis as well as structural yielding analysis and the safety was checked by finite element analysis (FEA) and analytical formula[9]. Pressure housing for underwater applications by K. Breddermann et al. was built in an additive manufacturing process from titanium and ceramic. These housings show promising performance and have a satisfying weight-to-displacement ratio[10]. A passive acoustic observation system used to study the effects of impact pile driving on the fish chorus MM Mahanty et al. consists of a data acquisition system and power pack enclosed in underwater pressure housing made of SS316L material[11]. The design and analysis of a pressure vessel were done by Madhavan Namboothiri et al.with titanium grade 2 material for undersea applications with an operating depth of 200 m[12]. Finite element analysis of type III pressurised cylinder was performed by Valter Luiz et al. to the internal working pressure of 300 bar and up to 600 bar of external pressure for different design and safety factors[12]. In this work, the focus has been to design, develop and operate a subsea pressure hull for accommodating a power pack and a data acquisition system with a high sampling rate for acoustic measurements over a longer period in a moored system in shallow waters up to 100 m depth, which can be operational even during extreme events like cyclones and storm surges.

2 Materials and Methods

2.1 Methodology

The housing is designed with conventional materials using ASME code Section VIII Division I, Section II Part D. Theories of failure were also considered and the thin cylinder theory was chosen for the design. The materials for building the pressure housing considered here include stainless steel SS316L, aluminium 6061 and titanium alloy-Ti-6Al-4V. The length of the cylinder was chosen as 0.7m (700 mm) and the inner diameter of the housing was set at 0.33m (330 mm) as per user requirements. The analysis of the pressure housing was performed with finite element analysis software ANSYS. The failure theory of maximum distortion energy theory was employed to compare the theoretical and analytical calculations of the pressure housing. Critical buckling of the pressure housing was calculated for all the three materials and the values were compared.

2.2 Design of pressure housing with ASME code

The subsea pressure housing as shown in the Figure 1 is assumed as a cylinder with both end capped and the materials chosen as SS316L, Aluminium 6061-T6 and Titanium 6Al-4V for operations up to water depth of 100 m in Open Ocean. The properties of the materials are listed in Table 1. The theoretical calculation of the pressure housing for the three different materials are shown in Table 2 and Table 3

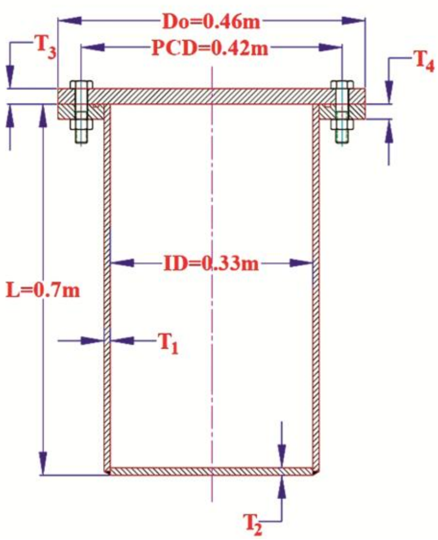

Fig. 1 — Sectional view of pressure housing.

2.2.1 Calculation of cylinder thickness:

The pressure housing dimensions are as follows:

Di- Inner Diameter, t- housing thickness, Do- Outer diameter, L-Length and p -design pressure as1MPa(10 bar)

$$\text{Ratio -1 (R1)} = \frac{L}{D_o} \quad \ldots (1.1)$$

$$\text{Ratio-2(R2)} = \frac{D_o}{t} \quad \ldots (1.2)$$

Using Eq. (1.1) & (1.2) Factor A and Factor B willbe obtained for the three materials with respective Chart in sub part of ASME, Section II, and Part D. The calculated values for the respective materials are listed in Table 3.The maximum allowable working pressure (Pa) is arrived as per ASME Section VIII Division 1 as,

Table 1 — Material Properties of the Pressure Housing

Properties	SS316L	Al6061	Ti-6Al-4V
Density(ρ), kg/m^3	8000	2700	4480
Young's Modulus(E), GPa	193	69	117
Poisson's ratio (γ)	0.27	0.33	0.31
Tensile yield Strength(σ_y), MPa	290	276	760
Tensile Ultimate Strength (σ_U), MPa	580	310	830

Table 2 — Calculated parameters of the pressure housing as per ASME code

Materials	D_o	T_1 (t)	R1	R2	Factor-A	Factor-B	P_a	T_2	T_3& T_4
SS316L	0.341	0.0055	2.05	68	0.0010	9000	1.27	0.018	0.023
Al6061-T6	0.343	0.0065	2.04	57	0.0011	7000	1.21	0.023	0.029
Ti-6Al-4V	0.342	0.006	2.05	62	0.0015	10875	1.20	0.023	0.029

Where L=Length of cylinder=0.7m, D_o= outer diameter, T_1=t= Housing wall thickness in m, Factor A & Factor B calculated with ASME Chart for respective materials based on R1 and R2, R1=L/D_o and R2=D_o/t, P_a= Allowable working pressure in MPa, T_2=End cover thickness, T_3& T_4 = Flange thickness in m

Table 3 — Calculated pressure housing dimensions

Material	σ_h	σ_l	σ_v	F_A	F_D	P_s
SS316L	34.22	17.25	29.79	1160	3874.73	12704.02
Al6061-T6	29.12	14.67	25.21	1180	3898.27	12772.79
Ti-6Al-4V	31.46	15.85	27.24	1176	3876.80	12804.04

Where, σ_h= Hoop stress in MPa,σ_l = Longitudinal Stress in MPa, σ_v = Von-mises Stress in MPa, F_A= Axial force in N, F_D=Drag force/Area in N, P_s= Pressure acting over the surface area in N/m^2

$$P_a = \frac{4B}{3\,(D_o/t)}$$ … (1.3)

If the maximum allowable working pressure (P_a) is greater than the design pressure (P), the Housing is safe for working at 10 bar or1MPa external pressure corresponding to 100 m water depth. A corrosion allowance of 0.0005m (0.5mm) will be added.

2.2.2 Design of end cover

A circular flat cover is chosen and the thickness of the end cover is calculated as per UG-34 of ASME Section VIII Division 1. The minimum required thickness (t) is given by:

$$t = d\sqrt{\frac{CP}{SE}}$$ … (1.4)

Where, d is the inner diameter of housing, C= 0.33, P is the design pressure= 145 Psi or 1.0MPa,

S = Maximum allowable stress value in Psi/MPa, E =0.9 for Joint Efficiency and t= thickness in m.

2.2.3 Design of Flange

The sealing of the flanged connection is provided by 'O' ring made EPDM elastomer. Hence Gasket Factor (m) and Gasket Seating Stress (y) will be zero. Since m and y are zero, the flanges have to be designed for bolting up condition. Bolt material selected is SA 193 Gr B8M and allowable stress for the above material is (F_b) = 16700 Psi = 115.17E+06 N/m^2

Assuming 12 numbers of M12 bolts for bolting of the flanges,

$$\text{Bolting Area,} A_b = \frac{\pi}{4} d^2 = \frac{\pi}{4} 12^2 = 0.001356$$ … (1.5)

$$\text{Design Load (W)} = F_b \times A_b$$ … (1.6)

Design load (W) for bolting condition is obtained by substituting the value F_b and A_b- in Eq. (1.6). W= 11.51 x 1356.91 = 15618.03 kg

$$\text{Bolt Circle Diameter (BCD)} = \text{Housing OD} + 2\,(1.41\,g_o + R)$$ … (1.7)

Where, $g_o$is thickness of the hub at back of flange=thickness of cylinder

R is the minimum radial distance of the bolt = 30 mm=0.03m

Substituting the values of (g_o) &(R) in Eq. (1.7), BCD is

$$\text{Flange OD} = \text{BCD} + 2\,E$$ … (1.8)

Where, E= the Edge distance from the Bolt circle diameter and E=20.0mm=0.02m,

'O' ring groove dimensions are selected from standard manual (IS 9975, 1984) for 'O' ring subjected to external pressure and for axial static sealing application[13].

'O' ring groove inner diameter (O-ring diameter) =0.366m

Groove Width =0.97±0.010m, Groove depth =0.572±0.05m

$$\text{Corresponding lever arm } h_G = \frac{1}{2}(C-G)$$ … (1.9)

Where, C is the bolt circle diameter, G=the mean diameter of the 'O' ring, G=0.3757m

$$\text{Flange Moment (M)} = W \times h_G$$ … (1.10)

Flange Thickness is calculated (Dennis Moss, 1987) with[14]

$$t = d\sqrt{\frac{CP}{SE}} + 1.9\frac{Wh_g}{SEd_3}$$ … (1.11)

The parameters calculated for the pressure housing for the respective materials as per the ASME code are listed in Table 2.

2.2.4 Calculation of stresses with thin Cylinder theory

The pressure housing is subjected to hydrostatic external pressure of 1MPa (10bar) and the free body diagram of the housing with all the forces and pressure for the subsea environment is listed in the Figure 2[15]. If the cylinder wall thickness is less than one tenth of its diameter then it is called as thin cylinder. The various stresses like hoop stress, axial

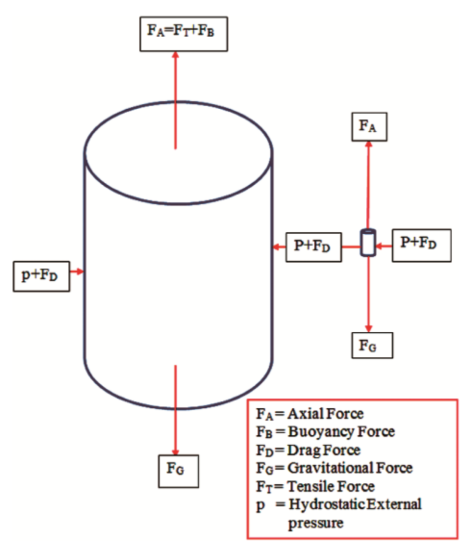

Fig. 2 — Free body diagram of pressure housing.

stress and Von-mises stresses are calculated with thin cylinder theory. In addition to that, drag force due to underwater current and axial load acting on the mooring of the system are also calculated. The calculated values of the various stresses are shown in Table 3

2.2.5 Buoyancy Force

$F_B=\rho\times v$=kg or N,
where ρ=density in kg/m^3 and v= volume in m^3

… (1.12)

F_T = The upward balancing-tensile force acting on the pressure housing due to the mooring system,Axial Force(F_A) = F_B+F_T

2.2.6 Drag force/Area

$$F_D=\frac{1}{2}\times\rho\times C_d\times A\times v^2 \quad \text{… (1.13)}$$

$$\text{Where, } A=\frac{\pi}{2}\times d\times l \quad \text{… (1.14)}$$

A= Surface/frontal area of pressure housing, C_d=1.2, Coefficient of drag,ρ =1025kg/m^3,

Density of sea water, and, v= velocity of underwater current in m/s, F_D = Drag force

2.2.7 Pressure acting over the surface area (Ps)

The axial load acting /area of cross section is as follows

$$P_S=\frac{F_A}{\pi/4\times d^2} \quad \text{… (1.15)}$$

Where, F_A is the axial force acting over the area, d is the diameter of the pressure housing

2.2.8 Longitudinal or Axial stress (σl)

$$\sigma_l=\frac{pd}{4t}=\frac{(p+p_S)\times d}{4\times t} \quad (1.16)$$

Where, σ_l is the longitudinal stress, MPa, and t is the wall thickness of the pressure housing

2.2.9 Hoop stress (σh)

$$\sigma_h=\frac{pd}{2t}=\frac{(p+F_D)\times d}{2\times t} \quad \text{… (1.17)}$$

Where, σ_h = Hoop stress, MPa, and F_D= Drag force acting/area

2.2.10 Calculation Von-mises Stress (σ_V)

$$\sigma_V=\sqrt{(\sigma_h)^2-(\sigma_h\times\sigma_l)+(\sigma_l)^2}\text{, MPa} \quad \text{… (1.18)}$$

Assuming the factor of safety is 1.2 and total design pressure is 1.2x 1.0=1.2 MPa. If the Hoop stress (σ_h) as per strength of material is lesser than the maximum allowable stress of the respective material as per ASME code then the design will be safe.The calculated values of the pressure housing as per the ASME code and thin cylinder theory for three materials are presented in Table 3.

2.3 Modeling of pressure housing

The pressure housing is modelled as a three-dimensional symmetric or axisymmetric body with an element size of 0.002 units. In an axisymmetric model, it was pinned in the Y direction at a point of the section that is to be revolved for the selected materials/metallic alloy like SS 316L, Al6061 and Ti-6Al-4V. The boundary condition for the subsea pressure housing is shown in Fig. 3. The cylinder was subjected to an external hydrostatic pressure of 1.2 MPa at on all four sides. The X, Yand Z translations were arrested at both ends. Since cylindrical pressure housing is an axisymmetric model, it was pinned in the Y direction at a point of the section that is to be revolved. An axisymmetric cylindrical model was created for the pressure housing with solid-8 node 183 elements in the FEA (ANSYS, 2013) software[16]. The symmetric boundary condition was applied to the model for external hydrostatic pressure of 1.2 MPa. The drag force and gravity forces were also applied to the pressure housing in the *Y*-direction and the calculated forces are given in Table 3. Hoop stress developed for the corresponding external hydrostatic pressure for the materials SS316L, Al 6061 and Ti-6Al-4V are shown in the Figures 4.0-9.0.

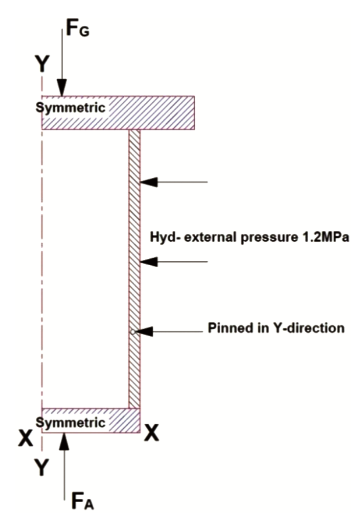

Fig. 3 — Boundary condition for the pressure housing.

Table 4 — Calculated buckling pressure of housings

Material	Design pressure (MPa)	Critical buckling pressure(MPa)
SS316L	1.20	12.2
Aluminium 6061-T6	1.20	8.70
Ti-6Al-4V	1.20	4.8

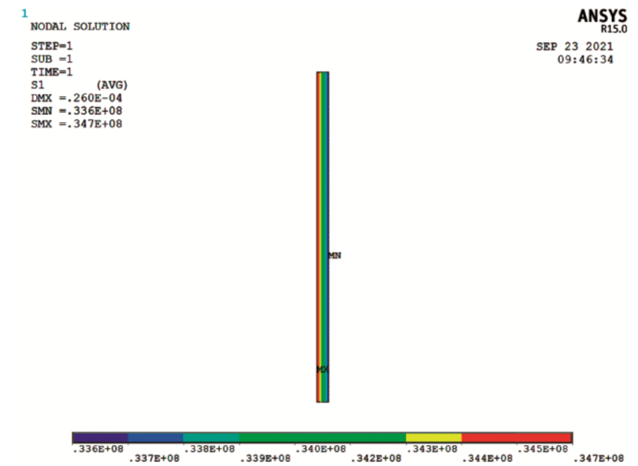

Fig. 4 — Hoop Stress for SS316L pressure housing.

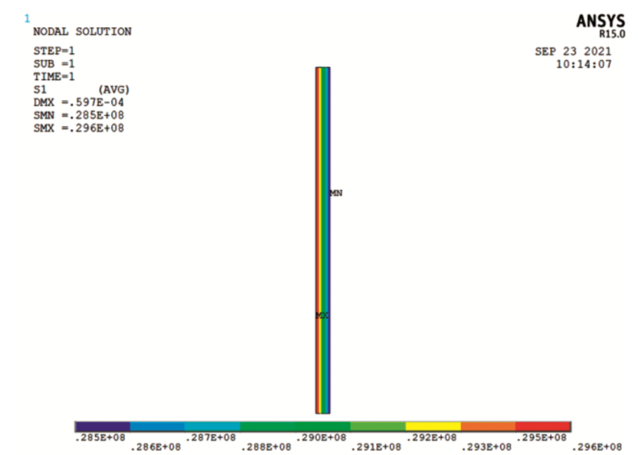

Fig. 5 — Hoop Stress for Al-6061 pressure housing.

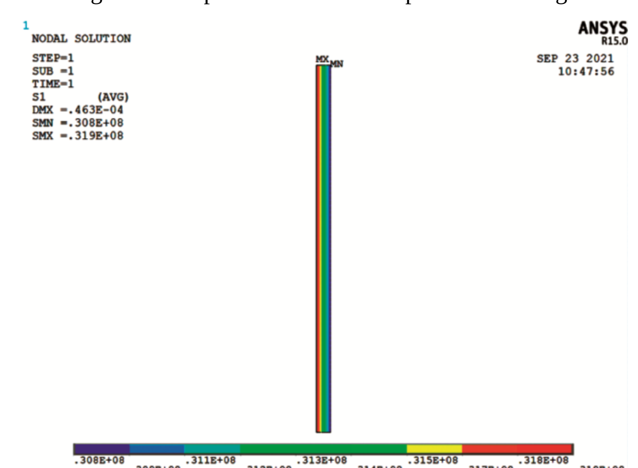

Fig. 6 — Hoop Stress for Titanium 6Al-4V Pressure housing.

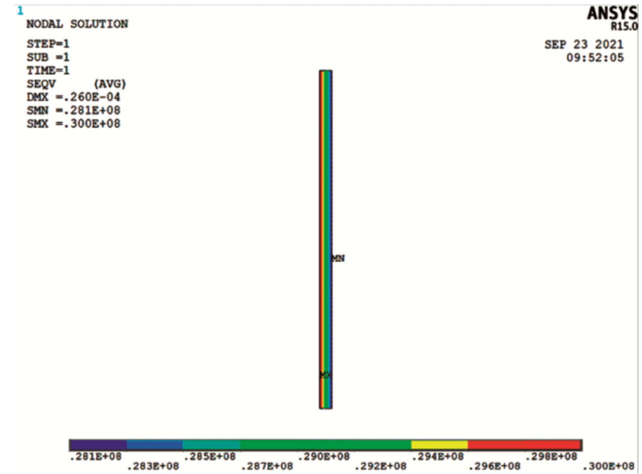

Fig. 7 — Von-mises stress for SS316L housing.

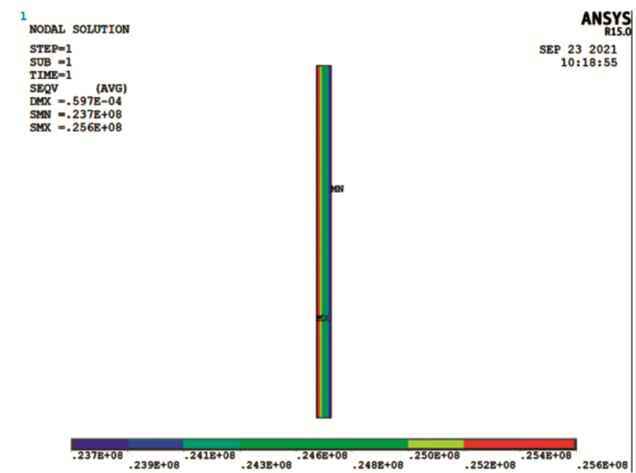

Fig. 8 — Von-mises Stress for Al-6061 housing.

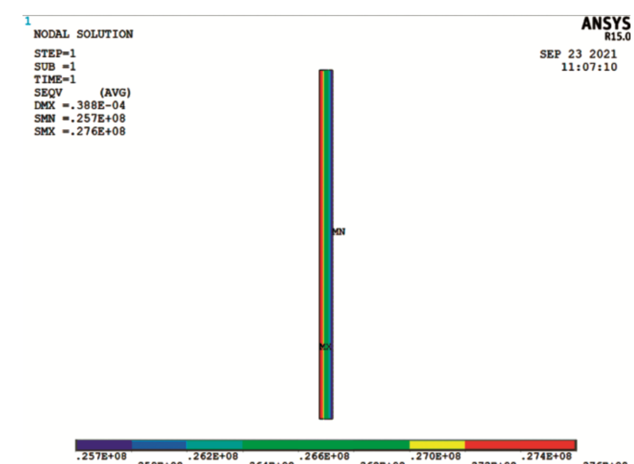

Fig. 9 — Von-mises Stress for Ti-6Al-4V Pressure housing.

2.4 Buckling Load calculation

When thin cylindrical pressure housing is subjected to compression and uniform lateral pressure, the cylinder will buckle even if it maintains a cylindrical shape. The simpler formula (Andrew PF Little) for elastic buckling of cylindrical pressure housing is as follows:

$$P_{cr} = \frac{2.42E\left(\frac{t}{2r}\right)^{\frac{5}{2}}}{\left(1-\gamma^2\right)^{0.75} \times \left[\left(\frac{l}{2r}\right) - 0.447\left(\frac{t}{2r}\right)^{\frac{1}{2}}\right]} \quad \text{... (1.19)}$$

(Where, P_{cr}= Critical Buckling pressure in MPa, E=Young's Modulus in GPa, t=Thickness of cylinder in m, r = radius of cylinder in m,ν=Poisson's ratio and l=length of cylindrical pressure housing).The critical buckling pressure of the pressure housing with various materials like SS316L, Aluminium 6061 and Titanium alloy are calculated and listed in the Table 4. The calculated critical pressure at which the cylindrical pressure housing may buckle is much above the design pressure of the cylindrical pressure housing of the various materials, the assumed wall thickness of the pressure housing as per ASME code and thin cylinder theory is satisfactory and the design is safe.

2.5 Corrosion studies

The corrosion due to pitting, crevice and stress are the major corrosion types that affect the marine materials. Stainless steel 316L is a well-recognised and prominent material for ocean applications because of its corrosion-resistant properties for any type of corrosion. Aluminium 6061 is normally corrosive and susceptible to intragranular attack. Mostly sacrificial zinc anode and proper insulators are provided in the pressure housing to avoid the pitting and crevice type of corrosion. Titanium alloys are superior for anti-corrosion and minimum level of erosion and corrosion. It is generally suited for deep water rather than shallow water applications.

3 Results and Discussion

Cylindrical pressure housings using the materials SS316L, Al6061-T6 and Ti-6Al-4V were designed using the ASME code and thin cylinder theory. Finite element modelling of pressure housing was performed using ANSYS software. The maximum stress values of the materials as given in the thin cylinder theory and standard material strength values of the ASME code are used to calculate the factor of safety (FOS). The values are tabulated in Table 5 and it is found that the maximum stress is at the hoop direction of the pressure housing. The pressure housing with SS316L material is found to be better with a FOS of 3.36 when compared with the maximum stress of other materials. Comparing all the three materials, SS316L material has higher buckling strength than the other two materials. The subsea housing thus designed was fabricated with SS316L and operated at a depth of 30 m in shallow waters of Goa for a period of six months continuously. The uniqueness of the SS316L pressure housing is its capability to accommodate higher sampling acoustic data acquisition electronics equipment and a larger capacity power pack for continuous measurements of ambient noise measurement mooring system for a longer period. In addition, the housing was designed to withstand extreme events like cyclones and storm surges in the

Table 5 — Comparison of Maximum stress (σ_h) Vs Factor of Safety (FOS)

Parameters	SS316L	Al 6061-T6	Ti-6Al-4V
Max.Stress(MPa)-thin cylinder theory	34.22	29.12	31.46
Max.Stress(MPa)-FEA	34.7	29.6	31.9
Max.Stress (MPa)-ASME code	115	69	75
Factor of Safety (FOS)	3.36	2.36	2.38
Von-mises Stresses (MPa)-Theoretical	29.79	25.21	27.24
Von-mises Stresses (MPa)-FEA	30.0	25.60	27.6

Fig. 10 — The retrieved Subsea pressure housing with SS316 L deployed off Goa.

shallow waters of Indian seas. The SPH operated off the coast of Vishakapatnam withstood the Jal cyclone and the one operated off the Chennai coast withstood the Vardah cyclone and recorded data during the extreme events, which were very helpful for rain noise analysis and noise characterisation.

4 Conclusion

The subsea pressure housings with the materials SS316L, Al 6061-T6 and Ti-6Al-4V were designed for open ocean operations up to a depth of 100 m depth using ASME code and thin cylinder theory. The housings were modelled and analysed using FEA. The results closely match with 5% variation. The Vonmises stresses for the yield criterion for both theoretical calculations and analysis agree with each other. The critical buckling pressure is higher for SS316L compared to the design pressure for all the two materials. In addition, the formability is much easier for SS316L compared with other materials and cost of the pressure building housing with SS316 L material is three to five times lesser than the other two materials. Thus SS316L is proved to be ideal for shallow-water applications and the SPH thus designed and operated off the Indian coast were functional over 6 months period providing crucial data during the cyclones.

Acknowledgments

We are thankful to Director, NIOT for encouraging us to carry out this work. We gratefully acknowledge the funding support from Ministry of Earth Sciences for carrying out this work. The authors pay their tribute to Late Professor Dr. Velamurali, Anna University, Chennai for mentoring this work initially.We immensely thank Ocean Acoustics team of NIOT for supporting at various stages in this work.

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