Design and analysis of International Maritime Organization tank

By

Naimish Bhalani

14MMED01

DEPARTMENT OF MECHANICAL ENGINEERING
INSTITUTE OF TECHNOLOGY
NIRMA UNIVERSITY
AHMEDABAD-382481
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Design and analysis of International Maritime Organization tank

Major Project Report

Submitted in partial fulfillment of the requirements

For the Degree of

Master of Technology in Mechanical Engineering (Design Engineering)

By

Naimish Bhalani

(14MMED01)

Guided By

Prof. V. M. Bhojawala

Co-Guided by

Prof. Keyur Hirapara

NIRMA UNIVERSITY
INSTITUTE OF TECHNOLOGY
NAAC ACCREDITED 'A' GRADE

DEPARTMENT OF MECHANICAL ENGINEERING
INSTITUTE OF TECHNOLOGY
NIRMA UNIVERSITY
AHMEDABAD-382 481
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1. The thesis comprises of my original work towards the degree of Master of Technology in Mechanical Engineering (Design Engineering) at Nirma University and has not been submitted elsewhere for a degree.

2. Due acknowledgment has been made in the text to all other material used.

Naimish Bhalani
14MMED01

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This is certify that the Major Project Report entitled "Design and analysis of International Maritime Organization tank" submitted by Mr. Naimish Bhalani(14MMED01), towards the partially fulfillment of the requirements for the awards of Degree of Master of Technology in Mechanical Engineering (Design Engineering) of Institute of Technology, Nirma University, Ahmedabad is record of work carried out by him under our supervision and guidance. In our opinion, the submitted work has reached a level required for being accepted for examination. The result embodied in this major project, to the best of our knowledge, has not been submitted to any other University or Institute for award of any degree.

Prof. V. M. Bhojawala
Department of Mechanical Engineering,
Institute of Technology,
Nirma University,
Ahmedabad-382481.

Prof. Keyur Hirapara
Department of Mechanical Engineering
Institute of Technology,
Nirma University,
Ahmedabad-382481

Dr. R. N. Patel
Professor and Head,
Department of Mechanical Engineering,
Institute of Technology,
Nirma University,Ahmedabad-382481.

Dr. P N Tekwani
Director
Institute of Technology,
Nirma University,
Ahmedabad-382481

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Naimish Bhalani

14MMED01

Abstract

IMO tanks are widely used in many industries for storing and transporting hazardous and non-hazardous liquid material. It is important to take care of all the safety measures as failure of these tanks causes danger to human being. This is also because the failure of tanks and their accessories is not limited to the immediate danger to nearby human lives, but also to a large extent leads to serious consequences and very likely to long-term environmental damages. So, the design of the tank should be done properly considering all safety measures. Design of the tank is done using specified standards such as ASME, IMDG and also top of the tank is provided with manhole for purpose of cleaning, inspection and maintenance. The nozzle at the manhole is designed using ASME code and verified stress with finite element analysis.

The main objective of this project work is, To carry out design & analysis of International Maritime Organization (IMO) tank. The IMO standard is suitable for transporting most hazardous materials. Tank may affected by corrosion due to wet atmosphere it is required that the vessels needs to be constructed from steel which has good corrosion resistant. Frame which is used for supporting tank is design using ISO standard. the design and necessary structural analysis of frame is also carried out.

Contents

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Nomenclature

Variables	Physical Quantity	Unit
P	Design Pressure	MPa
P_o	External Pressure	MPa
D_i	Inside Diameter	mm
E	Joint Efficiency	-
t_s	Thickness of Shell	mm
L_{T-T}	WL to WL for inner vessel	mm
SF	Straight flange	mm
V_{Head}	Volume of Head	mm^3
V_{Shell}	Volume of Shell	mm^3
V_g	Total Volume	mm^3
E	Modulus of Elasticity of Material	MPa
L	Design Length	mm
A	Factor A	-
B	Factor B	-
P_a	Allowable external pressure	MPa
I_s	Required moment of inertia	mm^4
W	effective length	mm
$\bar{y}$	Center of gravity	mm
L_s	Length between stiffeners	mm
S_v	Allowable stress of vessel	MPa
t_r	Required wall thickness of vessel	mm
t_{rn}	Required wall thickness of nozzle	mm
D_p	Diameter of reinforcement	mm

Variables	Physical Quantity	Unit
W	Weight	kg
y	Deflection	mm
Z	Section modulus	m^3

Greek		
γ	Shell parameter	-
β	Attachment parameter	-
ρ	Density	kg/m^3
ρ_{fluid}	Density of Fluid	kg/m^3

Acronyms	
MAWP	Maximum allowable working pressure
ASME	American Society of Mechanical Engineers
IMDG	International Maritime Dangerous Goods Code
IMO	International Maritime Organization

Chapter 1

Introduction

Pressure Vessel is used for store and transmission of liquids, gases and vapors by road, rail and ships. Pressure vessels are used in a variety of applications in both private and industry sector. They used in these industrial sectors as compressed air receivers and domestic sectors as hot water storage tanks. Other examples of pressure vessels are re-compression chambers, diving cylinders, autoclaves, distillation towers, and many other vessels in petrochemical plants and oil refineries, mining operations, nuclear reactor vessels, space ship habitats and submarine, hydraulic reservoirs and pneumatic reservoirs under pressure, rail and road vehicle air brake reservoirs and storage vessels for cryogenic liquefied gases such as hydrogen, oxygen, chlorine, ammonia, butane, propane and LPG.

International Maritime Organization (IMO) Tank is thermally insulated portable tank for transportation of refrigerated liquefied gases. IMO tanks are design using IMO IMDG.1: International Maritime Dangerous Goods Code (Volume 1) and ASME Section VIII Div. I. Main components of IMO Tanks are as follows:

1.1 Components of Chemical Storage Vessel

The major components of IMO Tank are vessel, piping, insulation, frame, cladding, heating system valves and gauges. The schematic of storage vessel with some of its components is shown in Fig 1.1

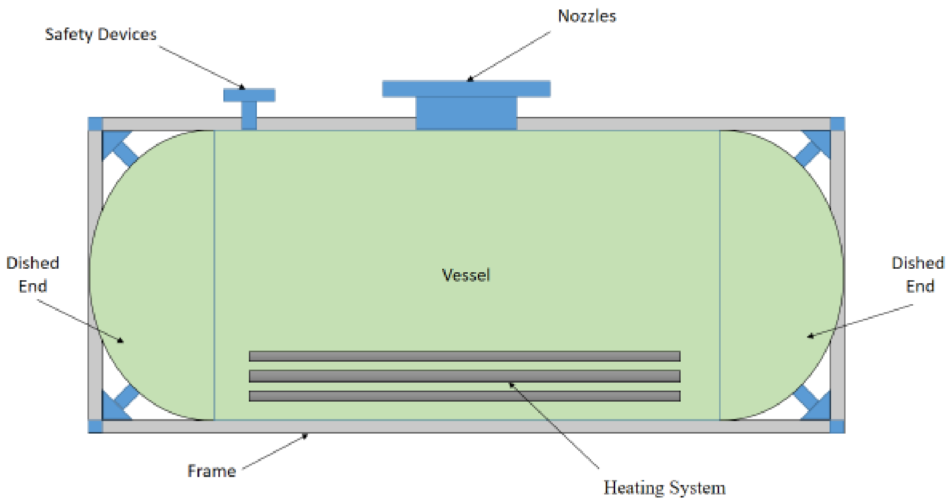

Figure 1.1: Components of Chemical Storage Vessel

Vessel

The vessel encloses the chemical fluid to be stored. It is also called the product container. The product container must be design according to pressure and temperature applied, the weight of the fluid within the vessel and the bending stress as a result of beam bending action. Also it is tube designed considering the external pressure applied on the vessel. The material selected for the construction of inner vessel must be compatible with the cryogenic fluid. The commonly used materials are stainless steel, 9% nickel steel and in some cases copper.

Piping

Piping is necessary to fill and remove liquid from the vessel and to vent vapor from the vessel. To minimize the heat in leak to the product container the length of the pipe should maximum and the thickness should be minimum. The minimum wall thickness of piping should be designed on the basis of internal pressure and availability.

Frames

The following frame structures are used for supporting IMO tank,

1. Full frame
2. Beam frame.

Full frame supports the tank by the side rails. Full frame also called Box type frame. Beam frame is support tank by top and bottom rails of frame. Weight of Beam frame is lower because of its construction, due to this type of construction load carrying capacity of the beam frame is less.

Cladding

The cladding jacket is used for protection against rain and sea water. The tank cladding is the outer most skin of the tank container, covering the insulation and tank shell. The commonly used materials for cladding are Glass reinforced plastic(GRP)/ Fiber reinforced polyester (FRP)stainless steel and aluminum. When it is to be designed for long term purpose common used material are GRP/FRP.

Insulation

In order to minimize the heat in leak from vessel at ambient temperature to the vessel at cryogenic temperature, thermal insulation is used. The effectiveness of a thermal insulation is judged on the basis of thermal conductivity. While selecting insulation it is necessary to keep the economical aspect of providing insulation in mind since return to investment has to be compared against saving in energy. Standard thickness is 50mm. The commonly used materials are Polyurethane foam (greenish), Rockwell, or Fiberglass.

Heating System

Heating system is required to prevent freezing of the tank sub-layer. The most common heating system used for IMO Tanks are steam heating and electrical heating. The steam channels is continuous loops of pipes. The location of pipes are lower half surface on the outside tank. The electrical heating system is consist of number of network elements. The location of electrical system is same as steam heating system.

Safety Devices

Safety devices are used to relieve the over pressure so that damage to the vessel can be avoided. The commonly used safety device for IMO Tank is Safety relief valve. Safety relief valve is used for protection against excessive over pressure of the tank.

Valves

Valves are used for controlling the flow and pressure of stored fluid. In IMO tank two type of valves are used. The valves are for filling and discharge of stored fluid.

1.2 Objective of project work

1. Design and analysis of IMO tank for 25 m^3 capacity and support structure for the vessel.

1.3 Thesis Organization

Ch. 2 include literature review which is carried out during project work.

Ch. 3 covers the design of vessel for internal pressure and design of varies attachment to the vessel.

Ch. 4 covers the design of vessel against external pressure.

Ch. 5 covers the design of structural frame.

Ch. 6 Summery and future work for the project.

Chapter 2

Literature review

Moss R. Dennis [1] is an important book for designing components of pressure vessel, vessel support and nozzle. This book is also used for determination of stress induced in vessel, nozzle analysis and for determination of local stress. The formulas given in this book are based on ASME Code, ASCE 7-95 and Uniform Building Code.

Henry H Bednar[2] has discussed the membrane stress analysis for vessels, discontinuity of stress, and local stress analysis for attachments to the pressure vessel. The book explains the procedure of design considering theoretical calculations and also explain fundamentals concept. It also provides guideline for the selection of materials.

Eugene F. Megyesy [4] has explained the design of pressure vessel and construction methods. This book provides simple formulas, technical data and construction methods for pressure vessels. The design of this book is based on ASME Code for Pressure Vessel, Sec. VIII Div. I. This book also helps in identifying construction methods, loading and materials.

P. P. Bijlard, I. C. Wang & R. J. Dohrmann[13] has worked on junction of nozzle to vessel for same diameters. This paper provides methods for design of pipe-lines, header nozzle junction, Nuclear reactors and process reactors etc. The derivation are based on shell theory considering vessel as thick cylinder. The solution of partial differnetial equations of shell is obtained by using series of solution and satifying continuity conditions at junction of the shells.

C. Nadarajah, A. S. Tooth and J. Spence [14] has worked on design of local stress for attachment on vessel. To identified limitation of deflection and rotation by using displacement analysis method for neglecting the rigidity of attachment. By changing number of attachment pad thickness to determine effect of the attachment rigidity.

C.J. Dekker and J. Cuperus[15] has given accurate formula for determining the stress induced considering various location and size of the reinforcing pad . From large number of linear elastic Finite element analysis non-dimensional parameter such as shell parameter and attachment is developed. In addition applicable stress limit is proposed for local stress at vessel reinforcing location.

C. J. Dekker and H. J. Bos[16] has worked done on comparison of local load stress calculation method for different condition and also investigation of finite element analysis for nozzles wall. The use of shrink ring method and devise multiplication charts for deriving local load nozzle stresses. By use of local load stress result to evaluate pressure induced stresses at nozzles.

J. Fang, Z.F. Sang and Q.H. Tang[18] has studied strength behavior for cylindrical shell intersections with and without reinforcement pad under plane moment loading on nozzle. To performed three dimensional non-linear finite element numerical analysis, load displacement and local strain test for investigate effect of reinforcement pad. The results of test indicate decreasing maximum elastic stress and increasing plastic limit load. The use of test data for understanding reinforcement pad and development of design method by limit analysis for pad reinforced cylindrical vessels under external loads applied on nozzle.

Georgios E. Mageirou and Charis J. Gantes [19] has done a work to simplified approach to evaluation of the critical buckling load of multi story frames with semi rigid connections. so that, respective graphs describing for the boundary conditions expression are suggested for the calculation of the effective buckling length coefficient for different levels of frame sway ability. And also the derivation of rotational stiffness co-efficient is done, which consequently used for the replacements of member converging at the bottom and top ends of the column in question by equivalent spring.

C. de Paor, D. Kelliher, K. Cronin, W.M.D. Wright, S.G.Mc Sweeney [20] has done research about uniform external pressure applied on thin cylindrical shells and effect of buckling capacity by geometric imperfection. By using FE analysis and geometrically non linear static analysis has been undertaken to investigate the collapse pressure and post buckling mode shape.

Chapter 3

Design of Vessel using ASME and IMDG Code

3.1 Selection Material

The criteria for the selection of material for vessel are as follows:

1. Strength of material
2. Fabricability
3. Corrosion Resistance
4. Fracture Toughness
5. Cost, availability and ease of maintenance.

Table 3.1: Material selection guide [2]

Design Temperature, °F		Material	Plate	Pipe	Forgings	Fittings	Bolting
Cryogenic	−425 to −321	Stainless steel	SA-240-304, 304L, 347, 316, 316L	SA-312-304, 304L, 347, 316, 316L	SA-182-304, 304L, 347, 316, 316L	SA-403-304, 304L, 347, 316, 316L	SA-320-B8 with SA-194-8
	−320 to −151	9 nickel	SA-353	SA-333-8	SA-522-1	SA-420-WPL8	
Low temperature	−150 to −76	3½ nickel	SA-203-D	SA-333-3	SA-350-LF3	SA-420-WPL3	SA-320-L7 with SA-194-4
	−75 to −51	2½ nickel	SA-203-A				
	−50 to −21	Carbon steel	SA-516-55, 60 to SA-20	SA-333-6	SA-350-LF2	SA-420-WPL6	
	−20 to 4		SA-516-All	SA-333-1 or 6			SA-193-B7 with SA-194-2H
	5 to 32		SA-285-C	SA-53-B SA-106-B	SA-105 SA-181-60,70	SA-234-WPB	
Intermediate	33 to 60 61 to 775		SA-516-All SA-515-All SA-455-II				
Elevated temperature	776 to 875	C-½Mo	SA-204-B	SA-335-P1	SA-182-F1	SA-234-WP1	
	876 to 1000	1Cr-½Mo	SA-387-12-1	SA-335-P12	SA-182-F12	SA-234-WP12	
		1-¼Cr-½Mo	SA-387-11-2	SA-335-P11	SA-182-F11	SA-234-WP11	
	1001 to 1100	2¼Cr-1Mo	SA-387-22-1	SA-335-P22	SA-182-F22	SA-234-WP22	with SA-193-B5 SA-194-3
	1101 to 1500	Stainless steel	SA-240-347H	SA-312-347H	SA-182-347H	SA-403-347H	SA-193-B8 with SA-194-B
		Incoloy	SB-424	SB-423	SB-425	SB-366	
	Above 1500	Inconel	SB-443	SB-444	SB-446	SB-366	

The material selected for various component shell, head and nozzle are stainless steel SA-240 Type 304, 304L, 347, 316 and 316L for cryogenic fluid as shown in Table 3.1.

TYPE		CHEMICAL COMPOSITION				MECH PROPERTIES			REMARKS
		Cr	Ni	C (Max)	Other (Max)	Tensile KSI	Yield KSI	Elong (%)	
AUSTENITIC	201	16-18	3.5-5.5	0.15	Mn 5.5-7.5 N .25	115	55	55	Low Ni Version of 301
	202	17-19	4-6	0.15	Mn 7.5-10 N .25	105	55	55	Low Ni Version of 302
	203EZ	16-18	5-6	0.07	Mn 5.5-6.5 Cu 1.75-2.15 S.18-.35	90	40	50	Free machining grade
	216	19.75	6	0.08	Mn 8.25 Mo 2.5 N.37	100	55	45	Similar to 316 w/ better high strength properties
	301	16-18	6-8	0.15	Mn 2 Si 1	110	40	60	High work hardening, structural grade
	302	17-19	8-10	0.15	Mn 2 Si 1	90	40	55	General purpose SST
	303	17-19	8-10	0.15	Mn 2 Si 1 P .2 Mo .6 S .15 Min	90	35	50	Free machining version of 302
	303SE	17-19	8-10	0.15	Mn 2 Si 1 Se .15 Min P .2 S .06	90	35	50	Better surface finish than 303
	303PB	17-19	8-10	0.15	Mn 2 Si 1 Mo .6 Pb .12-.3	90	35	50	Leaded version of 303 for high volume machining
	304	18-20	8-10	0.08	Mn 2 Si 1	85	35	55	Low carbon variation of 302
	304L	18-20	8-10	0.03	Mn 2 Si 1	80	30	55	Extra low carbon 304
	305	17-19	10-13	0.12	Mn 2 Si 1	85	35	55	Low work hardening. Good spinning and deep drawing
	309	22-24	12-15	0.2	Mn 2 Si 1	95	40	45	High temperature applications
	310	24-26	19-22	0.25	Mn 2 Si 1.5	95	45	50	Excel corrosion resistance
	316	16-18	10-14	0.08	Mn 2 Si 1 Mo 2-3	85	35	60	Best corrosion resistance of standard SST's. High temperature strength
	316L	16-18	10-14	0.03	Mn 2 Si 1Mo 2-3	78	30	55	Extra low carbon 316
	317	18-20	11-15	0.08	Mn 2 Si 1Mo 3-4	90	40	50	316 w/ better creep resistance
	321	17-19	9-12	0.08	Mn 2 Si 1 P.04 S .03 Ti 5 x C Min	85	35	55	Stabilized w/ Ti
	347	17-19	9-12	0.08	Mn 2 Si 1Cb-Ta 10 x C Min	95	40	50	Stabilized w/ Cb

Figure 3.1: Material property[1]

The material selected for various component shell, head and nozzle for is SA-240 Type 316L because of corrosion resistance of the material is high. The material properties for the same are given in Table 3.2.

Table 3.2: Material property of SA-240 type 316L

Property	Value	Unit
Tensile Yield Strength	170	MPa
Ultimate Tensile Strength	485	MPa
Young's Modulus	190000	MPa
Poisson's Ratio	0.22	-
Allowable Stress at -40 $°C$ to 150$°C$ temperature	115	MPa
Bulk Modulus	158000	MPa
Shear Modulus	73000	MPa
Density	8027	kg/m^3

3.2 Design Input

The IMO tank which is to be designed has following specification:

Maximum allowable working pressure (MAWP)		0.4	MPa
Corrosion allowance for shell		0.2	mm
Corrosion allowance for dish end		0.1	mm
Design Temperature	Maximum	150	$^{\circ}C$
	Minimum	-40	$^{\circ}C$
Specific gravity of stored fluid (ρ_{fluid})		2297	kg/m^3
Radiography		full	-
Joint Efficiency E		1	-
Dimension of IMO Tank	Length	6058	mm
	Width	2438	mm
	Height	2591	mm

3.3 Inside Diameter of Shell

According to the dimension of IMO tank, width of the tank is 2438 mm. For insulation purpose internal diameter of vessel is 2378 mm.

3.4 Internal Design Pressure

3.4.1 Design Loading

Design loading as per UG-26 include the following

1. Internal Pressure.
2. Weight of pressure vessel and test conditions.

3. Maximum static head (SH) of contained fluid under normal operating conditions.

4. Loads at attachment of components.

3.4.2 Design Pressure

The design pressure as per UG-21 define as difference between internal pressure and external pressure, and addition of static head (SH) at bottom of vessel due to stored fluid.

Assume the max height for static head is internal diameter of vessel.

$$Static\ Head = \rho_{fluid} \times g \times H$$

For Calculation Static Head = $0.0535 MPa$.

According to IMDG code, external pressure on tank is 0.041 MPa

$$Design\ Pressure = MAWP - External\ Pressure + Static\ Head$$

For Calculation Design Pressure= 0.4126 MPa.

3.5 Thickness of Cylindrical Vessel

The minimum thickness required by cylindrical vessel will be the maximum value calculated by using as per UG-27 , UG-16 and IMDG code.

By considering Longitudinal Stress, Circumferential Stress and thickness arrived by calculation adding factor to the C.A.

Circumferential Stress

When the cylindrical vessel under internal pressure then thickness of shell not more than one-forth of inside diameter ($t < Di/4$) and design pressure (P) is not more than $0.385SE$ ($P < 0.385SE$)

$P = 0.385SE = 44.275 > P$

$$t = \frac{P \times R}{S \times E - 0.6 \times P} + C.A. = 4.39\ mm$$

Longitudinal Stress

When the cylindrical vessel under internal pressure then thickness of shell not more than one-forth of inside diameter ($t < Di/4$) and design pressure (P) is not more than $1.25SE$ ($P < 1.25SE$)

$P = 1.25SE = 143.75 > P$

$$t = \frac{P \times R}{(2 \times S \times E + 0.4 \times P)} + C.A. = 2.29\ mm$$

Minimum thickness as per UG-16.

$$t = 1.5 + C.A. = 1.7\ mm$$

Minimum thickness According IMDG Code.

The diameter of the tank is more than 1800 mm the required thickness is

$$t_s = 6\ mm$$

$$Minimum\, thickness = maximum(4.35,\ 2.27,\ 1.7, 6)\ mm$$

Minimum required thickness for shell= 6 mm.

3.6 Thickness calculation for Dish end

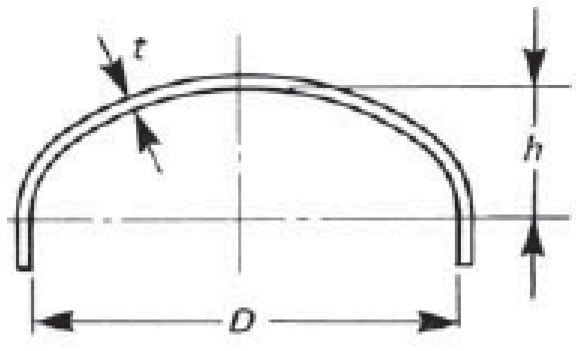

Figure 3.2: Ellipsoidal head[11]

Ellipsoidal head also called 2:1 elliptical head. The height of Ellipsoidal head is quarter of the diameter of head. Ellipsoidal heads radius varies from major axis to minor axis.

As per UG-32, required minimum thickness of 2:1 Semi Ellipsoidal head evaluating by following formula,

$$t = \frac{P \times D}{2 \times S \times E - 0.2 \times P} + C.A. = 4.61\ mm$$

Minimum thickness required for Ellipsoidal head= 6 mm.

3.7 Capacity Calculations for Vessel

Input data:

1. Inside diameter of vessel = 2378 mm
2. Tangent to tangent distance for inner vessel (Ls)= 4836 mm

3. Straight flange(SF)= 18 mm

4. Density of stored fluid = 2297 kg/m^3

Volume of the head,

$$V_{Head} = \frac{\pi \times {D_i}^3}{24} = 1.76\ m^3$$

Volume of the shell,

$$V_{Shell} = \frac{\pi \times {D_i}^2 \times (Ls + 2 \times SF)}{4} = 21.48\ m^3$$

Total volume of vessel,

$$V_g = 2 \times V_{Head} + V_{Shell} = 25\ m^3$$

Volume capacity of the tank is $25m^3$

3.8 Bending

Assumptions

Cylindrical vessel is consider as beam with uniformly distributed load applied. Also assumes the model is fixed at both side.

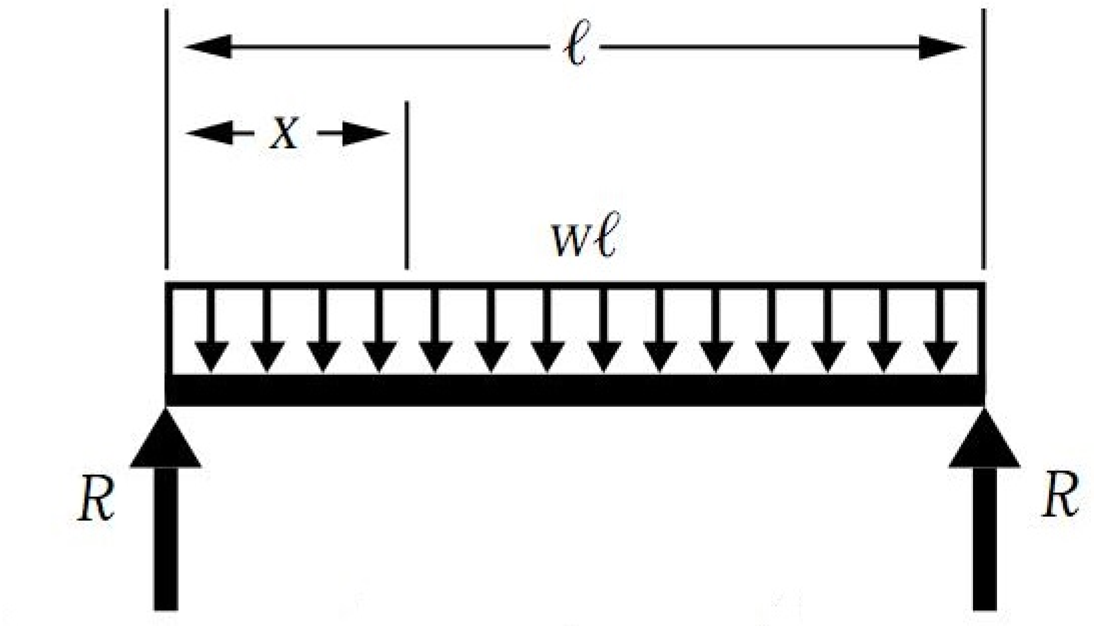

Figure 3.3: Fix supported beam diagram[12]

The Fix supported uniformly distribution load as show in Figure 3.3,

Weight of liquid (w) = 32000 kg

Length of vessel (l) = 4.836 m

Cross Section of Beam = hollow cylinder

Inside diameter of cylinder (D_i) = 2378 mm

Thickness of cylinder (t) = 6 mm

Yield strength of cylinder (S_y) = 170 MPa

Total load applied on beam (W) = $32000 * g = 320\ kN$

Reaction on Support $(R) = W/2 = 160\ kN$

Bending moment at center of beam $(M) = R \times l/2 = 160 \times 4.836/2 = 386.88\ kNm$

Moment of inertia of hollow cylinder

$$I = \frac{\pi}{64}(D_o^4 - D_i^4) = \pi/64((2378 + 2 \times 6)^4 - (2378)^4) = 3.190 \times 10^{10}\ mm^4$$

Longitudinal stress,

$$\sigma_l = \frac{Pd}{4t} = \frac{0.4126 * 2378}{4 * 6} = 40.88\ MPa$$

Bending moment stress,

$$\sigma_m = \frac{My}{I} = \frac{386880 \times 10^3 \times 1189}{3.190 \times 10^{10}} = 14.42\ MPa$$

Total bending stress,

$$\sigma_b = \sigma_l + \sigma_m = 40.88 + 14.42 = 55.30\ MPa < \sigma_y$$

3.9 Nozzle Calculation

In many industrial applications nozzle connections is importance for vessel. Other applications for nozzle connection is pipelines, process reactors and header nozzle junctions. In such cases, calculating stresses at junction of a nozzle to be most important.Locations of nozzles on vessel show in Figure 3.4

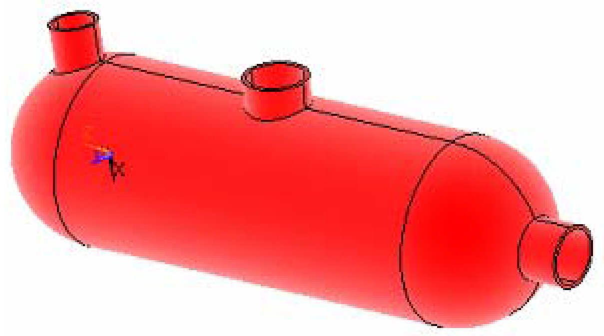

Figure 3.4: Locations of Nozzles on Vessel

In regular practice design of nozzle connection is based on stress analysis methods given by ASME Code. The requirement of acceptability of design is that design should be such that stresses shall not exceed the limit given in the code.

3.10 Design Input for Nozzle

The Nozzle which is to be designed has following specification:

Inside diameter of vessel (D_i)	2378	mm
Thickness of vessel (t)	6	mm
Allowable stress of vessel (S_v)	115	MPa
Joint Efficiency (E)	1	
Allowable stress of nozzle (S_n)	115	MPa
Corrosion allowance (C.A)	0.2	mm
Design Pressure (P)	0.41258	MPa
Size of welding Leg (leg)	6	mm
Distance of nozzle under the inner surface of the vessel	0	mm

3.11 Reinforcement Calculation

3.11.1 Reinforcement Required for Openings in Shell

As per UG-37,

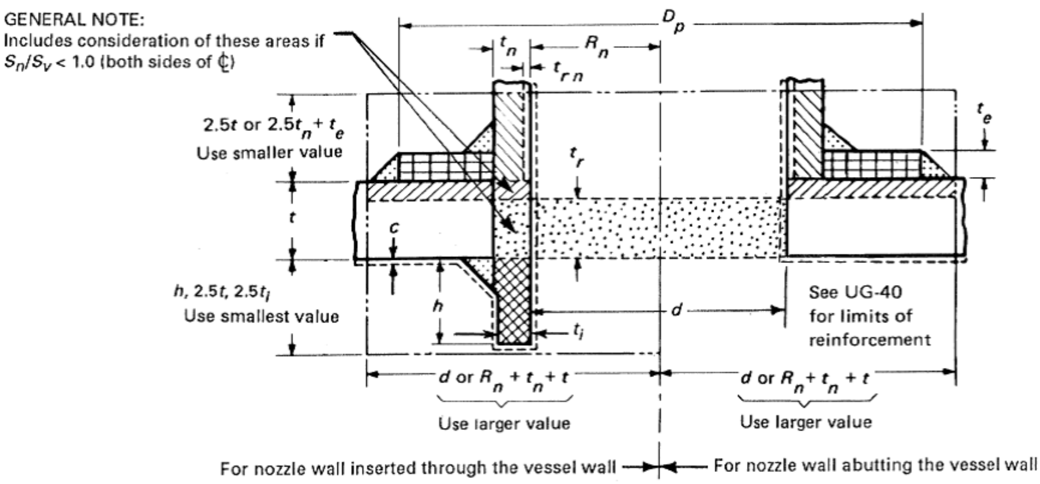

Figure 3.5: Reinforced Openings[11]

Required wall thickness of vessel

$$t_r = \frac{PR}{(SE - 0.6P)} = 4.27\ mm$$

Opening diameter of nozzle = 500 mm

Required wall thickness of nozzle

$$t_{rn} = \frac{PR_n}{SE - 0.6P} = 0.88\ mm$$

Calculate the required reinforcement area, A

$$A = dt_rF + 2t_nt_rF(1 - f_{r1})$$

Here, $F = 1, f_{r1} = f_{r2} = f_{r3} = f_{r4} = S_v/S_p = 1$ because the material of Nozzle, Shell and Reinforcement element are same.

Assume thickness of nozzle = 6 mm

$$d = Diameter\ of\ opening - 2(wall\ thickness) + 2(C.A.) = 488.4\ mm$$

$$t_n = Thickness\,of\,nozzle - C.A. = 5.8\ mm$$

$$t = Thickness\,of\,vessel - C.A. = 5.8\ mm$$

Required area of reinforcement $A = 2087.90\ mm^2$

Calculate the reinforcement area that is available in vessel shell, A_1

$$A_1 = d * (E_1t - Ft_r) - 2t_n * (E_1t - Ft_r)(1 - f_{r1}) = 744.82\ mm^2$$

$$A_1 = 2 * (t + t_n)(E_1 t - F t_r) - 2t_n(E_1 t - F t_r)(1 - f_{r1}) = 35.38\ mm^2$$

$$A_1 = maximum(744.82\ mm^2, 35.38\ mm^2)$$

Available reinforcement area in vessel shell $A_1 = 744.82\ mm^2$

Calculation the reinforcement area that is available in nozzle wall, A_2

$$A_2 = 5(t_n - t_{rn}) f_{r2} t = 142.76\ mm^2$$

$$A_2 = 5(t_n - t_{rn}) f_{r2} t_n = 142.76\ mm^2$$

$$A_2 = minimum(142.76\ mm^2, 142.76\ mm^2)$$

Available reinforcement area in nozzle wall $A_2 = 142.76\ mm^2$

Calculation the reinforcement area that is available in nozzle extends the inside vessel wall, A_3

$$A_3 = 5 t t_j f_{r2} = 0\ mm^2$$

$$A_3 = 5 t_j t_j f_{r2} = 0\ mm^2$$

$$A_3 = 2ht_j f_{r2} = 0\ mm^2$$

$$A_3 = minimum(0\ mm^2, 0\ mm^2)$$

Available reinforcement area in nozzle extends the inside vessel wall $A_3 = 0\ mm^2$

Calculation the reinforcement area that is available in various welds

$$A_{41} = (leg)^2 f_{r2} = 36\ mm^2$$

$$A_{43} = (leg)^2 f_{r2} = 0\ mm^2$$

Total Available reinforcement area without reinforcing element,

$$A_1 + A_2 + A_3 + A_{41} + A_{43} = 923.58\ mm^2$$

Here, Total available reinforcement area without reinforcing element is less than the required area of reinforcement. Reinforcing element are required for nozzle.

3.11.2 Reinforcement Calculation of Reinforcing Element

Assume diameter of reinforcing element is $(D_p) = 612$ mm and thickness of reinforcing element is $(t_e) = 10\ mm$.

Required area of reinforcement $A = 2087.90\ mm^2$

Available reinforcement area in vessel shell $A_1 = 744.82\ mm^2$

Calculation the reinforcement area that is available in nozzle wall, A_2

$$A_2 = 5(t_n - t_{rn})f_{r2}t = 142.76\ mm^2$$

$$A_2 = 2(t_n - t_{rn})(2.5t_n + t_e)f_{r2} = 201.83\ mm^2$$

$$A_2 = minimum(142.76\ mm^2, 201.83\ mm^2)$$

Available reinforcement area in nozzle wall $A_2 = 142.76\ mm^2$

Available reinforcement area in nozzle extends the inside vessel wall $A_3 = 0\ mm^2$

Calculation the reinforcement area that is available in various welds

$$A_{41} = (leg)^2 f_{r3} = 36\ mm^2$$

$$A_{42} = (leg)^2 f_{r4} = 0\ mm^2$$

$$A_{43} = (leg)^2 f_{r2} = 60\ mm^2$$

Calculation the reinforcement area that is available in Cross section area of material added as reinforcement A_5

$$A_5 = (D_p - d - 2t_n)t_e f_{r4} = 1120 \; mm^2$$

Total Available reinforcement area with reinforcing element,

$$A_1 + A_2 + A_3 + A_{41} + A_{42} + A_{43} + A_5 = 2103.58 \; mm^2$$

Here, Total available reinforcement area with reinforcing element is greater than the required area of reinforcement.

3.12 Load to be carried by welds

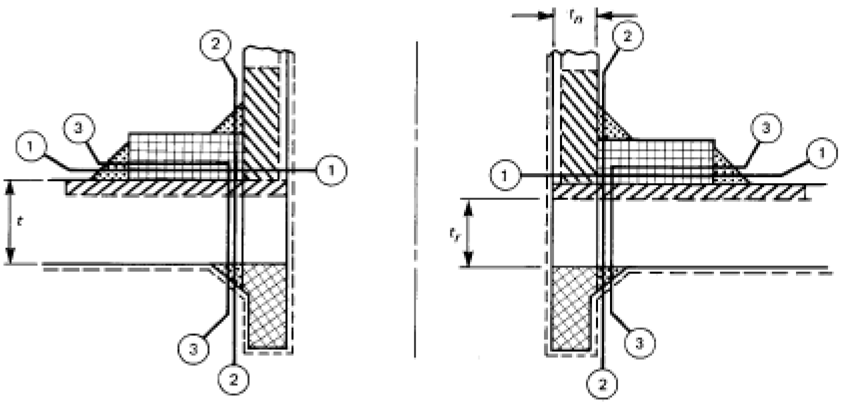

Figure 3.6: Nozzle Attachment Weld Strength Paths and Weld Loads[11]

Calculation of total weld load by using UG 41, W

$$W = [A - A_1 + 2t_n f_{r1}(E_1 t - F t_r)]S_v = 156488 \; N$$

Weld load for path 1-1, W_{1-1}

$$W_{(1-1)} = (A_5 + A_2 + A_{41} + A_{42})S_v = 149357\ N$$

Weld load for path 2-2, W_{2-2}

$$W_{(2-2)} = (A_2 + A_3 + A_{41} + A_{43} + 2t_n t f_{r1})S_v = 28294\ N$$

Weld load for path 3-3, W_{3-3}

$$W_{(3-3)} = (A_2 + A_3 + A_5 + A_{41} + A_{42} + A_{43} + 2t_n t f_{r1})S_v = 163994\ N$$

3.13 Strength of connection elements

From Figure 3.6,

Unit stress of inner fillet weld shear $= 0.49 \times S_v = 56.35\ MPa$

$$Inner\ fillet\ weld\ shear = \frac{\pi}{2} \times nozzle\ O.D. \times Leg \times Unit\ stress\ of\ Inner = 271704\ N$$

Unit stress of outer fillet Weld Shear$= 0.49 \times S_v = 56.35\ MPa$

$$Outer\ fillet\ weld\ shear = \frac{\pi}{2} \times Reinforcing\ Element\ O.D. \times Leg \times Units\ tress\ of\ outer$$

$$= 325025\ N$$

Unit stress of Groove weld tension $= 0.74 \times S_v = 85.10\ MPa$

$$Groove\ weld\ tension = \frac{\pi}{2} \times nozzle\ O.D. \times t \times Unit\ stress\ of\ Groove = 396650\ N$$

Unit stress of Nozzle wall shear= $0.7 \times S_v = 80.50\ MPa$

$$Nozzle\ wall\ shear = \frac{\pi}{2} \times mean\ nozzle\ dia \times t_n \times Unit\ stress\ of\ Nozzle = 383747\ N$$

Strength Paths 1-1 = Inner fillet weld shear + Nozzle wall shear = 655451 N

Strength Paths 2-2 = Inner fillet weld shear + Groove weld tension = 668354 N

Strength Paths 1-1 = Outer fillet weld shear + Groove weld tension = 721675 N

Here, All paths are stronger the strength of W required.

3.14 Total Stress at Nozzle

The calculation of the stresses is divided into four major component of applied loading. The fore major loading are radial force, bending moment, torsional moment and shear force. They are combined to become the maximum stress.

As per WRC (Welding Research Council)

- Radial Load (P_r) = 100 N
- Circumferential Moment (M_c) = 0 Nm
- Longitudinal Moment (M_L) = 0 Nm
- Torsion Moment (T_r) = 0 Nm

Geometric Parameter

Shell parameter

$$\gamma = \frac{R_m}{t + t_p} = 100$$

Attachment parameter

$$\beta = \frac{0.875 * r_o}{R_m} = 0.366$$

From Figure 3.7 Compression Stress factor $C_p^{'} = 0.89$

The maximum tensile stress due to Radial Load is given by,

$$\sigma_t = C_p^{'}(\frac{P}{t^2})$$

The maximum tensile stress due to Radial Load $\sigma_t = 2.32\ N/mm^2$

The maximum tensile stress due to Design Pressure is given by,

$$\sigma_t = \frac{PDi}{2t}$$

The maximum tensile stress due to Design Pressure $\sigma_t = 79.91\ N/mm^2$

Theoretical Total Stress$= 2.32 + 79.91 = 82.23\ N/mm^2$

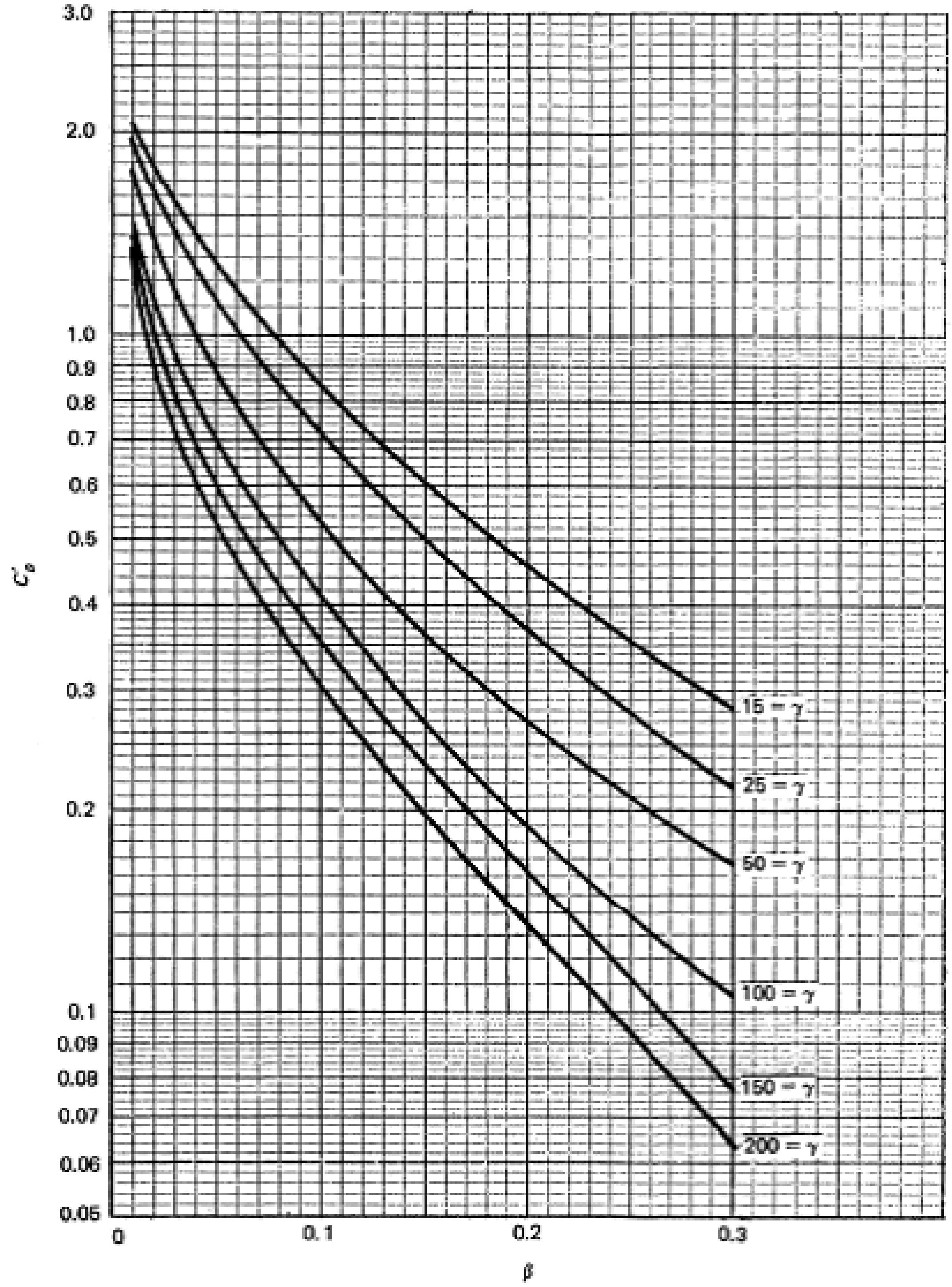

Figure 3.7: Compression Stress factor C_p' [2]

3.15 Finite Element Method

3D modeling

The 3D modeling of vessel is prepared by using Autodesk Inventor as shown in Figure 3.8 and basic dimension selected by the calculation is shown in table 3.3

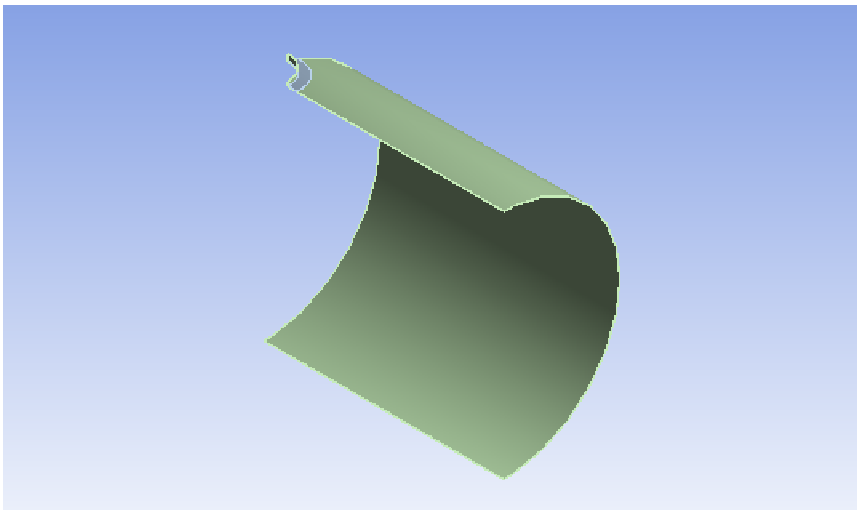

Figure 3.8: 3D model of Vessel

Mesh by Workbench

The meshing is created with tri element as shown in Figure 3.9

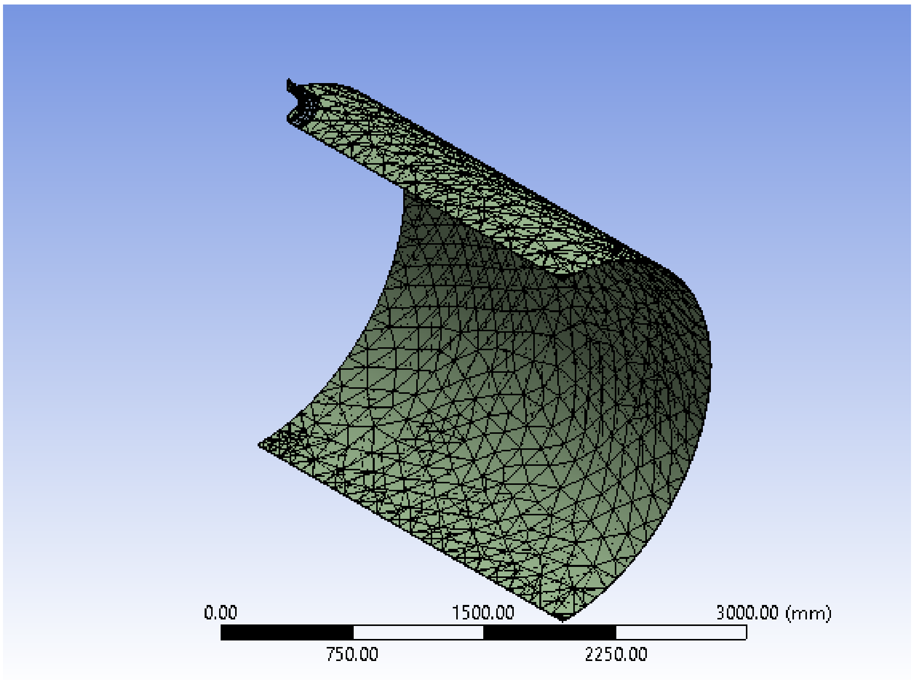

Figure 3.9: Mesh generation

Boundary Condition

The boundary condition is applied as show in Figure 3.10

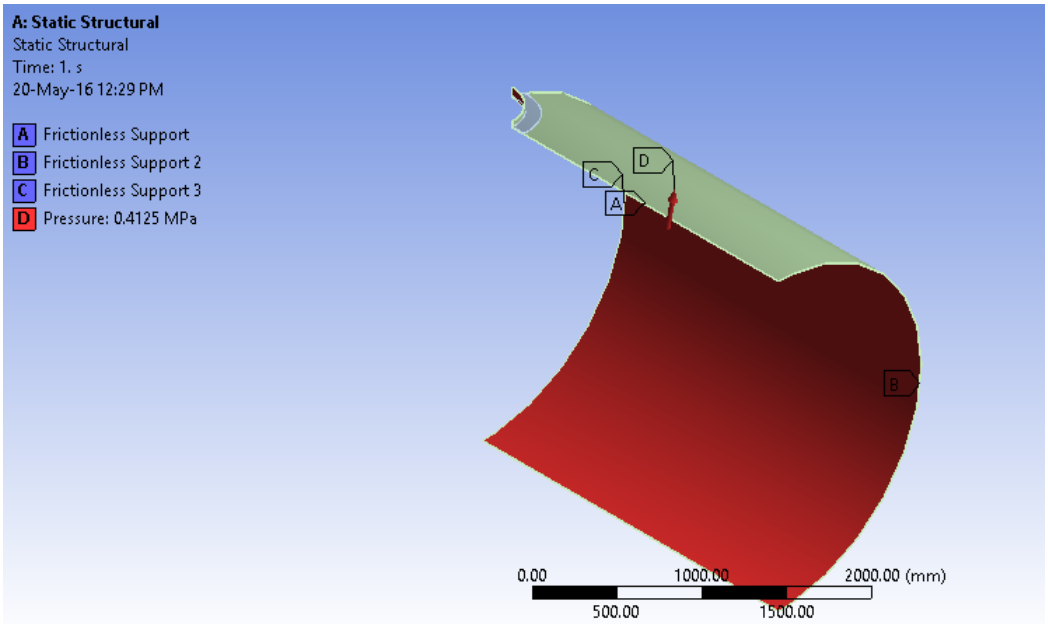

Figure 3.10: Boundary condition

Stress calculation from Workbench

The Equivalent (von-Mises) Stress at Nozzle as show in Figure 3.11. The average stress at nozzle is 140 MPa.

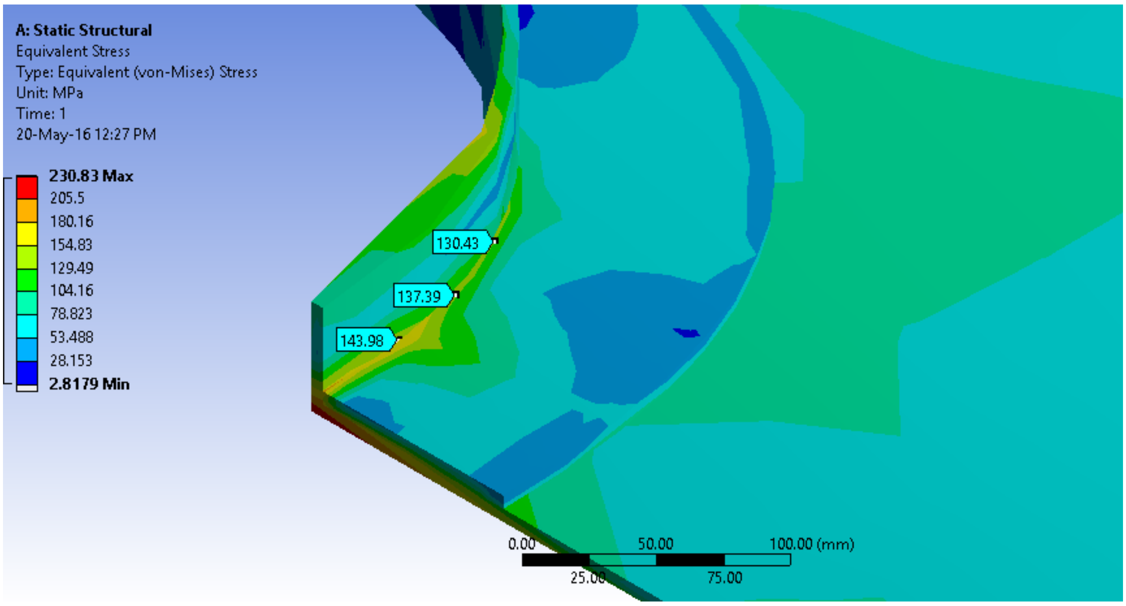

Figure 3.11: Equivalent (von-Mises) Stress

Chapter 4

Design for External Pressure

4.1 Introduction

Pressure vessel is subjected to compressive forces such as dead weight, earthquake, wind and atmospheric pressure. Pressure vessel component behave differently under this compressive forces when internal tensile forces (internal pressure) applied. This difference in behavior is due to elastic instability, which makes shell weaker in compression force than tensile force. Because failure of elastic instability, the vessel is said to buckle or collapse. Allowable stress is not used as design pressure vessels because of elastic instability. The design pressure is based on prevention of elastic instability under the applied external pressure. The allowable external pressure can be increased by joining stiffening rings around the circumferential direction of the vessel or increasing thickness of vessel.

4.2 Design Input for Vessel

1. Inside diameter of vessel (D_i) = 2378 mm

2. Design pressure (P_o) = 0.44 MPa [10]

3. Tangent to tangent distance of vessel $(L_{T-T}) = 4932\ mm$
4. Depth of head $(h) = 563\ mm$
5. Modulus of Elasticity of Material (E) = 190000 MPa
6. Thickness of shell $(t) = 6\ mm$

4.3 Thickness of Shells under External Pressure UG-28

Step 1 : Assume a thickness of vessel = 6 mm

Step 2 : Calculate dimension Design length L and Design Diameter

$$L = L_{(T-T)} + 2/3 \times h$$

$$Do = Di + 2 \times t$$

Design length L = 5300 mm

Design Diameter Do = 2390 mm

Step 3 : Evaluate ratios of L/D_o and D_o/t

Ratio of $L/D_o = 2.06$

Ratio of $Do/t = 398.33$

Step 4 : Determine Factor A from Figure 4.1 by using ratios of L/D_o and D_0/t.

$$Factor\ A = 0.00007889$$

Step 5 : From Figure 4.2 value of Factor A is left side of the material.

Factor B = 0

Step 6 : The allowable external pressure P_a are evaluate by following formula.

$$P_a = \frac{2AE}{(3(Do/t))}$$

The allowable external pressure $P_a = 0.0251$ MPa < 0.44 MPa

Here, design external pressure (P_o) grater than allowable external pressure (P_a), stiffening rings are required.

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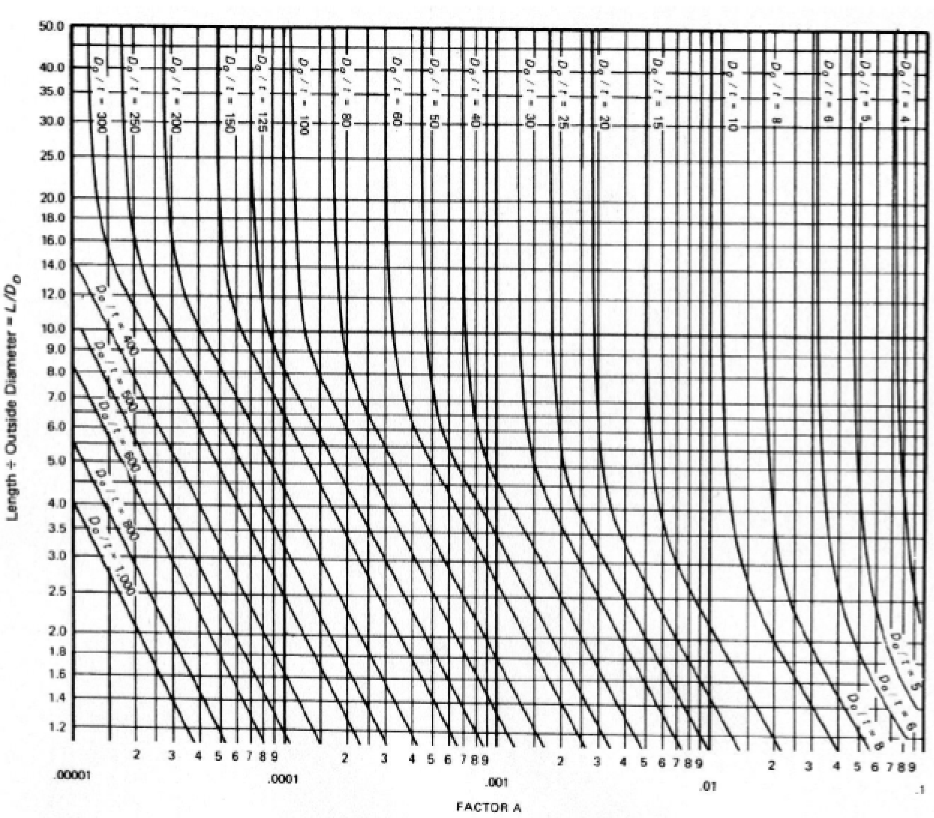

Figure 4.1: Geometric chart for components under external loadings[22]

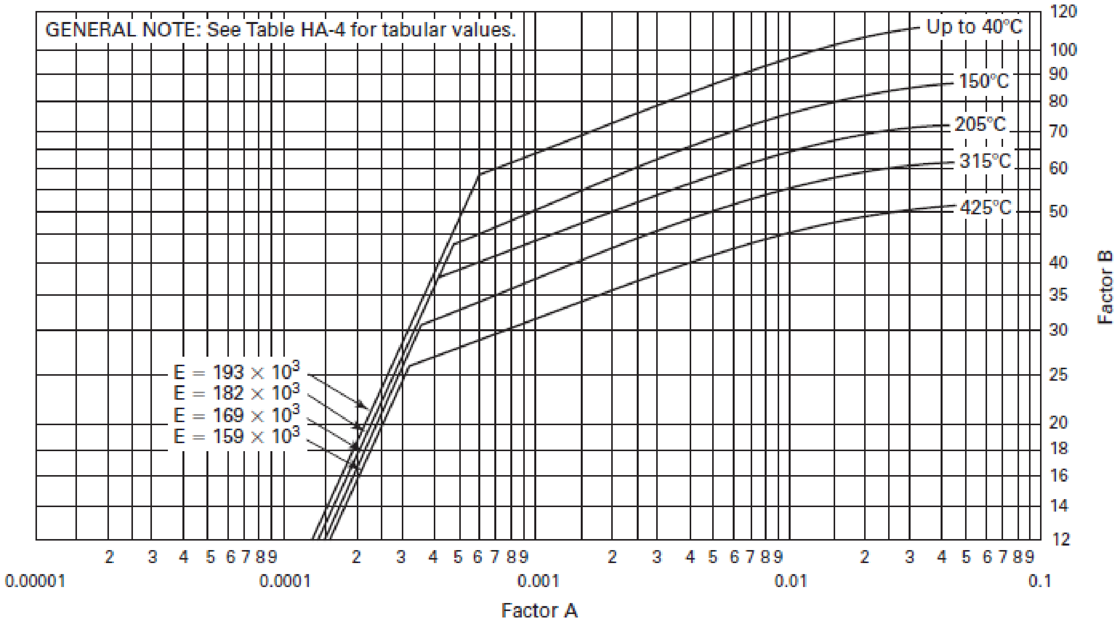

Figure 4.2: Chart for determining shell thickness of components under external pressure[22]

4.4 Design of Stiffening Ring

As per UG-29,

The minimum required moment of inertia (M.I.) is determined by following formula,

$$I_s = \frac{[{D_o}^2 L_s(t + A_s/L_s)A]}{14}$$

$$I_s^{'} = \frac{[{D_o}^2 L_s(t + A_s/L_s)A]}{10.9}$$

Where,

I_s= Required moment of inertia (M.I.) Figure 4.3 of given cross section of ring about it's natural axis,

$I_s^{'}$= Required moment of inertia (M.I.) of given cross section area of stiffener rings and effective shell about their combined natural axis.

The effective length of shell is given by following formula,

$$W = 1.10\sqrt{(D_o t_s)}$$

Calculation of Actual moment of inertia,

Assume cross section of stiffener ring is T-section as show in Figure 4.3

Size of stiffening ring (130 mm x 6 mm) (27 mm x 6 mm)

M.I. of the stiffening ring is $I_s = 36602\ mm^3$

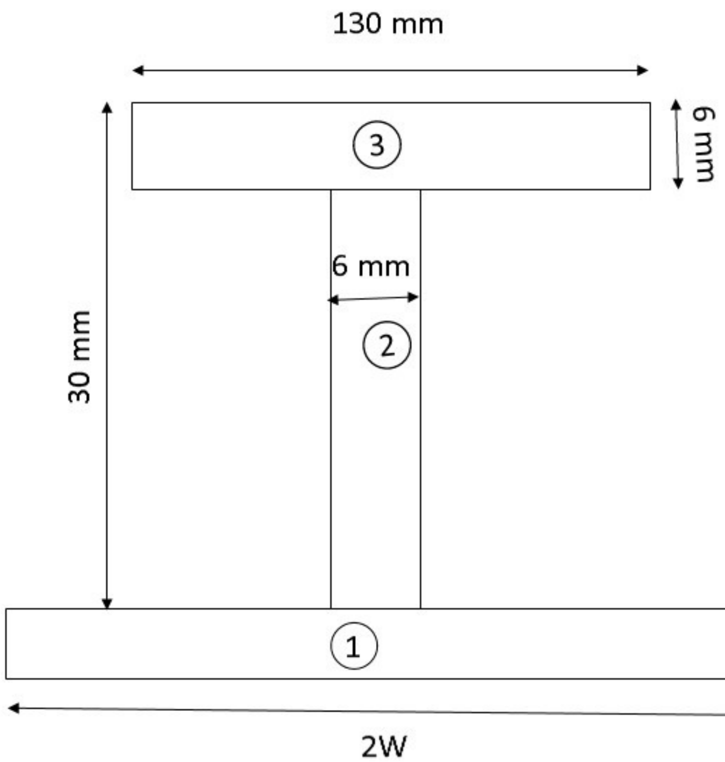

Figure 4.3: Stiffener

The effective length of shell W = 131.72 mm

Center of gravity of T Section show in Figure 4.3

$$\bar{y} = \frac{A_1 \times y_1 + A_2 \times y_2 + A_3 \times y_3}{A_1 + A_2 + A_3} = 13.20\ mm$$

$$I_s^{'} = (\frac{bd^3}{12} + A(y - \bar{y}))_1 + (\frac{bd^3}{12} + A(y - \bar{y})^2)_2 + (\frac{bd^3}{12} + A(y - \bar{y})^2)_3 = 487558\ mm^4$$

Calculation of required moment of inertia by formula,

Assume length between stiffeners ring Ls = 1200 mm

Cross section area of stiffeners ring As = 924 mm^2

$$B = \frac{3}{4}(\frac{PoDo}{t + (As/Ls)}) = 10.85$$

$$A = \frac{2B}{E} = 0.00011427$$

$$I_s^{'} = [{D_o}^2 L_s(t + A_s/L_s)A]/10.9 = 486485\ mm^4$$

Here, Required M.I. is less than Actual M.I. The design is optimum.

4.5 Number of Stiffening rings

Design length (L) = 5300 mm

Length between stiffeners (Ls) = 1200 mm

Number of stiffening ring

$$n = L/Ls = 4.42$$

Number of stiffening ring (n) = 5 $unit$

4.6 Weight of Stiffening Ring

No of stiffening ring = 5 $unit$

Density = 8027 kg/m^3

$$Volume\ of\ Stiffening\ Ring = As \times (\pi \times Do)$$

Volume of Stiffening ring = 6937767 mm^3

$$Weight\ of\ Stiffening\ ring = (No\ of\ stiffening\ ring \times Density \times Volume)/10^9$$

Weight of Stiffening ring=278.44 kg

Chapter 5

Design of Frame Structure

5.1 Introduction

Many structures require an evaluation of their structural stability such as columns, vacuum tanks and compression members. The current work frame structure is used for supporting vessel. the various types of Frames are as follows:

1. Sway Frame
2. Non-sway Frame
3. Partially sway Frame

Sway Frame

In a sway frame bottom of it is fix end with ground and top of it is pin joint as show in Figure 5.1 (a).

Non-sway Frame

In a non-sway frame bottom of it is fix end and top of it is joint with beam as show in Figure 5.2 (b).

Partially sway Frame

In a partially sway frame bottom of it is fix end and top of it is joint with two beam as show in Figure 5.3 (c)

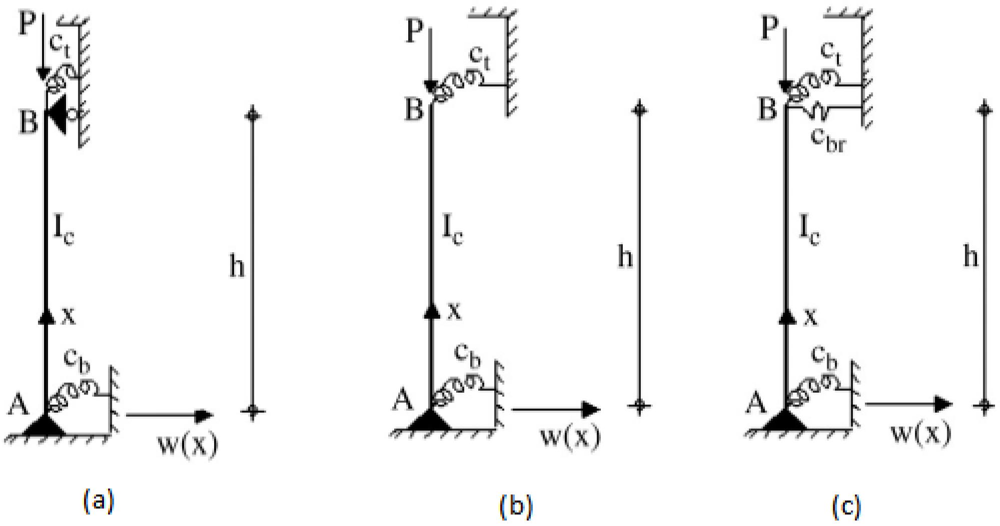

Figure 5.1: Types of Frames[19]

5.2 Selection of Material

The ratio of strength to volume, the availability of many standardized parts, the wide range of applications, the reliability of the material and the ability to give shape to nearly all engineering design. In some of the reasons to choose this material for the main structure and for other elements of a building or other construction. Safe and strong steel structures are assured by well educated designers with a Quality Controlled production and Quality Assured.

The criteria for the selection of material for vessel are as follows:

1. High strength to weight ratios

2. Uniform strength

3. Recyclability

4. Cost effectiveness

The material selected for various component frame is a500 Grade B cast iron. The cross section of metal a500 grade B is hollow and tubular. It's produced in different shape like round, square and rectangular in a variety of range of sizes.

The material properties for the same are given in Table 5.1

Table 5.1: Material property of cast iron a500 Grade B[22]

Property	Value	Unit
Tensile Yield Strength	317.16	MPa
Ultimate Tensile Strength	399.9	MPa
Young's Modulus (E)	210000	MPa
Poisson's Ratio (μ)	0.26	-
Bulk Modulus (K)	145000	MPa
Shear Modulus (G)	83000	MPa
Density	7850	kg/m^3

5.3 Loading on Frame

According to ISO 1496-1 code:1990 standard fully loaded containers must be capable of nine high stacking.The container rated is 24000 kg and acceleration force is 1.8g.

Test load on container $= 8 \times 24000 \times 1.8g = 3392kN$

5.4 Effective Length of Columns

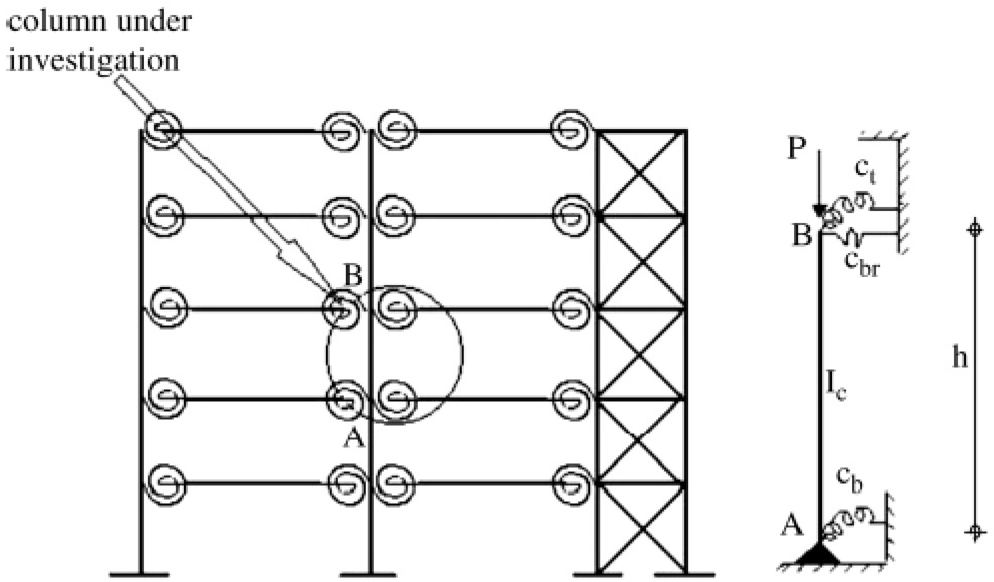

Figure 5.2: Diagram of multi structure frame[**19**]

Effective length of columns is depended on behavior of a column under compression loading. A number of boundary condition is identified by end conditions such as pinned, fixed, partially fixed, free and supported on rollers etc. In multi-layer frame, columns are continuous and connected to beam members rigidly in frame. At floor levels it is considered that they are connected rigidly. These columns become a part of either a non-sway or sway frame.

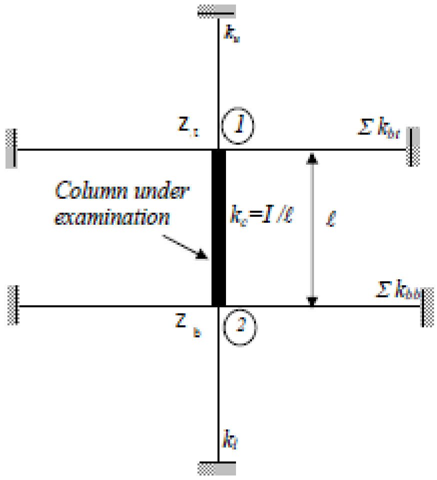

Figure 5.3: Diagram for Effective length determination[19]

$$z_t = \frac{k_c + k_u}{k_c + k_u + \sum k_{bt}}$$

$$z_b = \frac{k_c + k_l}{k_c + k_l + \sum k_{bb}}$$

The column, which is a part of the multi-layer non-sway frame, can be idealized to be a part of a limited sub-frame shown in Figure 5.3.

Let,

l_e is effective length of the column and

l is actual length between floor beams.

The effective length factor for the column is defined as

$$k = l_e/l$$

In the Figure 5.3,

k_u is relative stiffness I/l values for upper column.

k_l is relative stiffness I/l values for lower column.

$\sum k_{bt}$ is the sum of I/l values for beams framing into the column under examination at the top.

$\sum k_{bb}$ is the sum of I/l values for beams framing into the column under examination at the bottom.

The joint restraint coefficient z_n for the column at the top and bottom is obtained from

$$z_n = \frac{Column\ stiffness\ of\ columns\ meeting\ at\ the\ joint}{Total\ stiffness\ of\ all\ members\ meeting\ at\ joint\ n}$$

5.5 Buckling strength of columns[19]

The critical buckling load is then defined as

$$P_{cr} = \frac{\pi^2 EI_c}{(Kh)^2} \tag{5.1}$$

Consider the model of a column a non-sway frame, shown in Figure 5.4 resulting from the model of Figure 5.2 by replacing the transverse spring with a roller support. Denoting by w the transverse displacement and by ' the differentiation with respect to the longitudinal coordinate x, the equilibrium of this column in its buckled condition is described by the differential equation

$$w''''(x) + k^2 w''(x) = 0 \tag{5.2}$$

Where,

$k = \sqrt{\frac{P_{cr}}{EI_c}} = \pi/Kh$

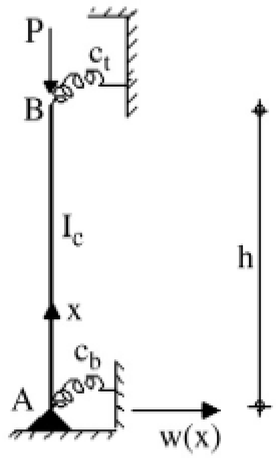

Figure 5.4: Non sway frame[19]

The general solution of this differential equation is given by:

$$w(x) = Asin(kx) + Bcos(kx) + Cx + D \tag{5.3}$$

Boundary conditions

Displacement of Transverse direction at the bottom of frame

$$w(0) = 0 \tag{5.4}$$

Equilibrium Moment at the bottom of frame

$$-EI_c w''(0) = -c_b w'(0) \tag{5.5}$$

Equilibrium Moment at the top of frame

$$-EI_c w''(h) = c_t w'(h) \tag{5.6}$$

Displacement of Transverse direction at the top of frame

$$w(h) = 0 \tag{5.7}$$

From above four boundary conditions Equation 5.4, 5.5, 5.6 and5.7 have a non-trivial solution for four constant A, B, C and D. This criterion yields the buckling equation for the effective length factor K,

$$32K^3(z_t - 1)(z_b - 1) - 4K[8K^2(z_t - 1)(z_b - 1) + (z_t + z_b - 2z_tz_b)\pi^2]cos(\pi/K)$$

$$+\pi[-16K^2 + 20K^2(z_t + z_b) + z_tz_b(\pi^2 - 24K^2)]sin(\pi/K) = 0 \tag{5.8}$$

Where,

$z_b = k_c/(k_c + k_b)$

$z_t = k_c/(k_c + k_t)$

$k_c = 4EI_c/h$

$k_t = GJ/l$

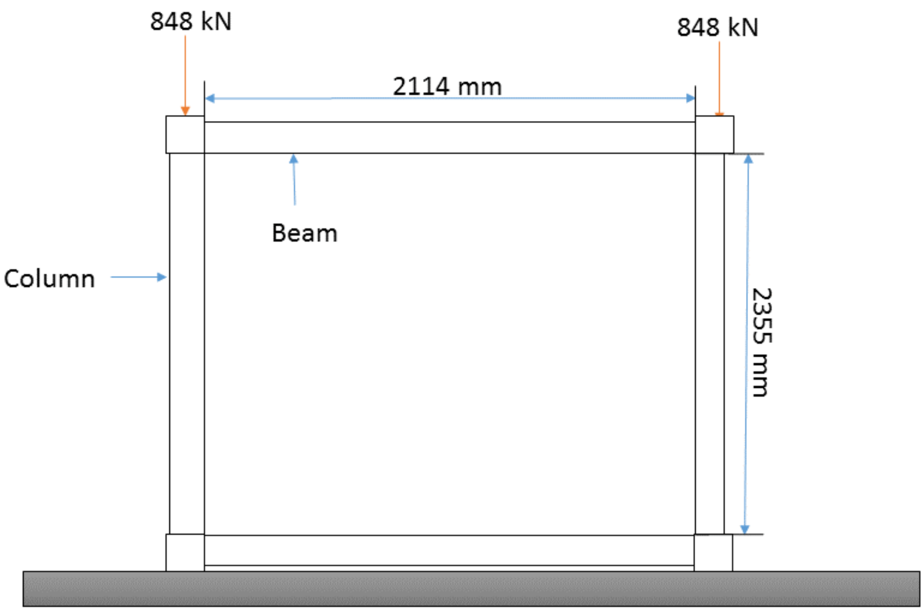

Figure 5.5: Diagram of Frame

The diagram of front view of frame is show in Figure 5.5. Assume the bottom of frame are fixed with ground.

$k_b = 0$

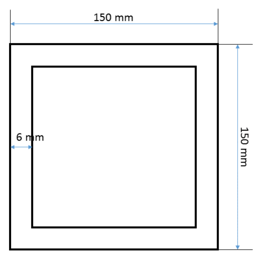

Figure 5.6: Cross section of column

The cross section of column member show in Figure 5.6,

Modulus of Elasticity$E = 210000\ MPa$

Moment of inertia of column,

$I_c = bd^3/12 - b^{‘}d^{‘3}/12$

$= 150 \times 150^3/12 - 138 \times 138^3/12 = 11.96 \times 10^6\ mm^4$

Stiffness of column,

$$k_c = \frac{4EI_c}{h} \tag{5.9}$$

$$k_c = \frac{4 \times 210000 \times 4267 \times 10^6}{2335 \times 10^6} = 4267.65\ kNm$$

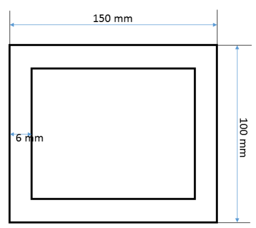

Figure 5.7: Cross section of beam

Cross section of the beam element show in figure 5.7,

Shear Modulus (G) = 83000 MPa

Polar moment of inertia

$J = bd[b^2 + D^2]/12 - b^{'}d^{'}[b^{'2} + d^{'2}]/12$

$= 150 \times 100[150^2 + 100^2]/12 - 138 \times 88[138^2 + 88^2]/12 = 13.51 \times 10^6$

$$k_t = \frac{GJ}{l} \tag{5.10}$$

$$k_t = \frac{83000 \times 13.51 \times 10^6}{2114 \times 10^6} = 530.64\ kNm/rad$$

$z_b = k_c/(k_c + k_b)$

$= \frac{4267.65}{4267.65+0} = 1$

$z_t = k_c/(k_c + k_t)$

$= \frac{4267.65}{4267.65+530.64} = 0.8894$

Substitute value z_b andz_t in Equation 5.8 , By using trail error method value of K,

K = 0.95594

Modulus of elasticity (E) = 210000 MPa

Height of column (h) = 2355 mm

Moment of inertia of column $(I_c) = 11.96 \times 10^6\ mm^4$

Substitute value K, E, h andI_c in Equation 5.1.

$$P_{cr} = \frac{\pi^2 E I_c}{(Kh)^2} = \frac{\pi^2 \times 210000 \times 11.96 \times 10^6}{(0.95594 \times 2355)^2} = 4891\ kN$$

The applied load is on single column is 848 kN which is less than critical load. Hence, member is safe under buckling.

5.6 Finite Element Analysis of frame

3D Modeling

A 3D model of structural frame is prepared. The cad model is shown in following Figure 5.8

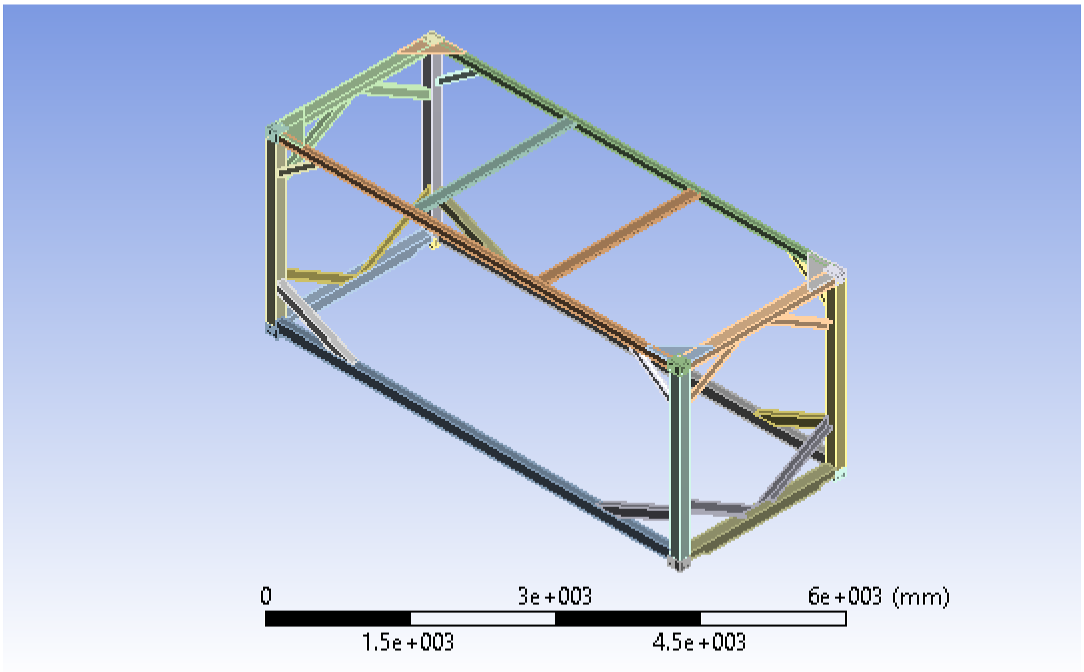

Figure 5.8: 3D Modeling

Meshed Model in Workbench

Following Figure 5.9 shows the meshed model of structural frame.

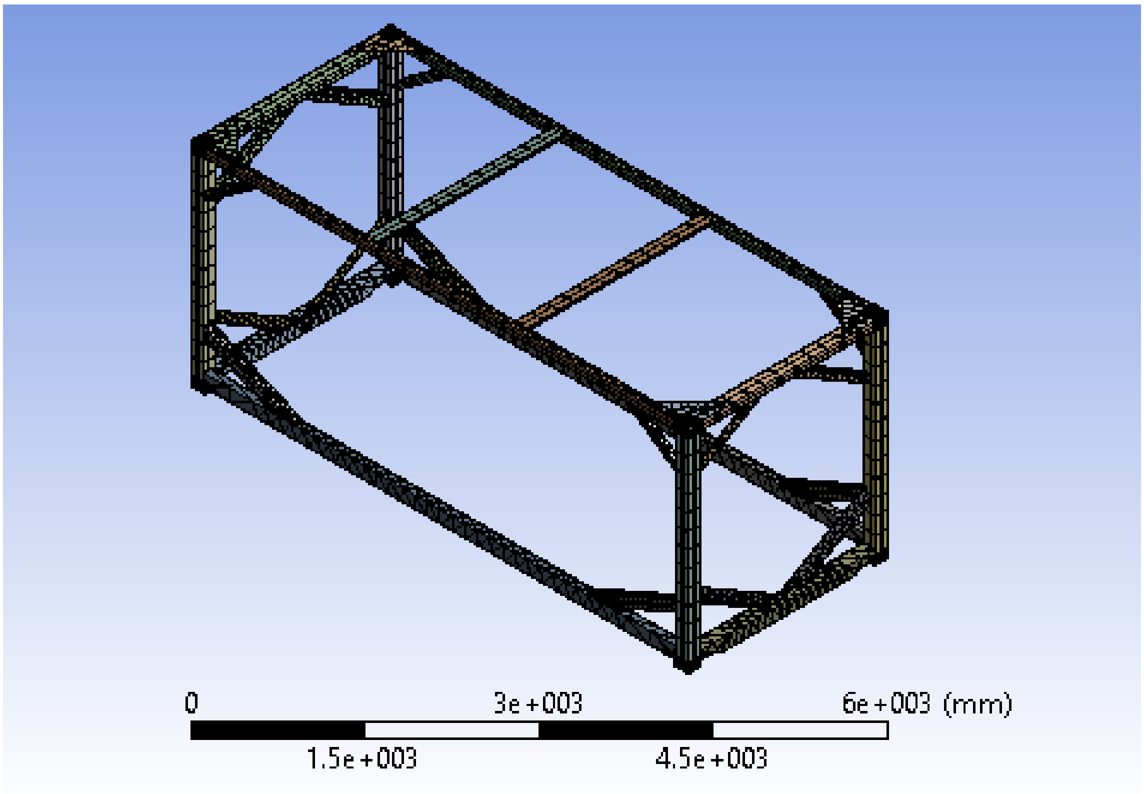

Figure 5.9: Model messing

Loads and Constraints applied

Loads applied to the structural frame are 848 kN each of the column member and bottom of the frame is fixed support.

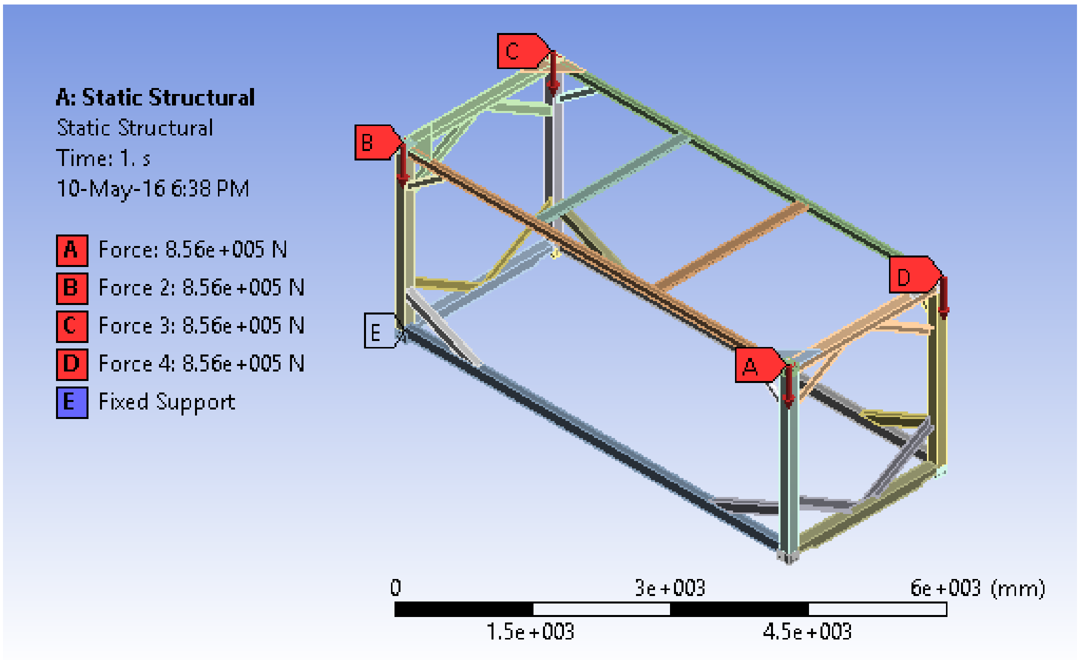

Figure 5.10: Boundary Condition

Eigenvalue Buckling

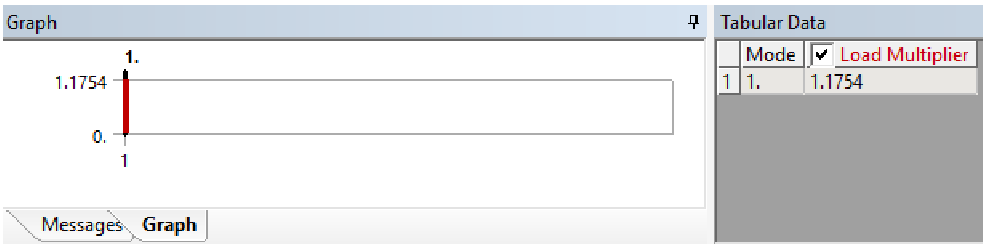

Figure 5.11: Eigenvalue Buckling

The solution of eigen value buckling analysis is show in Figure 5.11, The value of load multiplier (λ) = 1.1754.

To determine the buckling load we need product of structural loads and load multiplier (λ)

$$Buckling\ load = F \times \lambda \tag{5.11}$$

where, Structural load = 3392 kN

The analytical solution of Buckling load is 3987 kN.

The applied load is on single column is 848 kN which is less than critical load 3987kN obtained with FEA analysis. Hence, the frame structure is safe under buckling.

Chapter 6

Conclusion and Future scope

6.1 Conclusion

IMO tanks are widely used in many industries for storing and transporting hazardous and non-hazardous liquid material. The design of the tank should be done properly considering all safety measures. Design of the tank is done using specified standards such as ASME and IMDG.

The essential components of the tank is designed for internal pressure based on the maximum allowable working pressure given in addition to taking into account the static head. It is also designed based on the external pressure as per ASME code based on the magnitude of the external pressure which is mentioned in the IMDG code. The necessary ring stiffeners calculation are carried out with the iterative method so that the weight of the tank will not increase beyond the recommended value. The top of the tank is provided with manhole for purpose of cleaning, inspection and maintenance. The nozzle at the manhole is designed using ASME code and stresses at the reinforcement are verified using finite element analysis.

The tank is supported in the frame structure. Frame which is used for supporting tank is design using ISO standard. The design and necessary structural analysis of the frame structural analysis is also carried out.

6.2 Future scope

1. Design and analysis of connection between vessel with frame.

2. Thermal analysis is done for the heating system.

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Appendix

A1.1 Stored Fluid

Fluid is stored in IMO Tank show Table 6.1

UN No	Density	UN No	Density	UN No	Density
1089	783.4	2059	760	3071	1052
1108	640.5	2363	831.5	3145	940
1133	1440	2371	1594.9	3264	1000
1139	990	2478	1070	3265	1120
1155	700	2561	621.3	3266	1250
1218	680.6	2758	892	3267	975
1228	1170	2762	1600	3273	810
1243	1000	2784	960	3275	821
1250	966.3	2787	1270	3276	942
1263	910	2788	1040	3278	1273
1265	626	2801	1180	3279	871
1268	1000	2810	1580	3280	1760
1280	900	2902	980	3281	1380
1297	970	2903	910	3282	1324
1302	910	2920	1000	3286	1020
1556	1020	2924	1000	3287	1250
1719	1019	2985	878	3289	1305
1760	1480	2986	890	3295	750
1863	840	2991	980	3348	1255
1866	950	2992	1180	3351	960
1935	1130	2994	1800	3352	935
1941	2297	2996	1079	3362	890
1986	790	3006	1060	3413	1100
1988	900	3016	1170	3414	1190
1989	1299	3017	1070	3440	1486
1993	1590	3018	968	3469	900

Table 6.1: Stored Fluid