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DOC:	REV:	FILE:	DATE:
18024-10-100	A	18024.01	15 June 2018

Revision History

Section	Rev	Description	Date	Approved
All	-	Initial Issue	05/25/18	JMM
All	A	Revised to reflect design development and incorporate Columbia Power comments.	06/15/18	JMM

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Addendums

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2. WEC Float Design Loads Spreadsheet, Glosten
3. WEC Solidworks Model, 18024.SLDASM, Glosten
4. WEC Structure, Glosten Drawing Number 18024-100-01
5. WEC Ballast System, Glosten Drawing Number 18024-529-01
6. Glosten Structure Calculation 18024-835-001 Nacelle DnV Calculations-A
7. Glosten Structure Calculation 18024-835-002 Nacelle Tube DnV Calculations-A
8. Glosten Structure Calculation 18024-835-003 Pontoon DnV Calculations
9. Glosten Structure Calculation 18024-835-004 Ballast Tank DnV Calculations-A
10. Glosten Structure Calculation 18024-835-005 Upper Spar DnV Calculations
11. Glosten Structure Calculation 18024-835-006 Lower Spar DnV Calculations-A
12. Glosten Structure Calculation 18024-835-007 Knee Brace DnV Calculations-A
13. Glosten Structure Calculation 18024-835-008 Local Scantling DnV Calculations-A
14. Glosten Structure Calculation 18024-835-009 Mooring DnV Calculations
15. Glosten Structure Calculation 18024-835-010 Lifting Calculations-A
16. Glosten Structure Calculation 18024-835-011 Float DnV Calculations
17. Glosten Structure Calculation 18024-835-012 Float Arm DnV Calculations-A
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23. Glosten Ballast tank joint FEA REV A
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Introduction

This document records the work performed for the structural and ballast system design of a wave energy converter (WEC) prototype for Columbia Power (CPower). The design was developed to satisfy the CPower requirements elaborated in the System Engineering Design Requirements (Reference 1). Each report section corresponds to a Design Work Item Memo documenting the Glosten design effort.

Design Method and Basis

The WEC design methodology and basis was established in conjunction with CPower. See Addendum 1 for details.

Design Loads

This section describes the approach to developing design loads for the WEC central body and float body from the time series simulation results provided by CPower. It elaborates on the design methodology described in the design basis, providing assumptions, modeling details, and results of the finite element analyses.

CPower Simulation

The CPower provided simulation results as time series in MatLab format. Reference 1 documents the CPower hydrodynamic model, simulation approach, and load balance calculations. Reference 15 provides details of the MatLab file data structures.

The design loads developed by Glosten from these time series for the purpose of strength design consider only the extreme sea state condition documented in Reference 14. Therefore, all design load calculations assume that the float is overtopped and positioned aft of the nacelle.

We note the following clarifications and points of emphasis to Reference 1:

1) Body drift and rounding errors in the AQWA model affect the dynamic load balance. Glosten used the 9-body result set, which assumes that the float and float arms consist of a single body.
2) The time series sampling rate is approximately 0.11 Hz as opposed to 0.1 Hz to work around AQWA limitations on the number of time steps. The time step varies slightly as the time series progresses, so a constant time step cannot be assumed.
3) The Euler transformation matrix order following traditional naval architecture conventions is yaw*pitch*roll. The MatLab code in Reference 1 defines roll rotation about the Z-axis instead of the traditional X-axis. Similarly, yaw is defined as rotation about the X-axis instead of the traditional Z-axis. The code correctly defines the transformation matrix; however, the variable names are misleading.
4) A complete, continuous time series is provided without excluding any results due to initial ramping or artificially imposed end stops in the simulation. The consumer of the simulation results must apply the logical quality control signal (QCIndex discussed below) for valid results.

Coordinate system transform

AQWA dynamic simulation results are provided in terms of a fixed reference axis (FRA). The resulting loads must be transformed to a body-fixed coordinate system to apply them to a finite element model where the body orientation is fixed. The coordinate transformation varies for each time step as the bodies rotate.

Roll about FRA x axis
$$r(\theta) := \begin{pmatrix} 1 & 0 & 0 \\ 0 & \cos(\theta) & -\sin(\theta) \\ 0 & \sin(\theta) & \cos(\theta) \end{pmatrix}$$

Pitch about FRA y axis
$$p(\theta) := \begin{pmatrix} \cos(\theta) & 0 & \sin(\theta) \\ 0 & 1 & 0 \\ -\sin(\theta) & 0 & \cos(\theta) \end{pmatrix}$$

Yaw about FRA z axis
$$q(\theta) := \begin{pmatrix} \cos(\theta) & -\sin(\theta) & 0 \\ \sin(\theta) & \cos(\theta) & 0 \\ 0 & 0 & 1 \end{pmatrix}$$

Overall motion
$$qpr := q(\theta_3) \cdot p(\theta_2) \cdot r(\theta_1)$$

Figure 1: AQWA coordinate system transformation from Reference 16

The transpose of the qpr matrix in Figure 1 is used to convert FRA forces, acceleration, and velocities to body fixed coordinates.

QCIndex

The QCIndex is a logical quality control signal that must be applied to the time series to filter out the initial simulation ramp and the periods of artificially imposed end stops. Glosten extended the initial ramp and filtered the first 10 minutes of each simulation to allow the WEC to reach its mean position on the mooring system.

Pre-screening

CPower provided 10 x 3-hr hydrodynamic simulations for both the damped and undamped condition of the WEC, as well as 2 x 3-hr simulations for bidirectional wind and wave cases. An outline of the procedure for deriving design loads from the simulation results follows:

1) Pre-screen 10 x 3-hr simulations to select 4 x 3-hr worst case simulations for both the damped and undamped cases for analysis of structural loads. The pre-screening criteria considers both maximum body motions and maximum joint loads of interest. The pre-screening criteria are:
    a. Central body max heave
    b. Central body max pitch
    c. Float max heave
    d. Float max pitch
    e. Pontoon to Nacelle max bending (My)
    f. Lower Spar to Nacelle max bending (My)
    g. Ballast Tank to Lower Spar max tension (Fz)
    h. Nacelle to Float Arm max radial load (Fxz)
2) Use the 8 down selected time simulations and 2 bi-directional time simulations to develop dynamically balanced load cases with negligible restraint force for each time step.

Table 1: Pre-screening results, central body

Criteria	# FileName	VarName	Units	mean	max	min
Damped						
Max CB Heave Acceleration	b'ex_seastate_3hr_damped_6_CentralBody.csv'	b'a3'	m/s^2	-0.001337	5.41	-3.564
Max CB Pitch Acceleration	b'ex_seastate_3hr_damped_10_CentralBody.csv'	b'a5'	rad/s^2	-0.000642	0.29	-0.5731
Max FB Heave Acceleration	b'ex_seastate_3hr_damped_10_FloatBody.csv'	b'a3'	m/s^2	6.43E-03	14.85	-11.81
Max FB Pitch Acceleration	b'ex_seastate_3hr_damped_5_FloatBody.csv'	b'a5'	rad/s^2	0.002985	2.416	-1.931
Max My PontoonP to Nacelle	b'ex_seastate_3hr_damped_3_Joints_PontoonP.csv'	b'fj2_'	Nm	-2279000	1441000	-9044000
Max My LowerSpar to Nacelle	b'ex_seastate_3hr_damped_3_Joints_LowerSpars.csv'	b'fj6_'	Nm	4820000	19380000	-4402000
Max Fz Ballast to LowerSpar	b'ex_seastate_3hr_damped_6_Joints_BallastTk.csv'	b'fj7_'	N	1438000	4487000	-1464000
Max Fxz Nacelle to Float Arms	b'ex_seastate_3hr_damped_7_Joints_Nacelle.csv'	b'Fxz'	N	177800	2086000	698.1
Selected cases	**3,5,6,10**					
Undamped						
Max CB Heave Acceleration	b'ex_seastate_3hr_undamped_6_CentralBody.csv'	b'a3'	m/s^2	0.005058	5.32	-3.44
Max CB Pitch Acceleration	b'ex_seastate_3hr_undamped_8_CentralBody.csv'	b'a5'	rad/s^2	-0.00403	0.3552	-0.5855
Max FB Heave Acceleration	b'ex_seastate_3hr_undamped_4_FloatBody.csv'	b'a3'	m/s^2	0.06276	16.25	-13.19
Max FB Pitch Acceleration	b'ex_seastate_3hr_undamped_10_FloatBody.csv'	b'a5'	rad/s^2	0.02399	2.413	-1.521
Max My PontoonP to Nacelle	b'ex_seastate_3hr_undamped_7_Joints_PontoonP.csv'	b'fj2_'	Nm	-2241000	1599000	-9036000
Max My LowerSpar to Nacelle	b'ex_seastate_3hr_undamped_7_Joints_LowerSpars.csv'	b'fj6_'	Nm	4738000	18570000	-3926000
Max Fz Ballast to LowerSpar	b'ex_seastate_3hr_undamped_6_Joints_BallastTk.csv'	b'fj7_'	N	1429000	4347000	-1384000
Max Fxz Nacelle to Float Arms	b'ex_seastate_3hr_undamped_8_Joints_Nacelle.csv'	b'Fxz'	N	197400	2565000	528
Selected cases	**5,6,7,8**					

Table 1 presents the head seas simulation seeds selected for further analysis based on the pre-screening criteria. A total of 10 simulations (4 damped, 4 undamped, and the 2 bidirectional cases) were selected.

Central Body Loads

The 10 pre-screened simulations were evaluated with a brute force finite element analysis at each time step. The finite element model was created in Nx Nastran using massless beam elements with cylindrical tube sections. The analysis assumes unstiffened cross section with rule minimum thicknesses for the tube walls in order to approximate the relative stiffness between members. Table 2 documents the assumed properties of each structural element.

Table 2: Beam section properties, central body

Structural element	Radius	Thickness
	[m]	[m]
Nacelle	3.8	0.007
Nacelle Tube	1.125	0.007
Pontoon P	1.913	0.007
Pontoon S	1.913	0.007
Ballast Tank	2.35	0.007
Upper Spar P	1	0.007
Upper Spar S	1	0.007
Kneebrace P	0.483	0.007
Kneebrace S	0.483	0.007
Lower Spar P	1	0.007
Lower Spar S	1	0.007

All elements are assumed to be steel with material properties defined in Table 3.

Table 3: Material properties

Material	Elastic Modulus	Poisson's Ratio	Density
	[Pa]		[kg/m^3]
AH36 Steel	2.06E+11	0.3	0

The mass of each structural element was modeled as a point mass located at the center of gravity. Mass nodes are attached to the beam elements using rigids. Table 4 lists the total mass of the model. The port and starboard lower spars and kneebraces in the AQWA simulations are modeled as single elements on centerline due to the limitations of AQWA. The mass and inertia for these elements was split into port and starboard elements for the finite element analysis to achieve the same total mass and inertia of the central body.

Table 4: Central body mass

Structural element	Mass	X	Y	Z	Ixx	Iyx	Iyy	Izx	Izy	Izz
	[kg]	[m]	[m]	[m]	[kg-m^2]	[kg-m^2]	[kg-m^2]	[kg-m^2]	[kg-m^2]	[kg-m^2]
Nacelle	127730	0.000	0.220	0.046	1766000	0	624100	0	10100	1755000
Pontoon P	28645	-6.011	-8.344	-1.778	79570	0	284400	-74550	0	257300
Pontoon S	28645	-6.011	8.344	-1.778	79570	0	284400	-74550	0	257300
Ballast Tank	523950	0.291	0.000	-13.830	18870000	0	1165000	45630	0	19300000
Upper Spar P	6607	-1.430	-8.344	3.930	20620	0	22990	6503	0	5121
Upper Spar S	6607	-1.430	8.344	3.930	20620	0	22990	6503	0	5121
Kneebrace P	1940	-5.524	-8.344	-8.608	8546	0	17797	8653	0	9704
Kneebrace S	1940	-5.524	8.344	-8.608	8546	0	17797	8653	0	9704
Lower Spar P	11096	0.000	-8.344	-6.135	107072	0	107072	0	0	10887
Lower Spar S	11096	0.000	8.344	-6.135	107072	0	107072	0	0	10887
Total Central Body	748256	-0.310	0.038	-9.970	55147502	8722	32192942	-3774485	291546	30537296

Figure 2 illustrates the finite element model and boundary conditions. Each load case is dynamically balanced with the calculated acceleration and velocities from the simulation time step. Therefore, the global restraints shown at the end of the pontoons are only necessary to constrain the slight imbalance between loads and inertia described in Reference 13. Constraint reaction forces and moments are small, on the order of 20 kN and 10 kNm respectively.

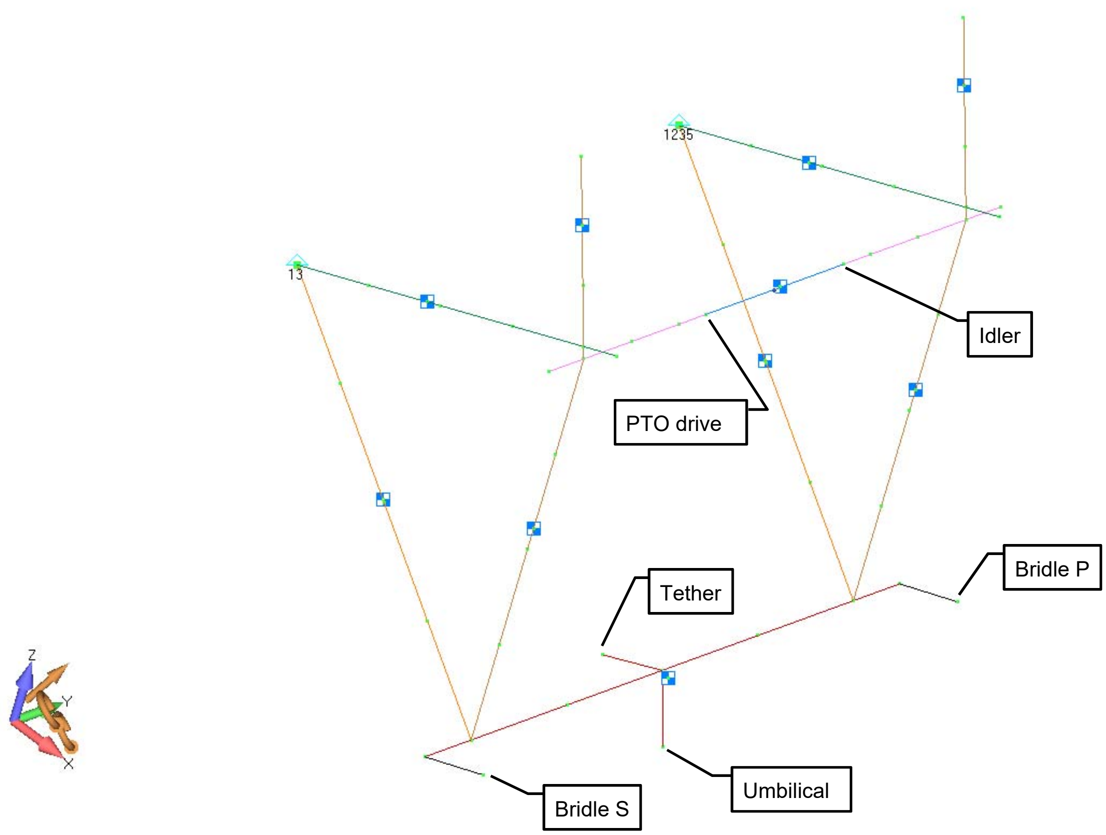

Figure 2: Central body FEA, boundary conditions

Functional loads from the PTO and float arm joint are applied at the float arm interface rather than the center of gravity of the nacelle. The PTO drive node is assumed to carry all moment and thrust. Radial loads from the float arm are divided between the drive side and idler side nodes.

The bridle, tether, and umbilical forces are applied at the mooring attachment points identified in Reference 15. The nodes representing the bridle points are connected to the beam model using rigid elements. The tether and umbilical are attached using stiff beams with a cross section equal to the ballast tank.

The finite element model was loaded and analyzed for the extreme sea state in a total of 10 x 3-hr simulations, including 4 x 3-hr simulations for the damped case, 4 x 3-hr simulations for the undamped condition, 1 x 3-hr bi-directional wind/wave simulations damped, and 1 x 3-hr bi-directional wind/wave case undamped. Every other time step in the simulation was evaluated in order to keep file size and analysis time within manageable limits.

The finite element analysis results were post-processed to identify the maximum and minimum bending moment, axial force, shear, and torsion in each of the 9 structural elements of the AQWA model. The post-processing step filters out time steps flagged by a CPower provided QCIndex. Table 5 presents the structural design load cases that maximize each process along with the concurrent loads at that time step.

The original intent of the 10 simulations was to provide data for statistical evaluation of the loads. This approach was abandoned in favor of providing concurrent balanced load sets for structural design that maximize each process of interest. The results in Table 5 indicate that the bidirectional cases govern for many load processes, so a statistical approach is not possible in those cases and the adopted approach lends consistency.

Table 5: Central body design load cases

Design Force and Moments - Extreme Seas							
	Notes:						
	1) *Bending moments and shear are provided as vector sum of values for local y and z element axes.						
	(Local element x-axis is in axial direction of each cylinder.)						
	2) One concurrent load case provided that maximizes each process.						
	3) Time in simulation = isub*0.1111 seconds.						
	4) Maximum values for any seed presented without averaging or partial safety factors.						
	5) FEA model assumes the drive arm side of the nacelle tube carries the entire moment.						
Process -->	**Bending Moment***	**Shear***	**Axial Force**	**Axial Force**	**Total Torque**		
Body	[kNm]	[kN]	**Max** [kN]	**Min** [kN]	[kNm]	**Source**	**isub**
Nacelle	**8557**	1025	---	-59	-23	5D_Nacelle	19524
Nacelle	8022	**1082**	---	-220	427	BiDirU_Nacelle	69688
Nacelle	5403	516	**611**	---	-166	BiDirU_Nacelle	89040
Nacelle	1947	383	---	**-646**	-485	BiDirD_Nacelle	90572
Nacelle	2651	568	---	-423	**1932**	BiDirD_Nacelle	50502
Nacelle Tube	**6073**	683	---	-303	-1737	10D_NacelleTube	7350
Nacelle Tube	605	**1144**	---	-84	192	BiDirD_NacelleTube	88970
Nacelle Tube	5012	516	**611**	---	-128	BiDirU_NacelleTube	89040
Nacelle Tube	3720	357	---	**-767**	-1028	BiDirU_NacelleTube	90572
Nacelle Tube	2710	663	---	-453	**1932**	BiDirD_NacelleTube	50502
Pontoon	**7755**	876	---	-161	35	10D_Pontoon	51312
Pontoon	4761	**894**	---	-497	48	6U_Pontoon	94862
Pontoon	3601	619	**468**	---	-152	5U_Pontoon	44820
Pontoon	518	426	---	**-866**	64	6U_Pontoon	45216
Pontoon	705	372	---	-155	**299**	BiDirD_Pontoon	62278
BallastTk	**18977**	2342	---	-67	954	BiDirD_BallastTk	88972
BallastTk	1812	**2373**	128	---	-1220	BiDirD_BallastTk	69692
BallastTk	3281	2080	**347**	---	-347	BiDirU_BallastTk	33222
BallastTk	3224	704	---	**-461**	-2386	10D_BallastTk	7350
BallastTk	634	520	89	---	**2307**	BiDirD_BallastTk	89030
UpperSpar	**511**	121	108	---	0	7U_UpperSpar	53772
UpperSpar	108	**148**	-59	---	-1	BiDirD_UpperSpar	92622
UpperSpar	67	58	**116**	---	1	7U_UpperSpar	53550
UpperSpar	181	35	---	**-100**	-1	BiDirD_UpperSpar	43644
UpperSpar	46	8	12	---	**3**	BiDirD_UpperSpar	71916
LowerSpar	**3203**	362	1108	---	-92	BiDirU_LowerSpar	89068
LowerSpar	2855	**418**	827	---	1487	10D_LowerSpar	7350
LowerSpar	1131	120	**1856**	---	227	BiDirD_LowerSpar	88970
LowerSpar	660	126	---	**-839**	-71	6U_LowerSpar	88530
LowerSpar	665	401	631	---	**1501**	10D_LowerSpar	7350
KneeBrace	**1019**	112	---	-227	174	10D_KneeBrace	7350
KneeBrace	193	**122**	---	-281	178	10D_KneeBrace	7350
KneeBrace	7	29	**1096**	---	34	3D_KneeBrace	30132
KneeBrace	121	38	---	**-382**	-71	10D_KneeBrace	33664
KneeBrace	287	53	223	---	**253**	BiDirU_KneeBrace	22606

The maximum local design pressures from depth and slamming were provided by CPower. These loads are not included in the finite element analysis, but they are included as loads in the design calculations.

Float Design Loads

Structural design loads for the float are also determined from processing the CPower hydrodynamic simulations. Given the relative simplicity of the float body compared to the central body assembly, a subset of cases was selected representing time steps when the vector components of the float body loads are either maximized or minimized. In determining this subset, the 10 x3-hr time series and 2 x3-hr bi-directional time series data are evaluated. As previously described, the results filter out time steps by QCIndex and the first 10 minutes of the simulation allowing the WEC to find equilibrium on the moorings. Table 6 lists the selected concurrent load cases for use in structural design of the float body assembly.

Table 6: Concurrent load cases, float body

FEA Case	Mode	Criteria	Fx [kN]	Fy [kN]	Fz [kN]	Mx [kNm]	My [kNm]	Mz [kNm]	Fxz [kN]	Mxz [kNm]	time index	source
5	1	max	**2569**	-133	18	-406	-150	667	2572	781	28524	Undamped_8
2	1	min	**-1913**	68	26	2320	-125	1030	1915	2538	6600	Bidir undamped
3	2	max	764	**660**	56	1965	-395	-811	1009	2125	12264	Bidir undamped
18	2	min	253	**-828**	22	1960	-99	-825	866	2127	89040	Bidir undamped
15	3	max	65	-114	**241**	-883	-536	-354	132	951	75270	Bidir damped
10	3	min	-133	-16	**-184**	-83	123	-95	134	126	59979	Bidir damped
1	4	max	-144	463	45	**3186**	-325	-181	484	3191	6221	Bidir undamped
17	4	min	-667	-286	66	**-3050**	-382	-1397	726	3355	88977	Bidir undamped
4	5	max	-94	1	110	-281	**350**	-77	94	291	28140	Bidir damped
14	5	min	663	-192	-63	-1909	**-598**	564	690	1991	75267	Bidir damped
13	6	max	-706	53	95	615	-78	**2253**	708	2335	73745	Bidir damped
16	6	min	1454	190	40	1477	-293	**-2436**	1466	2849	75878	Bidir undamped
6	7	max	2569	-133	18	-406	-150	667	**2572**	781	28524	Undamped_8
11	8	max	493	253	-63	3114	-513	-1817	554	**3605**	67226	Bidir damped

A simplified finite element model using beam elements was created for the float body and loaded with the concurrent loads defined above. Table 7 presents the assumed section properties for the structural elements in the float body model.

Table 7: Beam section properties, float body

Structural element	Shape	Size	Thickness
		[m]	[m]
Float	Cylinder	R 2.357	0.0079375
Float arms	Square tube	0.5588 x 0.5588	0.022225
Torque tube	Cylinder	R 1.143	0.0127

Material properties for the float body model are identical to those in Table 3.

Figure 3 illustrates the geometry and boundary conditions of the float body finite element model. The fixed node located at the center of gravity of the float body is not attached to the model and this constraint does not impact the results. The node is used as a reference point for defining rotational accelerations and velocities. Idler bearing nodes 1 and 16 are assumed to be pinned radially. Generator node 63 is pinned in three degrees of freedom and fixed in rotation about the Y-axis.

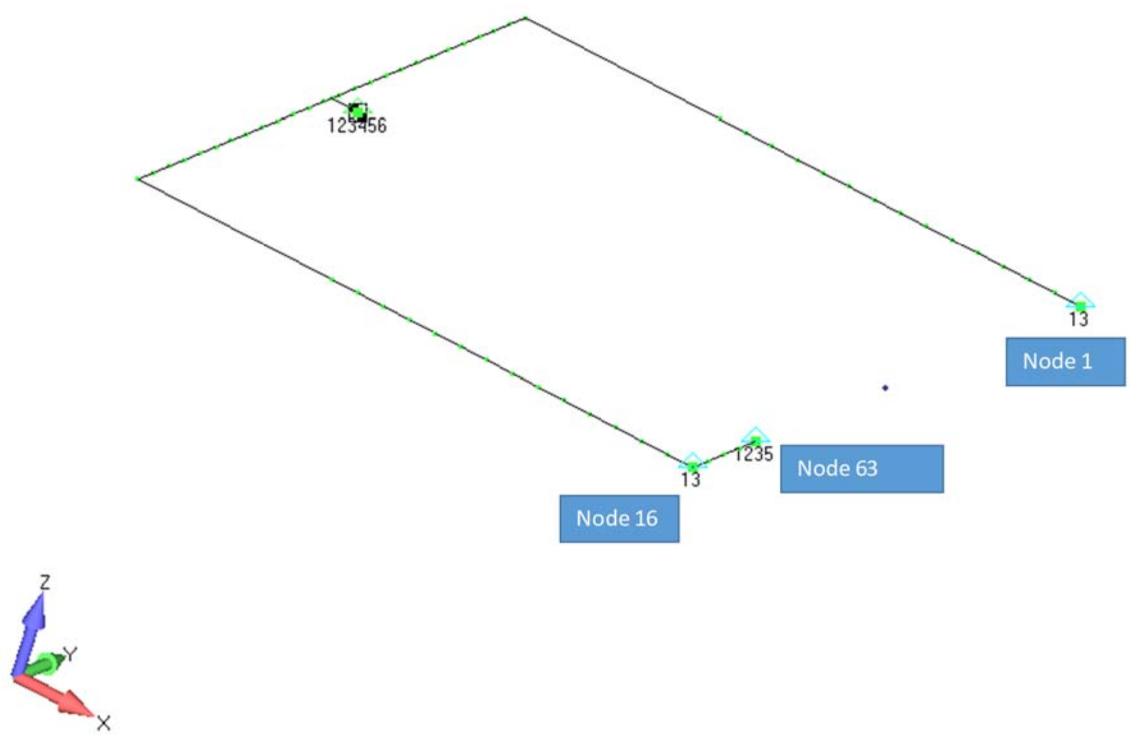

Figure 3: Central body FEA, boundary conditions

Hydrodynamic and inertial loads are applied at the float body center of gravity. The PTO torque is applied at the generator node. The port and starboard float arms are modeled as a single element in the AQWA simulation. Therefore, the following assumptions are made in applying the nacelle/float arm joint loads to the FEA model:

1) The radial loads are shared equally between nodes 1 and 16.
2) Radial moments resolved to a force couple between nodes 1 and 16.

The sum of the forces and moments applied to the finite element model is equal to zero, such that the load cases are fully balanced.

Table 8 summarizes the bending moments, axial force, shear forces, and torque for the float, float arms, and torque tube from the finite element analysis for design of these structural elements. The table presents the results in the element coordinate system shown in Figure 4. The element Y-axes are aligned with the global XY-plane.

Table 8: Float body design loads

Process		**Maximum values**			**Minimum Values**		
Process	Units	1..arm	2..float	3..torque tube	1..arm	2..float	3..torque tube
Beam EndA Axial Force	[N]	1055036	660223	828435	-1449256	-828435	-660223
Beam EndA Pl1 Shear Force	[N]	660223	1449256	1174700	-828435	-1155742	-1462496
Beam EndA Pl2 Shear Force	[N]	411786	388359	319927	-393789	-411786	-323793
Beam EndA Plane1 Moment	[Nm]	1567903	4834169	2499629	-1830740	-3871390	-2676373
Beam EndA Plane2 Moment	[Nm]	1690267	1838835	393452	-1372772	-1803240	-398205
Beam EndA Torque	[Nm]	393452	2640590	1052117	-398205	-2566062	-1052118
Beam EndB Axial Force	[N]	1055036	660223	828435	-1449256	-828435	-660223
Beam EndB Pl1 Shear Force	[N]	660223	1449256	1174700	-828435	-1155742	-1462496
Beam EndB Pl2 Shear Force	[N]	411786	388359	319927	-393789	-411786	-323793
Beam EndB Plane1 Moment	[Nm]	1825671	4616329	2848019	-1572972	-3961851	-2964690
Beam EndB Plane2 Moment	[Nm]	1743322	1780504	377490	-1481301	-1741364	-382062
Beam EndB Torque	[Nm]	393452	2640590	1052117	-398205	-2566062	-1052118

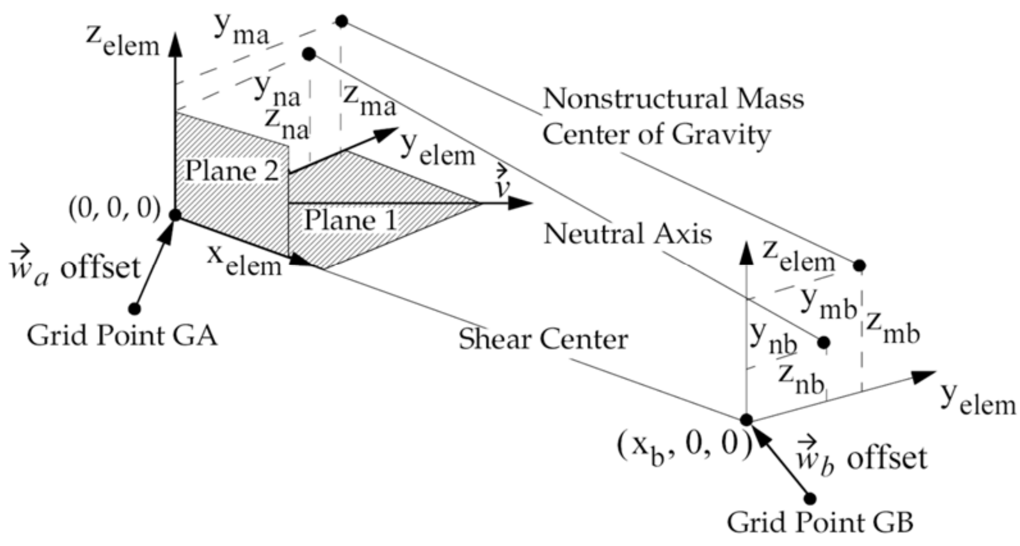

Figure 4: Beam element coordinate system

The float body design load spreadsheet, Addendum 2 contains further details of the float body design loads.

Structural design

The WEC is assumed to be fabricated steel structure formed by a combination of cylindrical shell structures. The major structural members are the nacelle, nacelle tube, pontoons, ballast tank, upper spar, lower spar, knee brace, float and float arms. The geometry was provided by CPower. The resulting 3D structural model and structural arrangement are provided in Addendums 3 and 4, respectively. The 3D structural model also includes notional CPower electrical cabinets and transformer within the nacelle providing one example equipment arrangement.

Structural analysis and calculations

The following section outlines the design development of the WEC.

Shells

Each cylindrical shell member of the WEC was analyzed in accordance with the LFRD method outlined in Reference 1.

Global analysis

Main body loads

Eight (8) independent load cases were considered for each major cylindrical shell member of the main body of the WEC (all members except float and float arm):

1. Positive bending
2. Negative bending (assumes same magnitude as positive bending and corresponding concurrent loads)
3. Shear
4. Positive axial
5. Negative axial
6. Torsion
7. Negative pressure (hydrostatic pressure acting inward)
8. Positive pressure (variable ballast compressed air pressure acting outward)

Each case included the concurrent values for the loads in the other degrees of freedom including the summation of theworst hydrostatic and wave pressure (external sea pressure) at the of the submerged bodies per Reference 8. The sea pressure is added to ends bulkheads to induce an additional component of the total axial stess.

The free body loads for Cases 1-6 were derived from an FEA described in the Design Loads section and are considered environment loads. A PSF of 1.35 was applied to all loads to account for the 50-year return period assumed to predict the forces outlined in the design loads (Addendum 2) as recommended in the IEC TS (Reference 4).

Cases 7 and 8 only considered the respective pressures as the events occur when there are no discernible free body loads, e.g., WEC completely submerged and static for Case 7 or ballasting in calm seas for Case 8. Case 7 is considered a permanent load and Case 8 is a functional load, both of which are well defined, therefore a PSF of 1.2 is assumed as per Reference 2, Section 2, D402.

Float loads

Four (4) independent load cases were considered for float and float arm:

1. Maximum for all degrees of freedom (no hydrostatic pressure)
2. Minimum for all degrees of freedom (no hydrostatic pressure)
3. Negative pressure (hydrostatic pressure acting inward)
4. Positive pressure (variable ballast compressed air pressure acting outward)

Similar to the main body loads, a PFS of 1.35 was applied to all free body loads outlined in Addendum 2 and 1.2 was applied to the pressure loads.

Resistance evaluation

The global stresses were derived from formulae for stresses in closed cylinders per Reference 3, Section 2.2. In general, the material PSF is 1.15 as per Reference 1. All shell plating is assumed to be NV AH36 steel (equivalent to ABS AH36) in an effort to reduce weight. All internals such as ring frames and bulkheads where required for buckling, slamming, or subdivision (see Ballast System and Control) are NV A (equivalent to ABS Grade A).

Shell and internal scantlings were developed to provide satisfactory resistance against ultimate limit state and buckling. In addition, resistance against fatigue limit state was checked (see Fatigue Assessment). See Addendums 6-12 and 16-17 for calculations. Table 9 summarizes the minimum scantlings of each member:

Table 9: Scantling Summary

Member	Plating	Stiffening
Nacelle	3/8” NV AH 36	5” x 1/4” web, 6” x 3/8” flange NV A
Nacelle Tube	1/2” NV AH 36	N/A
Pontoon	7/16” NV AH 36	4” x 7/16” web, 4” x 7/16” flange NV A
Ballast Tank	9/16” NV AH 36	4” x 1/4” web, 4” x 3/8” flange NV A
Upper Spar	1/4” NV AH 36	2” x 3/8” web NV A
Lower Spar	3/8” NV AH 36	3” x 5/16” web NV A
Knee Brace	5/8” wall ASTM 500 GR B tube	

Local analysis of shell plating in way of areas of slamming

The shell plate thickness of members subjected to slamming, i.e., those members at or above the free surface, was evaluated in accordance with Reference 2, Section 5, F300. Slamming loads were provided by CPower for the nacelle, pontoon, float and float arm in Reference 12. Slamming loads for the upper spar, lower spar and knee brace were taken as the greater of side shell sea loading, exposed deck loading or lowest tier forward external superstructure bulkheads per Reference 5. The pressure was applied over a 60° arc for conservatism. This approach accounts for the global stresses by a proportional reduction in design bending stress. A correction factor for curved plates, k_r, was applied to the required thickness for unstiffened shells as permitted by Reference 5, Section 6, H200. This slamming check is included in the addendums referenced in the previous section.

An FEA was developed to check the nacelle, pontoon and float ring frames against the slamming loads (see Figures below). The beams ends were defined as a 60° arc as this is considered self-supporting and fixed. The pressure was applied over the arc specified in Reference 11. A PSF of 1.5 was applied to the pressure and a material PSF of 1.15 was assumed. Solidworks Simulation files evaluating slamming pressure on the ring frames are provided in Addendums 27-29.

Local analysis of joints

FEA was performed on the following joints as there was no available SCF for the specific arrangements: 1) pontoon, nacelle tube, main spar, and upper spar; 2) pontoon and knee brace; and 3) ballast tank, main spar and knee brace. Solidworks Simulation files for these joints are provided in Addendums 21-23.

The FEM consisted of the primary member modeled to half span where it was fixed. The intersecting members were modeled to a point 1 m from the intersection to the primary members. The loads from the case that resulted in the highest global stress were applied to each intersecting member, respectively. Artificial axial, bending, and torsional loads were applied at the free end bulkhead (assumed as a 1" thick rigid plate to uniform ensure load transfer) of the primary member and shear loads at the bulkhead in way of the joint such that the reaction at the fixed end was equivalent to the worst case free body loads in order to develop the corresponding far field stresses.

The calculation of the hot spot stress at the joint to determine if it required additional reinforcement was performed in accordance with Reference 6, Section 4. Local and hot spot stresses are not to exceed yield as per Reference 7, 5.2.3.2.

Figures in Appendix A show the resultant Von Mises stress of the joints. Hot spot stresses extrapolated from the FEA are summarized in Table 11 and Table 12 in the Fatigue Assessment section.

Bulkheads

Typical bulkheads

Each bulkhead was designed in accordance with the DNV Ship Rules (Reference 5), Section 9 (bulkhead structures). The design pressure was assumed to be the greater of the pressures from Load Cases 7 and 8. A PSF was not applied to the pressure as the WSD approach inherent in the equation assumes a factor of safety that is comparable to that derived in the LFRD approach as the pressure is the only design load considered. It was assumed there was negligible membrane stress due to the pressure loading of the cylinders, i.e., the cylinder absorbed this stress as hoop stress.

The nontight nacelle diaphragm bulkhead adopts the same plating as the nacelle tube and a 50% reduction in stiffener scantlings as permitted by Reference 8, 3-2-7/5.1.

The pontoon and float end bulkheads were also checked to the requirements of Reference 5, Section 7 (side structures) assuming the external design pressures per Reference 5, Section 7, Table B1; and to the requirements of Reference 5, Section 10 (superstructure ends) assuming unprotected front bulkhead design pressures per Reference 5, Section 10, Table B1.

The upper spar top end cap was checked to the requirements of Reference 5, Section 8 (deck structures) assuming weather deck design pressures per Reference 5, Section 8, Table B1; and to the requirements of Reference 5, Section 10 (superstructure ends) assuming unprotected front bulkhead design pressures per Reference 5, Section 10, Table B1. It was assumed that the bulkhead in question would be at the waterline for the upper spar submerged case and would not be subjected to a hydrostatic pressure.

These calculations may be found in Addendum 13.

Nacelle end bulkheads

The design of the starboard end bulkhead in way of the generator assumed a radially stiffened grillage arrangement in order to provide access around the generator assembly and accommodate the bolting arrangement. An FEA was performed adopting the worst-case design condition of the body being entirely submerged. The resulting input loads equal the highest hydrostatic pressure times a PSF of 1.2 (per Reference 2, Section 2, D402) over the entire face as the bulkhead orientation with respect to the world could vary. In addition, the torsion induced by the generator was applied with a PSF of 1.35. A material PSF of 1.15 was assumed. Solidworks Simulation files of the bulkhead are provided in Addendum 24.

Platforms

Each platform within the nacelle was designed in accordance with the DNV Ship Rules (Reference 5), Section 8. The design pressure was assumed to be that for platform decks in machinery spaces as per Reference 5, Section 8 Table B1. A PSF was not applied to the pressure as the WSD approach inherent in the equation assumes a factor of safety that is comparable to that derived in the LFRD approach as the pressure is the only design load considered. These calculations may be found in Addendum 13.

Note, the platform may require local reinforcement pending equipment loads and foundation arrangements. Reinforcements will be determined by the fabricator during the detail design phase.

Gratings were not analyzed. Grating is assumed to be removable steel or aluminum plates in machinery spaces and fiberglass molded grating otherwise. In general, gratings are to be supported by a steel lattice composed of angle.

Mooring and Towing Attachments

The mooring and towing fittings and respective support structure were designed in accordance with the DNV Position Mooring Standard (Reference 9).

The mooring fittings are sized to accommodate the minimum breaking strength of mooring line of 1480 kN (151 t) provided by CPower.

The WEC ballast tank (main drag inducing component) has an approximate drag of 8 t at 3.5 knots. A load factor of 1.5 was assumed to account for wave making drag, and wave and wind loading resulting in a 12 t tow load. The minimum breaking strength of the tow line is assumed to be 3 time the tow load, or 36 t, as per Reference 9. The towing fittings design is based on the breaking strength of the line.

A notional tug suitable for towing the WEC is the Young Brothers' *Tug Miki'Ala.* It is a locally operated tug (based in HI) with sufficient bollard pull capacity of 82 kips (37 t). In addition, the large working deck can accommodate additional logistical support operations for WEC deployment.

For both, mooring and towing, RUD VRBS 50 t and VRBS 16 t load rings were selected as the mooring and towing fittings, respectively, for the hinge design to better manage the varying load angles. The rings are welded to base plates that stand off from the hull on foundation pedestals to preclude interference between the shackle and hull. First principle stress and beam analyses were performed to size the pedestals and hull internal backing structure. All structure was assumed to be NV A steel. Calculations are provided in Addendum 14 and the structural details of the mooring and towing fittings and foundations can be found in Addendums 3 and 4.

Umbilical Connection

The umbilical connection design was extended from the aforementioned mooring attachments as the loads were assumed to be comparable.

Fabrication and Assembly

The WEC is assumed to be of steel construction (see structural analysis). In general, all shell plating will be either rolled or chip broken plate. The one exception is the knee brace, which is made from pipe. The WEC was modeled in module assemblies to align with the following build sequence:

1. Nacelle (1C)
    a. Fabricate port and starboard nacelle structural assemblies (1C-100P and 1C-100S, respectively)
        i. Assemblies include nacelle tube up to ~1 m from pontoon intersection
            1. Starboard nacelle tube is loose
    b. Outfit port nacelle
        i. Install all electrical equipment from opening on inboard side
    c. Outfit starboard nacelle
        i. Install bearing housing (assume bearings are contained within housing) to loose nacelle tube from outboard side
        ii. Install nacelle tube/bearing assembly to bearing flange on nacelle bulkhead from inboard side
        iii. Install generator from opening on inboard side
    d. Join port and starboard nacelle at erection joint
    e. Install pipe systems
    f. Install port and starboard float arm idler bearing and housing assemblies (includes float arms, 4C-100P/S) to nacelle tube from free end
        i. Starboard side arm includes torque tube
2. Float (4C-100)
    a. Fabricate float assembly
    b. Outfit float assembly
    c. Attach float to float arms
3. Pontoon tripods (2P/2S)
    a. Fabricate and outfit port and starboard pontoon tripod assemblies
    b. Join tripods to nacelle at erection joint
    c. Install pipe system make-up spools
4. Ballast tanks (3C)
    a. Fabricate ballast tank assembly
    b. Joint ballast tank to tripods at erection joint

See Figure 5 below for conceptual exploded view.

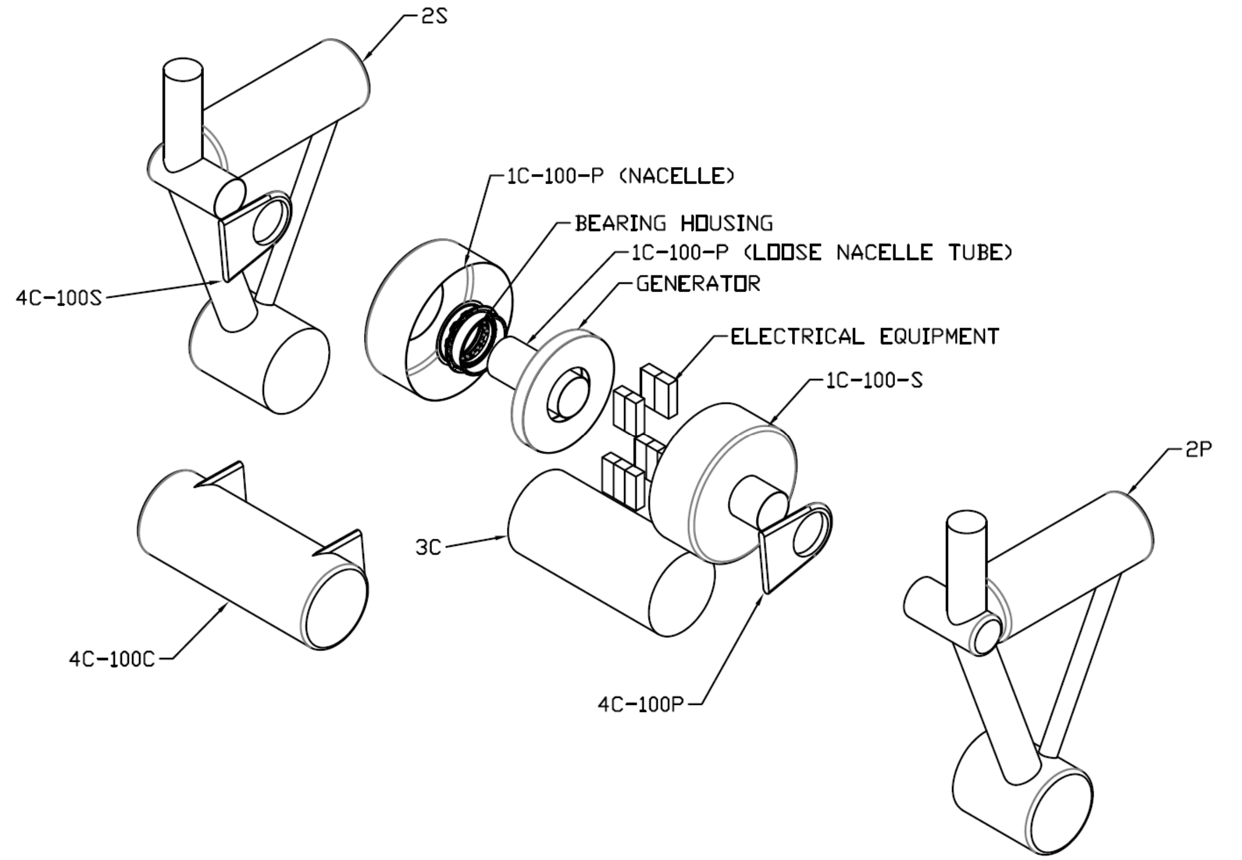

Figure 5: WEC Modules

The bearing interfaces in the WEC require machined surfaces. The PTO interfaces with a circular ring and flange on the nacelle end bulkhead. This flange will be machined flat so that the PTO bearing can be bolted directly to this surface. The PTO ring flange becomes the reference datum to align the remaining bearings.

The PTO has a second thrust bearing that interfaces with a ring welded to the exterior of the nacelle tube. This welded ring is the thrust ring. The thrust ring will need to be aligned with the bolt flange and machined flat. Alternately, Chockfast could be used to provide the interface between the PTO bearing and the thrust ring.

The idler bearings supporting the float arms also have machined surfaces. The design includes a steel bearing housing on the end of the float arm. The idler bearing will be inserted between the housing and nacelle tube. The idler bearing concept involves installing a machined circular stainless-steel liner over the nacelle tube at the idler bearing. This liner is aligned with the PTO bolt flange and secured to the nacelle tube with Chockfast. The nacelle tube liner forms the inner race. We assume the bearing material will be UltraComp, Orkot, or similar. The bearing material has to be machined to match the inner and outer race. The outer race could be another stainless steel liner that is Chockfast to the arm bearing housing. The idler bearing design will be accomplished by others in a subsequent design phase. The compatibility of the bearing housing must be verified against the final bearing design.

Another set of machined interfaces are the torque tube and mating flanges connecting the PTO to the drive arm bearing housing. These flanges have to be machined to allow the torque tube to be fit between these components and bolted together.

The lightship weight of the WEC is comprised of structure, electrical, control and monitoring equipment, auxiliary systems, and outfitting. The structural weight was calculated from the

Solid Works model (Addendum 3) and includes a 3% allowance for welding and millage. The structural weight is also burdened with a 5% margin. CPower provided weights for the generator and electrical systems. An estimated allowance is included for all other weights since details about these components and systems are not defined. The estimated weight of the WEC by module is summarized in Table 10 below (the spreadsheet is provided in Addendum 33).

Table 10: WEC Weight Estimate

	Weight kg	x m	y m	z m	Notes
WEC	**338125**	**-2.49**	**-0.17**	**-3.71**	"transport" configuration
Main body	**299229**	**-2.58**	**-0.17**	**-3.55**	
Nacelle	**124481**	**0.00**	**-0.40**	**0.00**	
1C	58474	0.00	-0.06	0.00	from SW model; includes all pertinent SWBS and 5% margin plus 3% weld/mill allowance
Paint	2244	0.00	-0.06	0.00	typ MARCO paint factor
Anodes	103	0.00	-0.06	0.00	
Generator (incl rotor, main bearing, drive shaft)	48000	0.00	-1.50	0.00	per CPwr; includes wire/cable and 20% margin
Control room (incl PE, SCADA)	12000	0.00	1.50	0.00	per CPwr; includes wire/cable and 20% margin
Transformer	2880	0.00	2.50	0.00	per CPwr; includes wire/cable and 20% margin
Outfit allowance (bilge, vents, lights)	779	0.00	0.00	-1.00	discrete weight plus 20% margin and guess at CG
Port Tripod	**73664**	**-4.34**	**8.17**	**-4.81**	
2P	69484	-4.39	8.17	-4.92	from SW model; includes all pertinent SWBS and 5% margin plus 3% weld/mill allowance
Paint	2601	-4.39	8.17	-4.92	typ MARCO paint factor
Anodes	480	-4.39	8.17	-4.92	
Port spar top equipment	600	-2.15	8.34	5.92	per CPwr; includes wire/cable and 20% margin
Outfit allowance (bilge, vents, lights)	500	0.00	8.35	-1.00	discrete weight plus 20% margin and guess at CG
Stbd Tripod	**73664**	**-4.34**	**-8.17**	**-4.81**	
2S	69484	-4.39	-8.17	-4.92	from SW model; includes all pertinent SWBS and 5% margin plus 3% weld/mill allowance
Paint	2601	-4.39	-8.17	-4.92	typ MARCO paint factor
Anodes	480	-4.39	-8.17	-4.92	
Stbd spar top equipment	600	-2.15	-8.34	5.92	per CPwr; includes wire/cable and 20% margin
Outfit allowance (bilge, vents, lights)	500	0.00	-8.35	-1.00	discrete weight plus 20% margin and guess at CG
BT	**27420**	**-4.78**	**0.00**	**-12.90**	
3C	26182	-4.78	0.00	-12.90	from SW model; includes all pertinent SWBS and 5% margin plus 3% weld/mill allowance
Paint	1046	-4.78	0.00	-12.90	typ MARCO paint factor
Anodes	193	-4.78	0.00	-12.90	
Float	**38896**	**-1.82**	**-0.19**	**-4.96**	in "transport" configuration (aligned with lower spar)
4C	36390	-1.82	-0.19	-4.96	from SW model; includes all pertinent SWBS and 5% margin plus 3% weld/mill allowance
Paint	2115	-1.82	-0.19	-4.96	typ MARCO paint factor
Anodes	390	-1.82	-0.19	-4.96	

Lifting Pad Eyes

Lifting points were designed to facilitate assembly and shipping. The lifting pad eyes were designed in accordance with Reference 10. The lifting arrangement consists of two lifting pad eyes on the aft bulkhead of each pontoon (P/S). The lifting pad eyes were designed to the estimated lightship weight of the WEC without the permanent ballast (Table 10). A 1.5 dynamic load factor was assumed to allow lifts in sheltered waters.

Each pad eye is rated for 372 kips (169 t). Crosby 200 t Wide Body Shackles with 1-1/8" 6x37 IWRC EIPS Wire Rope are assumed to attach to each lifting eye vertical to world. (Note, the shackle is not required to be sized to support the dynamic load factor if the WLL has a minimum SF on breaking strength of 3.) The resulting angle of 22° from the pontoon axis trims the WEC slightly bow down (~4°) with respect to the WEC's cribbing orientation (see WEC Support and Tiedown (transportability and dry dock)). The same lifting eye and shackle may be used to lift each fully outfitted pontoon/spar/knee assembly, albeit at half the WLL. The notional lifting arrangement is depicted in the following figure.

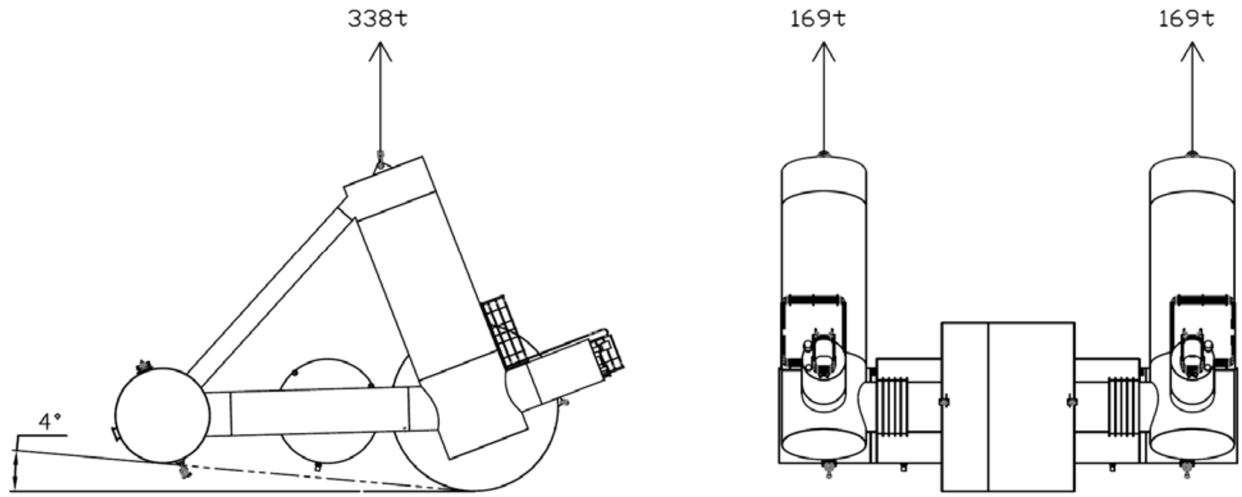

Figure 6: Lifting Arrangement

First principles were used to size the pad eye (see Addendum 15) and an FEA (see Addendum 26) was performed to verify the support structure. The longitudinal bulkhead ¾" inserts in way of the lifting eye were assumed to be NV AH36 steel and all other structure NV A. Results indicated the longitudinal bulkhead in way of the knee brace pipe backing structure was needed to run continuously through the pipe. FEA results are summarized in Appendix A.

WEC Support and Tiedown (transportability and dry dock)

The WEC may be lifted or rolled onto a deck cargo barge. The WEC is assumed to be moved and stowed (for shipping or dry docking) in approximately the towed orientation in order to facilitate deployment. Note, this results in all equipment and platforms being off axis by more than 60°. The lower point of ballast tank and nacelle cylinders is assumed to be 2 m off the ground/deck.

The blocking and seafastening arrangement was developed in accordance with the DNV Offshore Standard Sea Transport Operations (Reference 17). The DNV standard offers a simplified method to evaluate cargo accelerations experienced during ocean transport by barge. The barge transport design accelerations assume a nominal 275-foot barge operating in seas with a significant wave height of 6 meters or less. Weather routing is required to keep the WEC barge ocean transport within this operating limit and avoid storms.

Three load cases are evaluated representing either the worst-case roll, pitch, or combination of the two from quartering seas. The surge, sway and heave accelerations are determined for each case, and can be either in the positive or negative direction. The accelerations act at the stowed WEC center of gravity. The resulting inertial loads are resisted by the blocking, seafastenings, and lashings. Overturning moments are countered by variable blocking pressures. Lateral loads are absorbed by friction. Lateral loads that exceed the friction force are reacted by the seafastenings. The predicted accelerations do not tip the overall WEC, but there are localized uplift loads that are countered by the lashings.

Blocking and seafastenings are assumed to be capped with wood dunnage to evenly distribute loads into the WEC. The DNV standard limits the pressure applied to wood dunnage to 2 MPa. The highest blocking pressures occur when heave is positive which increases the effective vertical load. The maximum seafastening loads occur when heave is negative since that minimizes the effective vertical load and results in less of the lateral inertial load being absorbed by friction.

Seven (7) $1m^2$ blocks are to support the WEC: three (3) under the ballast, one (1) under each nacelle end bulkhead (P/S), and one (1) under each pontoon forward bulkhead (P/S). The float is assumed to be supported by two (2) $1m^2$ blocks. The WEC is to be lashed with 5/8" Grade 80 Alloy Chain at the lower spar-ballast tank joint, the nacelle tube-pontoon joint (P/S), and float arms (P/S). Seafastening is to be installed in way of the ballast tank and nacelle end bulkheads (P/S) to resist transverse loading and on forward and aft side of the ballast tank on CL (assuming barge coordinate system). Cribbing details are depicted in Figure 7.

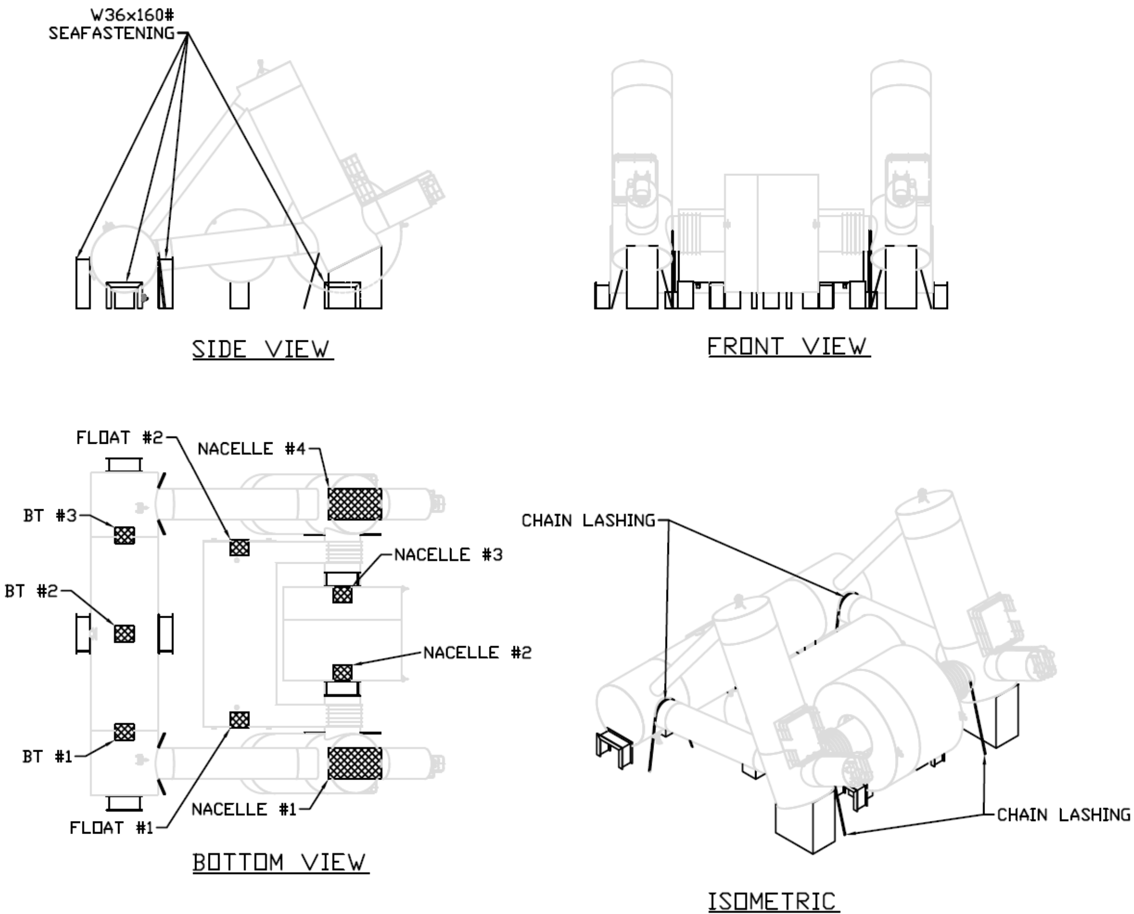

Figure 7: Seafastening and Blocking Arrangement

The WEC may be offloaded from the deck cargo barge by any of the following methods:

1. Dry dock
    a. Rolled off barge onto dry dock.
    b. Dry dock is submerged until WEC floats, then may be ballasted and towed to sea for deployment.
2. Submersible deck barge
    a. Deck cargo barge is submerged utilizing deck mounted variable ballast tanks until WEC floats, then may be ballasted and towed to sea for deployment.
3. Crane lift
    a. One (1) 400 t crane with spreader or two (2) 200 t cranes lifts WEC from deck cargo barge and lowers it in water. The WEC is then ballasted and towed to sea for deployment.

Ballast System and Control

Concept

The ballast systems on the WEC have two functions; controlling the attitude and stability of the WEC and enabling the auto-recovery of the float. The first system consists of fixed sand ballast with two variable seawater ballast tanks. The second system consists of vents and drains in the float which allow or prevent flooding as required by the situation.

The WEC is launched into the water without any ballast. At launch the WEC floats in the towing configuration with the ballast tank and nacelle at the waterline. Fixed ballast is added to the ballast tank decreasing the freeboard on the ballast tank. Variable ballast is then added in the ballast tank to rotate the WEC into its operating position where the ballast tank is completely submerged and the pontoons float at the waterline. The process is reversed at decommissioning.

The float is positioned forward of the nacelle in normal operation. In certain extreme wave conditions; the waves may push the float so that it rotates over the nacelle and is then floating between the pontoons. The float vents and drains are designed to allow the float to passively flood in the aft position. Once the float is submerged and after the extreme wave event has passed, the WEC PTO then applies torque to move the float back to the forward operating position where the float will gradually drain. After the float is drained it resumes normal operation.

The weights, stability, and ballast systems are further elaborated in this section. The hydrostatic calculations are attached in Addendum 30 through 32.

The weight estimate outlined in the Fabrication and Assembly section was used for purposes of the ballasting design. Changes in the weight estimate will require changes in the ballasting calculations.

Target Operating Displacement

Using the hydrostatic hull model based on the structural geometry, the WEC central body displaces the following mass and center of buoyancy when floating on the target design water line, at the origin (center of the nacelle) with zero trim:

	Weight (MT)	LCB (m)	TCB (m)
Displacement	754.06	3.649f	0.000

The hydrostatic model includes the WEC central body. The effect of the float is accounted for in actual loading conditions by a load imposed at the arm bearings. The origin of the hydrostatic model corresponds to the nacelle tube axis on centerline. The longitudinal axis in the hydrostatic model defines the forward direction from the nacelle center towards the pontoons. Vertical dimensions are measured positive above the baseline. The hydrostatic model is depicted in Figure 8 to Figure 10.

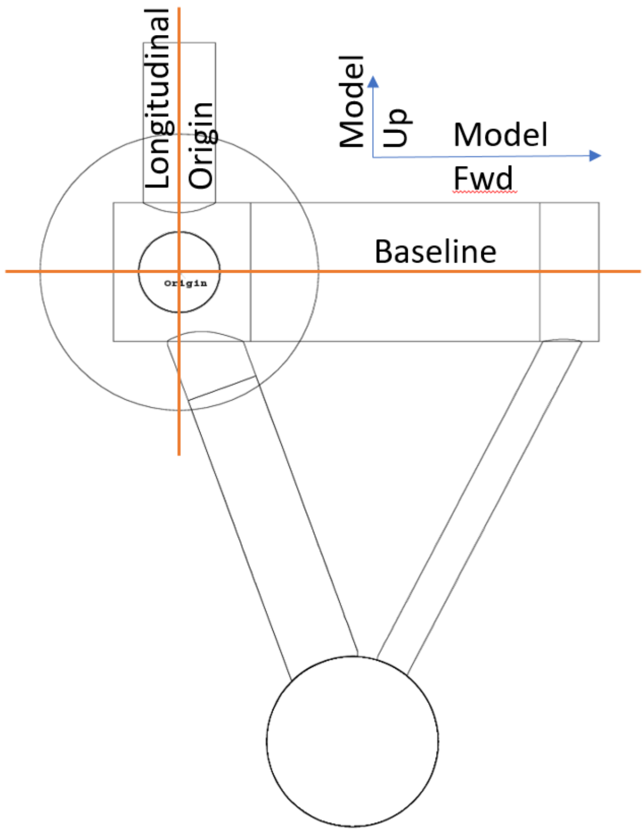

Figure 8: Hydrostatic Model – Profile View

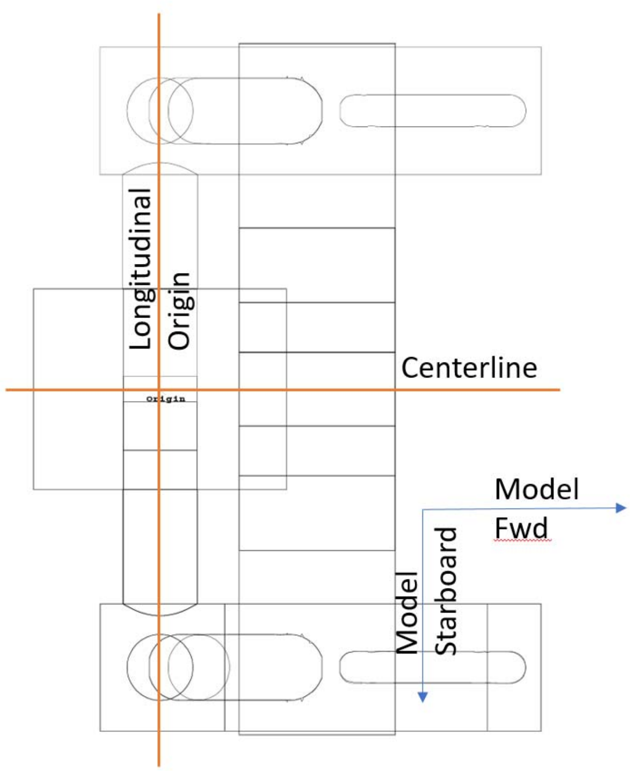

Figure 9: Hydrostatic Model – Plan View

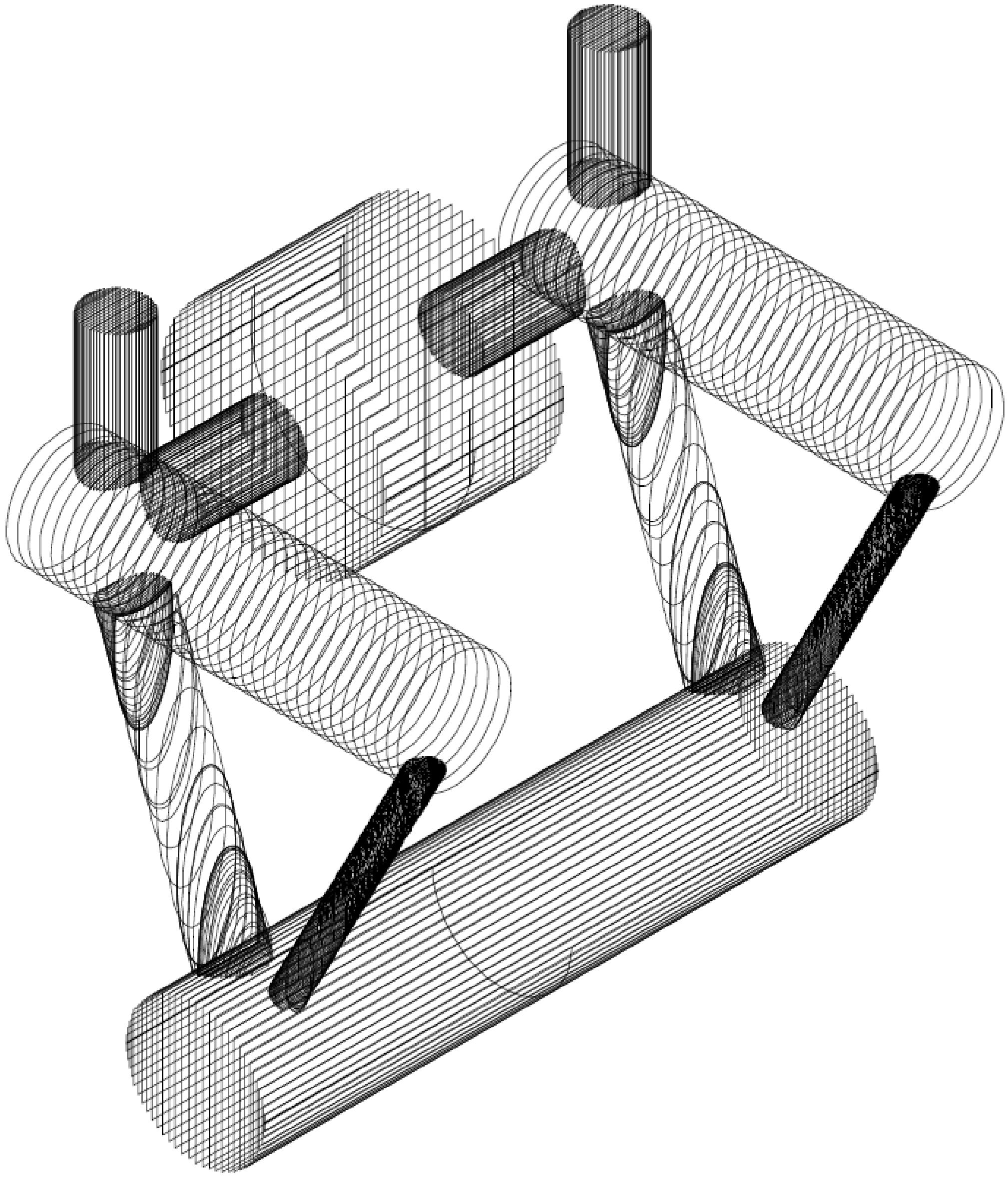

Figure 10: Hydrostatic Model - Isometric

Fixed Sand Ballast

The ballast tube is divided into seven tanks. The center five tanks are reserved for the permanent ballast while the outboard tanks are reserved for the variable seawater ballast. The permanent ballast, sand, tanks are sized such that:

- The ballast load takes no more than 80% of the tank volume to margin for future changes and easier loading.
- The tank end bulkheads land on one of the ring frame locations, 744mm spacing.
- The permanent ballast load results in a ballast tube freeboard of about 900mm with the WEC in the towing position and the SW ballast tanks empty.
- The seawater ballast tanks are sufficiently large to enable the ballasting evolution.

Fresh water and sand is used for permanent ballast. The permanent ballast is installed at WEC launch, through the manholes provided in the ballast tube. The sand tanks should be filled 100%

with fresh water before sand loading. Load the sand by pouring into the sand tank allowing the fresh water to overflow. Sand level is determined by monitoring ballast tube freeboard and heel angle. The sand is loaded asymmetrically between port and starboard compartments to achieve zero heel. Wet sand with a specific gravity of 1.92 was used in these calculations.

A total permanent ballast weight of 249.194 MT results in a ballast tube freeboard of 900mm. The amount of sand in each compartment varies between 43-73% of each space with the remaining permanent ballast volume filled by fresh water.

Variable Water Ballast

With the sand weight above, it takes 197.82 MT sea water to ballast the WEC to its design displacement. The ballast weight will be split into the two ballast tanks at 98.91 MT. This takes 82% of the SW ballast tank capacity.

Intact stability - Working

The working weight and centers with the above ballasting are:

	Weight (MT)	LCG (m)	TCG (m)	VCG (m)
Displacement	754.26	3.813f	-0.001	-9.123

When ballasted to the working orientation the WEC has a trim of 1.80 degrees, pontoon end down, and zero list. The trim cannot be corrected by changing the ballast weight. To adjust trim, corrector weights would be needed or alternatively, the geometry could be modified to move the ballast tube closer to the Nacelle. Neither strategy should be attempted without a detailed weight estimate including all equipment and outfitting.

The metacenter is a common point at which the righting force passes through as the vessel is inclined. The height of the metacenter above the center of gravity is known as GM and is a measure of hydrostatic stability. The definition is further elaborated in the damage stability section. In the working orientation the WEC has a GM of 5.135 m longitudinally and 12.107 m transversely. In the working orientation the VCB is 7.412 m below the origin. This is 1.711 m above the VCG therefore the WEC is stable.

Intact stability - Towing

The towing weight and centers with the above wet sand ballasting are:

	Weight (MT)	LCG (m)	TCG (m)	VCG (m)
Displacement	556.47	3.391f	0.002	-7.743

In the towing orientation at the WEC has a trim of 63.48 degrees, pontoon end up, and zero list.

In the towing orientation the WEC has a positive roll stability, maximum at 50 degrees, with a range exceeding 60 degrees. The WEC has positive pitch stability, maximum at 8 degrees, with a range exceeding 30 degrees. Therefore, the WEC is stable in the towing orientation.

Damage stability

The WEC is subdivided into the following water tight compartments:

- Nacelle
- Nacelle tube, fwd pontoon, top of spar, port and starboard
- Center pontoon, port and starboard
- Aft pontoon and Brace, port and starboard
- SW ballast tanks, port and starboard
- Sand permanent ballast tank
- Float

In the working orientation, any single compartment may be holed and flooded without the WEC sinking or becoming unstable. The damage stability calculations are attached in Addendum 32.

The metacenter is the point about which the center of buoyancy of the hull rotates, at small angles. GMt, the metrcentric height is the distance between the CG and the metacenter. It is also the slope of the righting arm curve at equilibrium. A positive number indicate the hull will return to equilibrium and the magnitude is a measure of roll stiffness. GMl works similarly for pitch stability.

The maximum, fully submerged, displacement of the WEC is 1092.68MT. The reserve buoyancy is the fraction of this that remains above the waterplane for each damage condition.

The following table summarizes the WEC stability assuming flooding a single compartment. The permanent ballast tank is not considered since it is already full in the operational condition.

Flooded compartment	GMt (m)	GMl (m)	heel (deg)	trim (deg)	Displacement (w/ added wt)	Reserve buoyancy
Intact	12.107	5.135	0.0	1.80	754.26	45%
SW ballast	14.064	6.101	+/-1.10	1.56	775.57	41%
Brace	11.543	4.679	+/-1.10	3.88	775.19	41%
Pontoon	7.176	3.965	+/-5.52	6.71	842.57	30%
Nacelle Tube, Fwd Pontoon, & Top Spar, Port	10.090	4.790	-2.20	-0.25	802.65	36%
Nacelle Tube, Fwd Pontoon, & Top Spar, Stbd	10.029	4.708	+2.31	-0.60	810.69	35%
Nacelle	9.222	3.250	+0.05	-7.97	951.56	15%

Ballast System description

The salt water ballasting system consists of two systems, completely independent one for the port ballast tank and one for the starboard. Each system consists of 4 pipes leading down from the upper spar deck to the ballast tank bellow:

- sounding tube
- tank vent
- tank fill/discharge
- stripping pipe

The sounding tube is a 1-1/2" pipe with a bronze flush plug at the upper spar deck leading down to a striker plate at the ballast tanks lowest point. The bottom of the sounding tube is in the lowest point in the working orientation.

The tank vent is an 8" pipe starting at a flanged connection above the upper spar deck leading down to the ballast tank top located in the lower spar. A flanged float check valve is provided for working operations and the ballasting operation. For de-ballasting an air manifold is provided which is bolted to the flanged connection. The air manifold is fabricated from 2" pipe and is provided with an air connection, a pressure regulator, 150 psig-40 psig, and a safety valve rated at 45 psi.

The tank fill/discharge is a 6" pipe starting at a flanged connection above the upper spar deck leading down to the lowest point in the ballast tank to a rose box.

The stripping pipe is a 2" pipe starting at a flanged connection above the upper spar deck leading down to the lowest point in the ballast tank to a rose box. The stripping line terminates inside the tank at the lowest point in the towing orientation.

The salt water ballast system is depicted in Addendum 5 and Figure 11 and Figure 12.

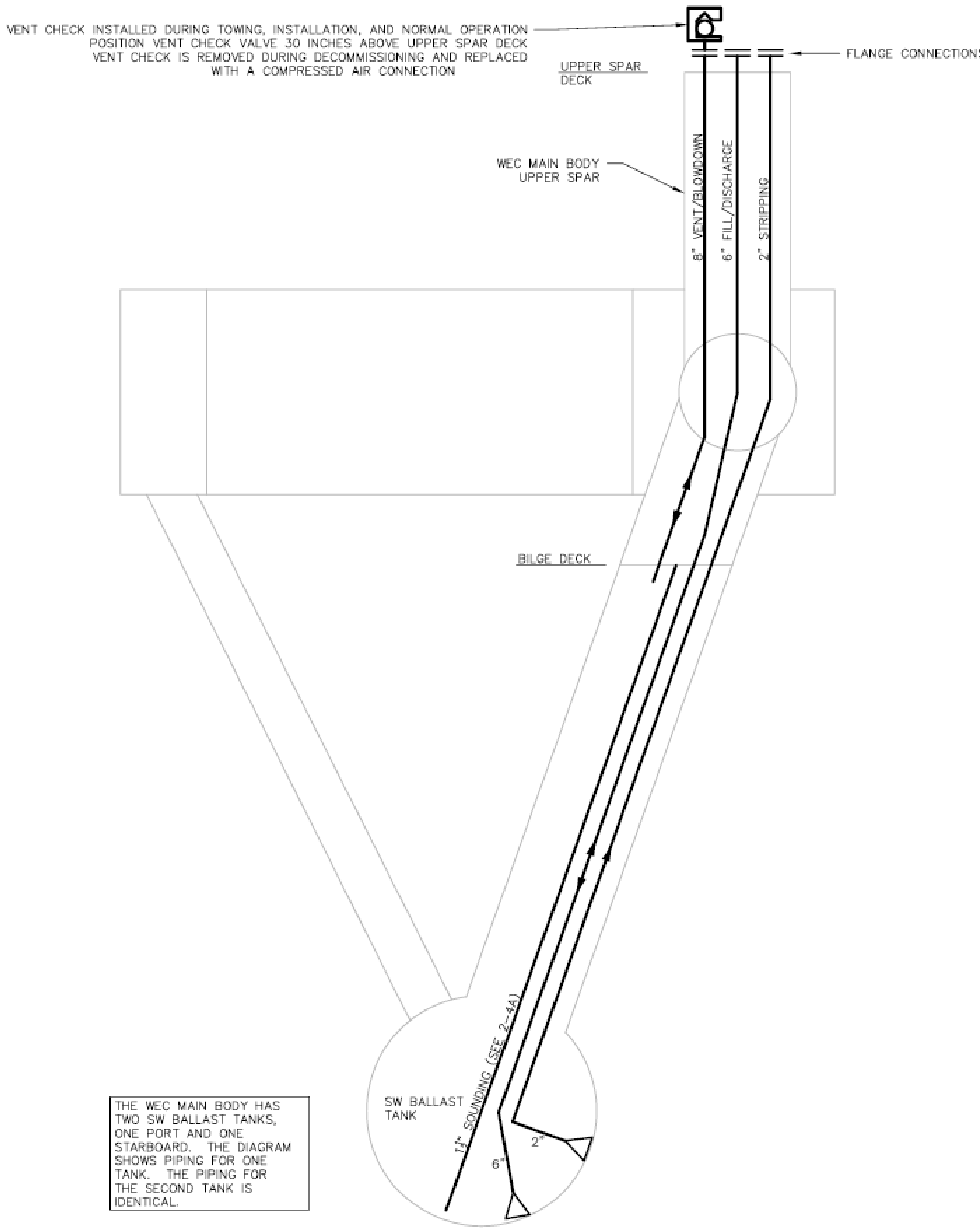

Figure 11: Salt Water Ballast System

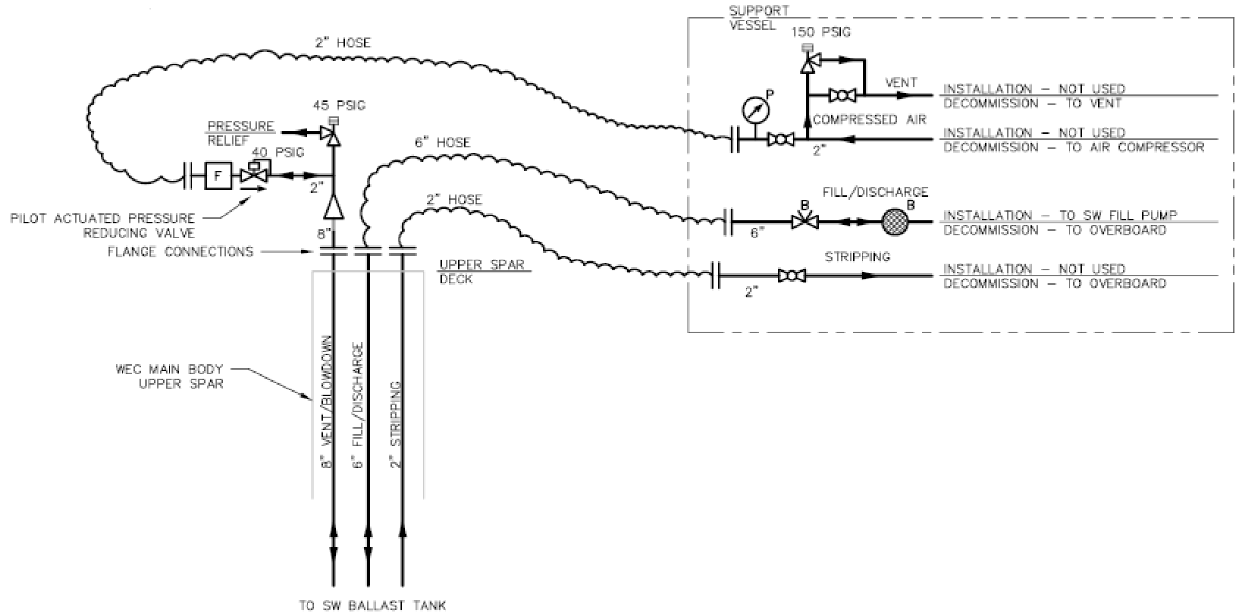

Figure 12: Ballast System Installation/Decommissioning Connections

Ballasting Sequence

- Carefully track the ballast tank freeboard and heel during loading of sand. Load only to the specified freeboard as it will be impossible to adjust after deployment.
- Fill sand tanks with Fresh Water.
- Load sand into sand tank through man holes letting water overflow. This may be accomplished in calm water with the WEC afloat. Maintain zero list and adequate freeboard by varying the sand load among the sand tanks. Securely seal the manholes.
- Install float checks on vent pipes if not already installed.
- On site, attach 6" lay-flat hoses to the flanges on the fill/discharge connections. Lead other ends of the hoses to the support vessel and to pump(s). Pumps must have the ability to pump 500 GPM at 30' head.
- Pump water until WEC rotates into working orientation. The total operation is expected to take approximately one hour with rotation in 10 minutes. Ensure symmetrical loading into the two ballast tanks by maintaining near zero list. Start/stop the pumps as required.
- After rotation, board WEC and use tank sounding or draft mark monitoring for final adjustment of the draft.
- Remove hoses and blank off flange connections.

De-ballasting Sequence

- Remove float checks from vent pipes. Install air manifolds.
- Run air hoses from support vessel compressor to manifolds. Compressor needs to produce 120 CFM at 125 psig.
- Attach 6" lay-flat hoses to the flanges on the fill/discharge connections. Lead other ends of the hoses to the support vessel. Add control valves, 6" butterflys, in an accessible location on support vessel and lead discharge hoses overboard. Both control valves should be in the same general location.

- Attach 2” hoses to the flanges on the stripping pipe connections. Lead other ends of the hoses to the support vessel. Add control valves, 2” ball valves, in an accessible location on support vessel and lead hoses overboard. Both control valves should be in the same general location as the 6” valves.
- Open fill/discharge valves and close the stripping valves.
- Start the air compressor and begin pressurizing the tanks.
- Monitor and control the WEC heel by actuating the 6” valves.
- The entire operation should take an hour. At about 50 minutes, the WEC will rotate into towing orientation. This happens when the ballast tanks are at the 15% full level.
- Continue blowing the tanks until air escapes from the discharge hoses. This may occur before the WEC rotates into towing orientation..
- Close the discharge valves and open the stripping valves.
- Continue blowing the tanks until air escapes from the stripping hoses.
- Stop compressor.
- Remove all hoses and air manifolds, replace float checks and blank off hose flange connections.

Float passive recovery

The float is designed to operate forward of the Nacelle, facing the waves. In large waves the float may find itself flipped over the Nacelle and oriented away from the waves. In this position, it is designed to self-flood to an almost neutral buoyancy and to be driven under the Nacelle by the PTO back into the forward position where it will self-drain and continue operating.

To achieve this, the float is constructed with upper and lower chambers separated by a “deck”. The upper chamber is designed to flood and the lower chamber is a non-flooding void. To drain effectively the deck must have adequate freeboard and be parallel to the waterplane. The distance of the deck above the water, the “freeboard”, should be maximized to promote drainage but is limited by the available PTO torque required to drive the float underwater.

Using a design torque of 500kNm, the resulting float freeboard is minimal and requires that the float arms are free flooding. The internal volume of the float arms is common with the float chamber.

The float will drain through 4 wafer spring check valves, one at each deck corner. Two float checks are provided to permit air inflow while preventing water inflow if submerged. The float arms are hollow and will have access openings, vents, and drains inside the float chamber. Additionally, a drain will be located on the underside at the bearing end of each arm.

When the float is inverted, the float checks will allow water to enter the “upper” chamber, the flapper checks will allow air to vent out, the arms will flood via the openings to the float, and the arms will vent through the “arm drains”.

With the upper chamber full the float will be awash, allowing the PTO to push it underwater to the vertically down position. At that point, the float will rise to the surface on its own or the PTO may continue to rotate it. Once at the surface, the buoyant lower chamber will gradually force the float up as the water drains out. The float checks and the flapper checks are provided to speed this process by not allowing wave action to partially refill the chamber. It is also possible for the PTO to continue driving the float upward, to the torque limit, which will speed drainage by increasing the head.

The float self recovery system is a very fine balance between the float weight and the float geometry. The estimated weight of the float is determined from the design scantlings.

The float scantlings result in a float weight with margins of 38,892 kg. The float has a deck height of 1219mm (48") above the bottom. This maxes out the 500kNm PTO torque to push the float below the surface. After recovery, and in operation, the deck will have 97mm (3.83") of freeboard. It may be lifted to 309mm (12.17") of freeboard to promote drainage with the PTO at max design torque, 500kNm.

The low freeboards in the normal operating condition could potentially result in poor float drainage. Additionally, marine fouling or other effects could prevent the check valves from sealing properly leading to ingestion of water inside the float during normal operating conditions. If the float drain system proves problematic, the check valves can be removed, and blind flanges installed on all drains. The blind flanges would not allow the float to drain automatically but would preserve the buoyant envelope of the float. The float position could still be manipulated by using the flange connections to attach hoses from a service vessel for compressed air or water as needed.

Flange connections on the float lower buoyancy chamber allow the float position to be manipulated if the PTO is offline and not available to apply torque. The flanges allow hose connections for water and compressed air from a service vessel. The upper and lower chambers are filled with water to submerge the float. The float position can be manipulated using lines and a service vessel. Then compressed air is used to empty the float.

Fatigue Assessment

This section describes the fatigue assessment for the prototype WEC structure carried out in accordance with guidance in Reference 6. Table 11 contains hot spot stresses calculated from detailed finite element analysis results according to Reference 6 / Section 4.2 for the three tubular joint models. Table 11 lists the calculated fatigue life for the tubular joint details calculated as a Miner's sum, assuming a Weibull shape factor of 1.0 and an S-N curve for tubular joints (Ref. 6/ 2.4.6) in seawater with cathodic protection.

The maximum stress range assumes that the hot spot stresses are fully reversing. The characteristic stresses reflect the 50-year return period of the extreme sea state in Reference 14. The average zero-upcrossing period was calculated for the WETS site from the wave scatter diagram in Reference 14 and an assumed wave period ratio (Tz/Te) of 0.71. The calculations assume that the wave zero-upcrossing period is representative of the load and stress response period. Addendum 19 contains the fatigue life calculations.

The current design does not achieve a 5-year fatigue life based on this conservative, simplified fatigue life estimate. The longitudinal bulkheads in way of the joints were inserted locally with thicker plate to achieve a 5-year fatigue life. Increasing plate thickness to reduce stress does not improve fatigue life in all cases, because allowable stress reduces as the thickness increases. A more refined fatigue analysis and/or structural modifications may be necessary to demonstrate a 5-year design life. Table 11 also provides the maximum allowable stress for a 5-year fatigue life for use in evaluating potential structural design modifications.

Table 11: Fatigue Life

Group	Joint	Connection	Location	Membrane stress [MPa]	Bending stress [MPa]	Hot spot stress [MPa]	Fatigue Life [Yr]	Allowable Stress [MPa]
1	J1	BT-LS	Brace	151	84	235	5	229
2	J1	BT-LS	Chord at crown	130	58	188	12	
3	J1	BT-LS	Chord at saddle	188	62	250	4	235
4	J1	BT-KB	Brace	140	90	230	4	219
5	J1	BT-KB	Chord at crown	153	42	195	10	
6	J1	BT-KB	Chord at saddle	172	125	297	2	235
7	J2	P-KB	Brace	146	104	249	2	203
8	J2	P-KB	Chord at crown	69	163	232	7	
9	J2	P-KB	Chord at saddle	45	68	114	151	
10	J2	P-KB	Long'l bhd*	186	3	189	6 *	
11	J3	P-NT	Brace	78	79	157	14	
12	J3	P-NT	Chord at crown	47	11	58	1456	
13	J3	P-NT	Chord at saddle	122	162	284	2	203
14	J3	P-US	Brace	131	127	258	4	242
15	J3	P-US	Chord at crown	39	29	68	664	
16	J3	P-US	Chord at saddle	111	50	161	12	
17	J3	P-LS	Brace	98	133	231	3	203
18	J3	P-LS	Chord at crown	114	52	167	11	
19	J3	P-LS	Chord at saddle	101	60	161	13	
20	J3	NT-US	Brace	207	85	291	3	242
21	J3	NT-US	Chord at crown	122	14	136	26	
22	J3	NT-US	Chord at saddle	179	92	271	2	203
23	J3	NT-LS	Brace	189	106	295	1	203
24	J3	NT-LS	Chord at crown	86	15	101	100	
25	J3	NT-LS	Chord at saddle	167	13	181	8	
26	J3	NT-LS	Long'l bhd*	182	3	185	5 *	

Notes: Connection defined as Chord-Brace with the following abbrevations:

BT = Ballast Tank KB = Knee Brace NT = Nacelle Tube

LS = Lower Spar P = Pontoon US = Upper Spar

* Longitudinal bulkheads inserted locally in way of the joint. FEA membrane stress scaled down linearly with plate thickness. Bending stress scaled quadratically.

Group 10 insert 1/2" plate

Group 26 insert 3/4" plate

Improved welds offer an alternate approach to increasing fatigue life. DNV allows the use of weld profiling, but discourages weld toe grinding or hammer peening at the design stage. Table 12 shows that the minimum calculated fatigue life of the WEC increases to five years if weld profiling is applied. Full penetration welds at the joints will be required for this approach. The labor cost for building the WEC will increase significantly due to weld profiling; however, we are not able to estimate the increase at this time.

Table 12: Fatigue life with weld profiling

Group	Joint	Connection	Location	Membrane stress [MPa]	Bending stress [MPa]	Hot spot stress [MPa]	Profiled stress [MPa]	Fatigue Life [Yr]	
1	J1	BT-LS	Brace	151	84	235	173	15	**
2	J1	BT-LS	Chord at crown	130	58	188	138	43	
3	J1	BT-LS	Chord at saddle	188	62	250	182	13	**
4	J1	BT-KB	Brace	140	90	230	170	14	**
5	J1	BT-KB	Chord at crown	153	42	195	142	38	
6	J1	BT-KB	Chord at saddle	172	125	297	220	6	**
7	J2	P-KB	Brace	146	104	249	185	9	**
8	J2	P-KB	Chord at crown	69	163	232	177	17	
9	J2	P-KB	Chord at saddle	45	68	114	86	501	
10	J2	P-KB	Long'l bhd*	186	3	189	135	22	*
11	J3	P-NT	Brace	78	79	157	117	61	
12	J3	P-NT	Chord at crown	47	11	58	42	9205	
13	J3	P-NT	Chord at saddle	122	162	284	213	5	**
14	J3	P-US	Brace	131	127	258	193	11	**
15	J3	P-US	Chord at crown	39	29	68	51	3701	
16	J3	P-US	Chord at saddle	111	50	161	118	59	
17	J3	P-LS	Brace	98	133	231	174	11	**
18	J3	P-LS	Chord at crown	114	52	167	122	50	
19	J3	P-LS	Chord at saddle	101	60	161	119	58	
20	J3	NT-US	Brace	207	85	291	213	7	**
21	J3	NT-US	Chord at crown	122	14	136	97	145	
22	J3	NT-US	Chord at saddle	179	92	271	199	6	**
23	J3	NT-LS	Brace	189	106	295	218	5	**
24	J3	NT-LS	Chord at crown	86	15	101	73	614	
25	J3	NT-LS	Chord at saddle	167	13	181	129	39	
26	J3	NT-LS	Long'l bhd*	182	3	185	132	18	*

Notes: Connection defined as Chord-Brace with the following abbrevations:

BT = Ballast Tank   KB = Knee Brace   NT = Nacelle Tube

LS = Lower Spar   P = Pontoon   US = Upper Spar

* Longitudinal bulkheads inserted locally in way of the joint.

**** Weld profiling required to achieve 5-yr fatigue life.**

Cost Estimate

This section contains narrative describing the cost estimates for both prototype construction and subsequent production at higher volume.

Prototype Construction

This sub-section describes the approach to developing a rough order engineering cost estimate for the WEC prototype construction. The cost reflects primarily WEC steel fabrication given information available at this time, with rough estimates for some of the expected outfitting. We estimate a total shipyard cost of $7.1 million, excluding class survey, the delivery transit, installation, testing, and commissioning. Table 13 provides a breakdown of the costs by percentage. Addendum 19 provides complete details of the estimate.

Table 13: Cost breakdown

Shipyard Engineering & Services	18%
Structure - WEC	59%
Electric Plant (Provided and installed by CPower)	0%
Command and Surveillance	0%
Auxiliary Systems	10%
Outfit & Furnishings - WEC	13%

The shipyard engineering and services estimate percentage is high due to the lack of design definition described further below.

The estimate in Addendum 19 is organized by SWBS (Ship Work Breakdown Structure) number, a common scheme in the marine industry. The estimate includes the cost to fabricate the WEC structure and ballast system based on current design information for these items. A number of design elements lack sufficient detail to estimate cost with any confidence, including outfit, auxiliary systems, and command and surveillance. The estimate assumes that the generating and electrical equipment is owner furnished and installed without yard support.

The steel cost estimate assumes the following:

Labor rate = $75 per hour

Steel fabrication rate = 150 hours per LT

Steel material cost = $0.754 per pound

The labor rate reflects construction of the prototype in the Pacific Northwest. The assumed material cost is based on a Seaport Steel quotation for AH36 steel in November 2017. Material costs have reportedly been volatile due to the current political climate.

The fabrication rate is based on data available for new steel construction augmented to reflect a rate somewhere between general construction and foundations. A higher fabrication rate is expected because a panel line cannot be used on the cylindrical structures and the end connection details for the cylinders approach the complexity of foundations.

Calculated steel costs use the weight estimate in the Fabrication and Assembly section.

The generator and associated electrical equipment are assumed to be owner furnished and installed. Command and surveillance systems, such as alarms and monitoring, are also assumed owner furnished and installed.

Auxiliary systems and outfitting are poorly defined at this point in design. Costs are not included for undefined systems (HVAC, lighting, lube oil, etc.); however, Addendum 20 includes high level placeholders for CPower to consider.

The cost estimate includes rough estimates for vents, bilge and ballast systems, and mooring fittings in Auxiliary Systems. Mooring hardware is assumed to be owner supplied. Installation of the sand ballast is included based on a quotation for a similar amount of high-density ballast in 2016.

The cost estimate includes rough estimates for outfitting, such as railings, ladders, grating, doors, hatches, and manholes based off the arrangement shown in the Solidworks model (Addendum 3); however, details of these outfitting items have not been developed. An estimate for blasting and painting both the interior and exterior of the WEC is included, assuming three coats and a total thickness of 10 mils. A paint specification is not available. Anodes for cathodic protection are also included.

Production

Production savings can derive from both design for production and high-volume manufacturing efficiencies. The wind energy industry provides an example of construction costs for more mature production methods. Wind turbine towers are constructed at a cost of about $1.50 per pound by eliminating ring frames and setting up production facilities to efficiently roll steel into large cylinders.

The ring frames in the prototype design can be eliminated by increasing shell thicknesses for the ballast tank, spars, nacelle, and pontoons. Table 14 presents estimated shell thicknesses and the associated weight increase for these elements. The ring frames in the float are not eliminated in this cost study, because they are considered technically necessary to minimize weight and insure functional operation of the float ballast system.

Table 14: Shell thickness without ring frames

Structural element	Shell thickness [in]	Weight Increase [t]	Cost Savings
Ballast tank	1.25	42	$194,000
Spars	0.6875	16	$130,000
Nacelle	0.75	10	$66,000
Pontoons	0.625	11	$222,000
Total	--	79	$612,000

Combining the steel cost savings with reducing shipyard engineering and services to 7% results in a rough-order, lower bound estimate of $5.4 million per unit for production. The cost per pound for the overall WEC structure is expected to be higher than the wind turbine tower cost because of the added complexity of the structural joints in the WEC. Therefore, achieving the total estimated savings is unlikely.

The design changes suggested in this section are for the notional cost estimate only. Feasibility of any design modifications would have to be checked from the standpoints of stability and power production. The weight increase will raise the center of gravity of the structure and reduce the amount of sand in the permanent ballast tank.

Conclusion

The design complies with the requirements of Reference 1 with one exception. Certain local structural details in the joints do not achieve a 5-year fatigue life when calculated according to the simplified fatigue life method. Weld profiling can be utilized to increase the calculated fatigue life to 5 years but will increase fabrication cost. A more refined fatigue analysis and/or structural modifications may be necessary to demonstrate a 5-year design life without weld profiling.

Appendix A Solid Works Simulation FEA Results

The following screenshots show basic finite element model geometry (including thickness), loads (forces/moments are in magenta and pressure is in red), boundary conditions (in green), and results. The results show elemental von Mises stress scaled to yield of the subject material. In general, rigid bodies are hidden for clarity.

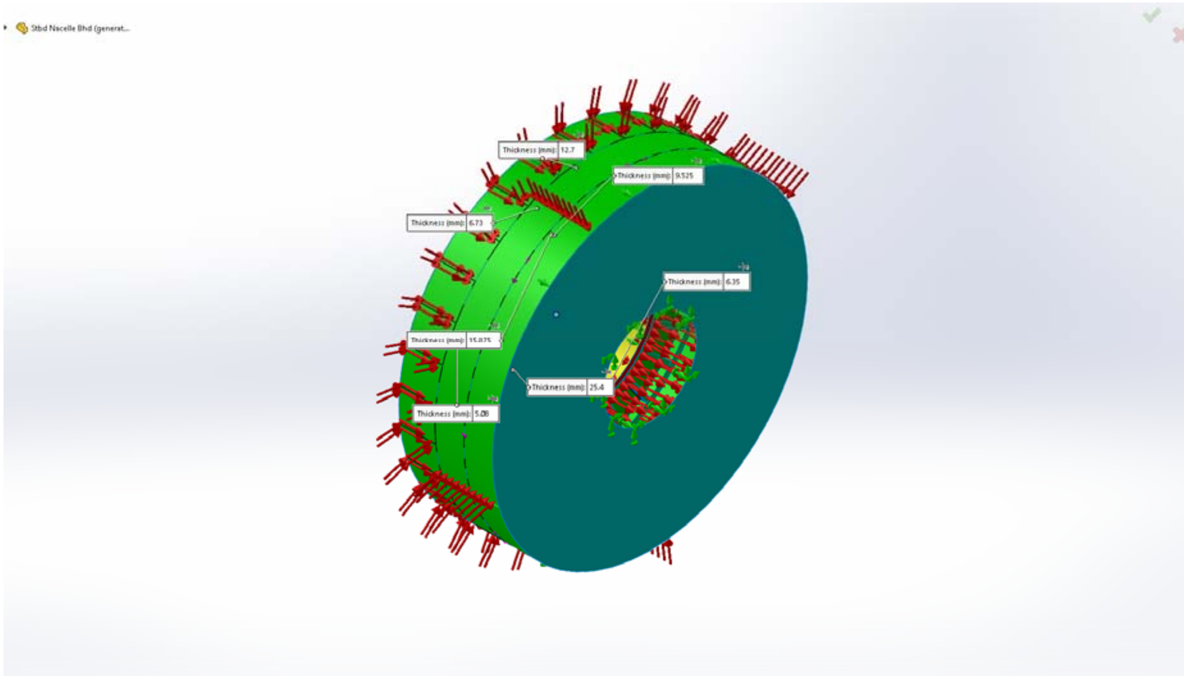

Figure A1. Starboard Nacelle Bulkhead with ‘centerline’ nontight bulkhead.

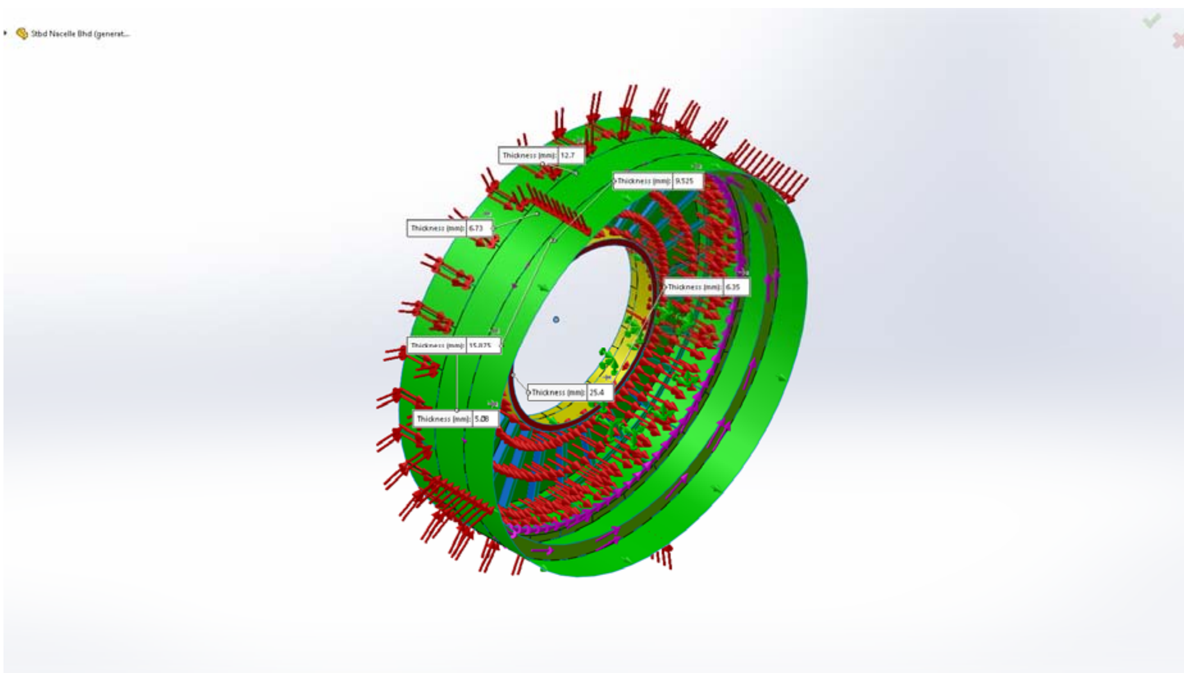

Figure A2. Starboard Nacelle Bulkhead (‘centerline’ nontight bulkhead removed for clarity).

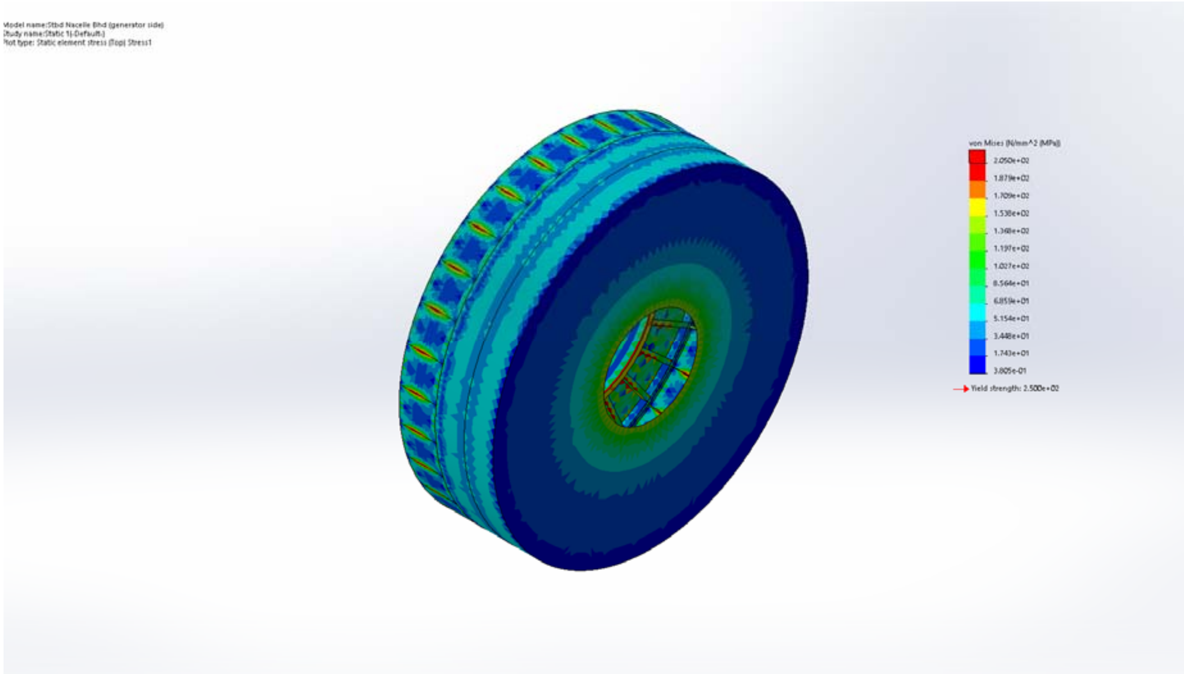

Figure A3. Von Mises stress of Starboard Nacelle Bulkhead with ‘centerline’ nontight bulkhead.

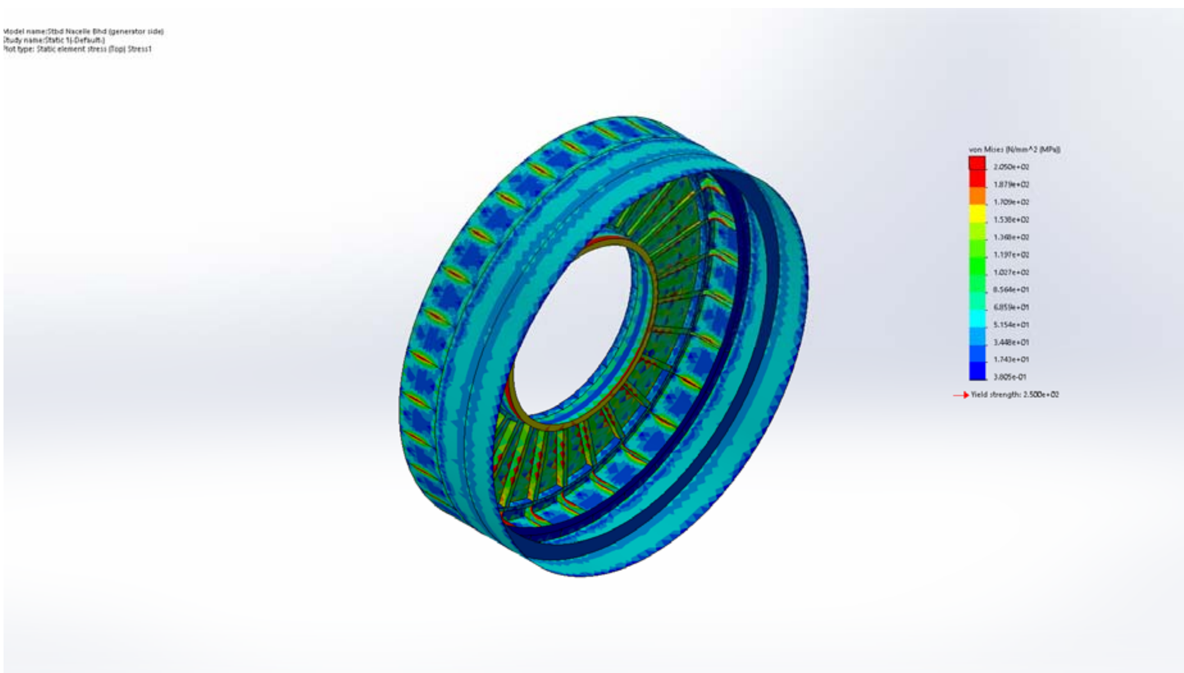

Figure A4. Von Mises stress of Starboard Nacelle Bulkhead (‘centerline’ nontight bulkhead removed for clarity). Note higher stress in way of radius of corner bracket. Final structural detail developed on port side that includes rider bar (see Figures A5-A6) was adopted.

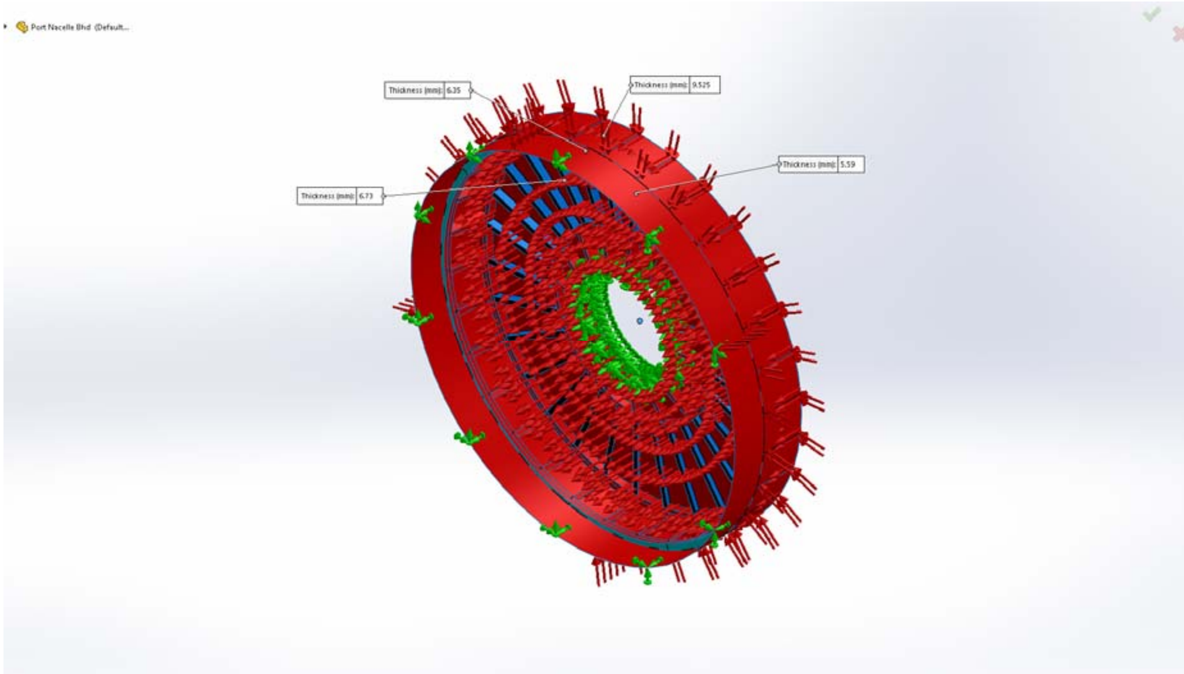

Figure A5. Port Nacelle Bulkhead.

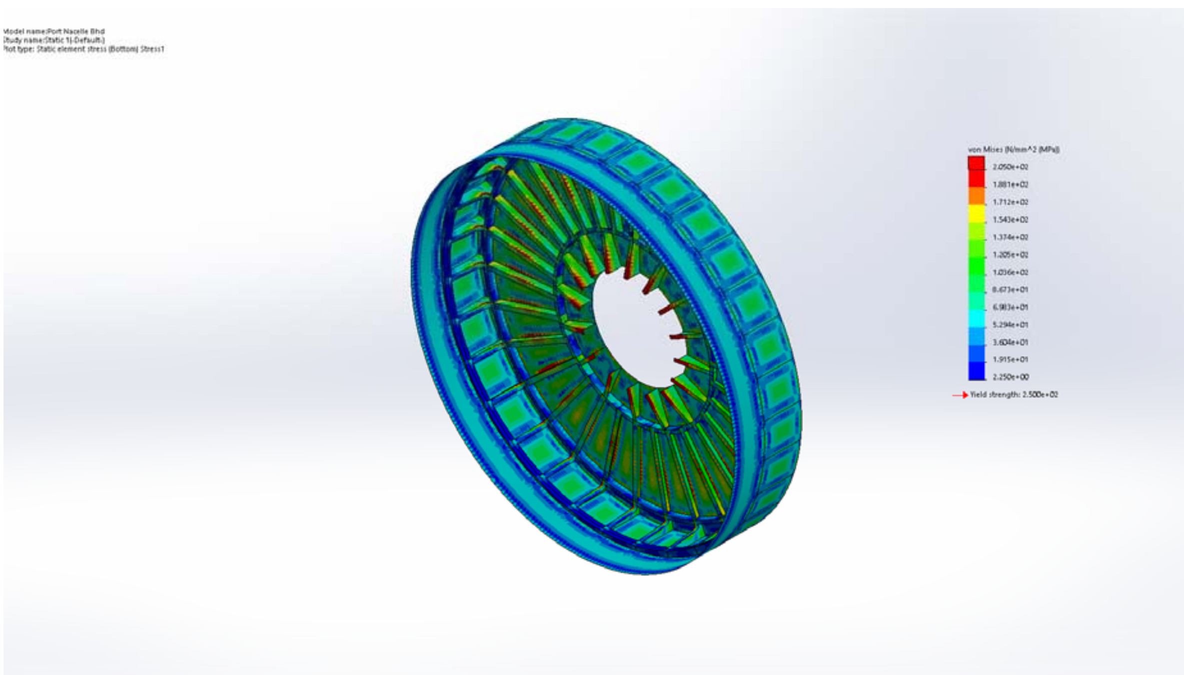

Figure A6. Von Mises stress of Port Nacelle Bulkhead.

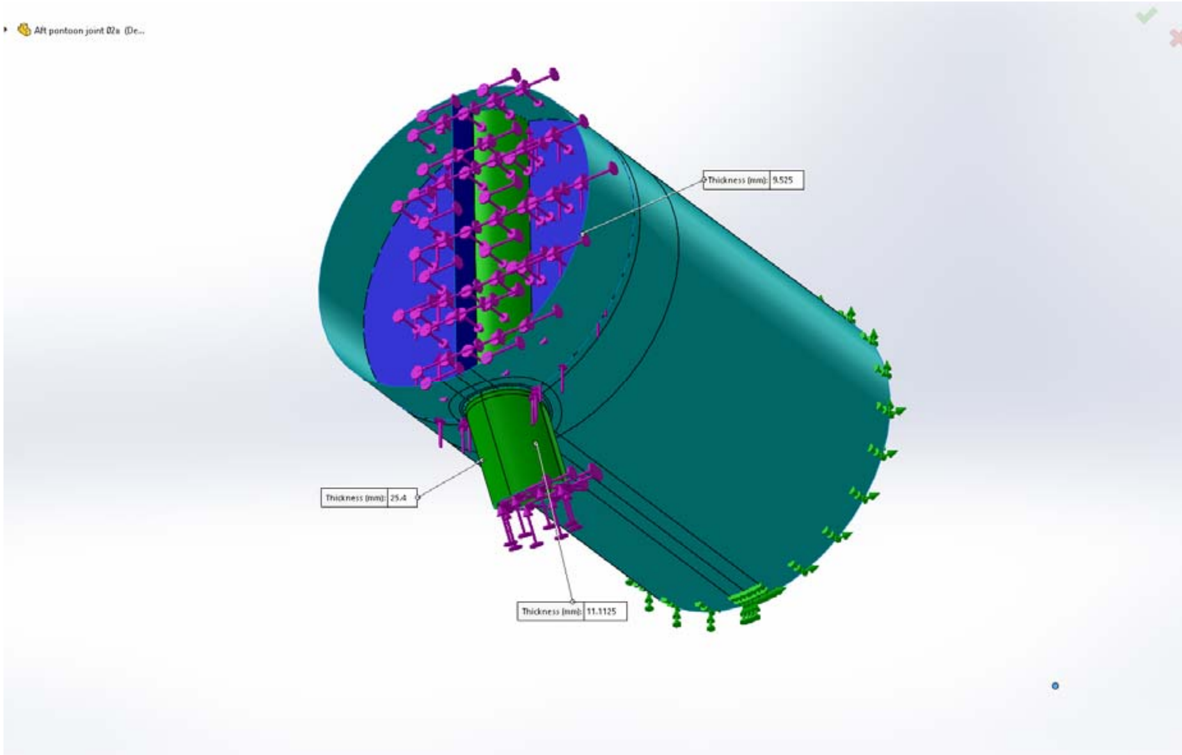

Figure A7. Aft Pontoon Joint (J2).

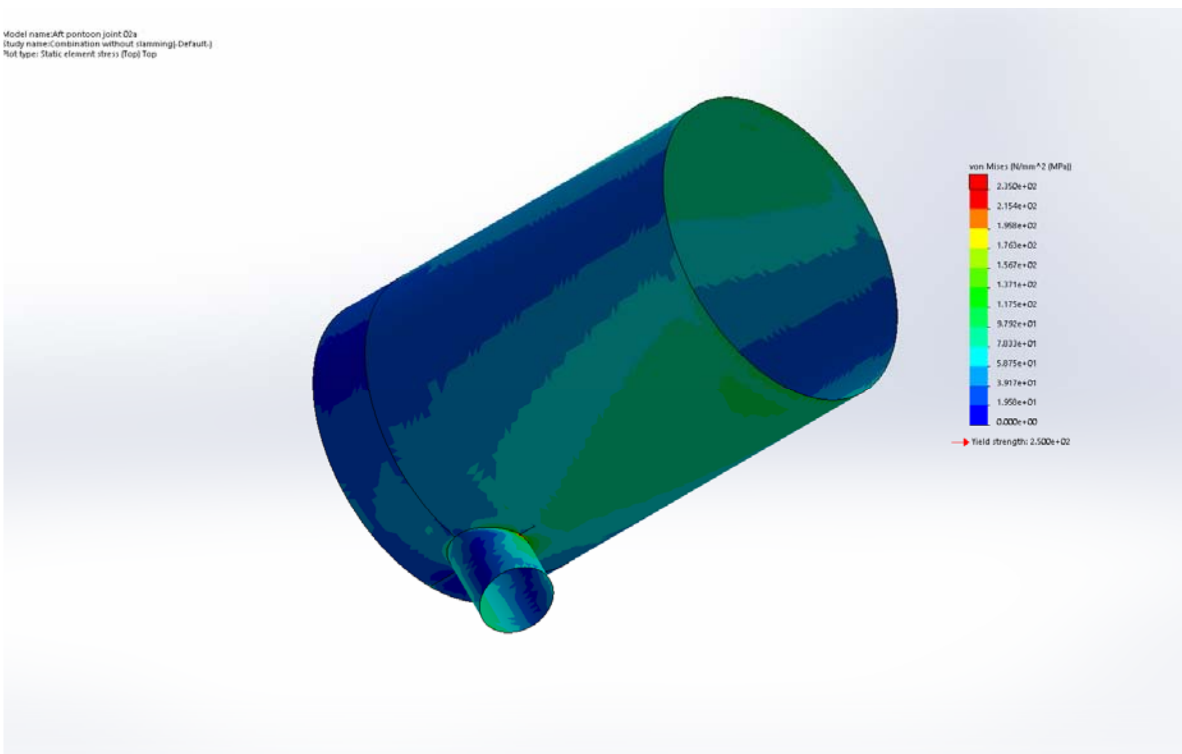

Figure A8. Von Mises stress Aft Pontoon Joint.

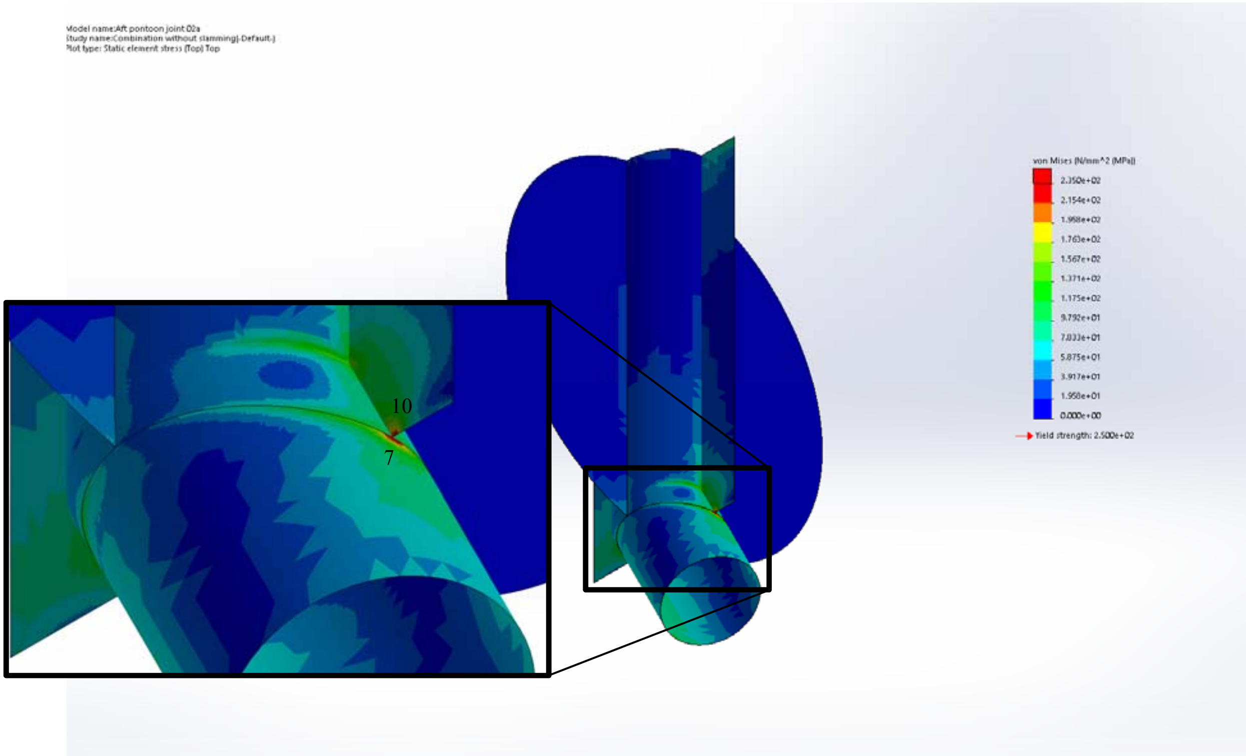

Figure A9. Von Mises stress Aft Pontoon Joint; shell removed for clarity; closeup of hot spot at Group 7 and 10.

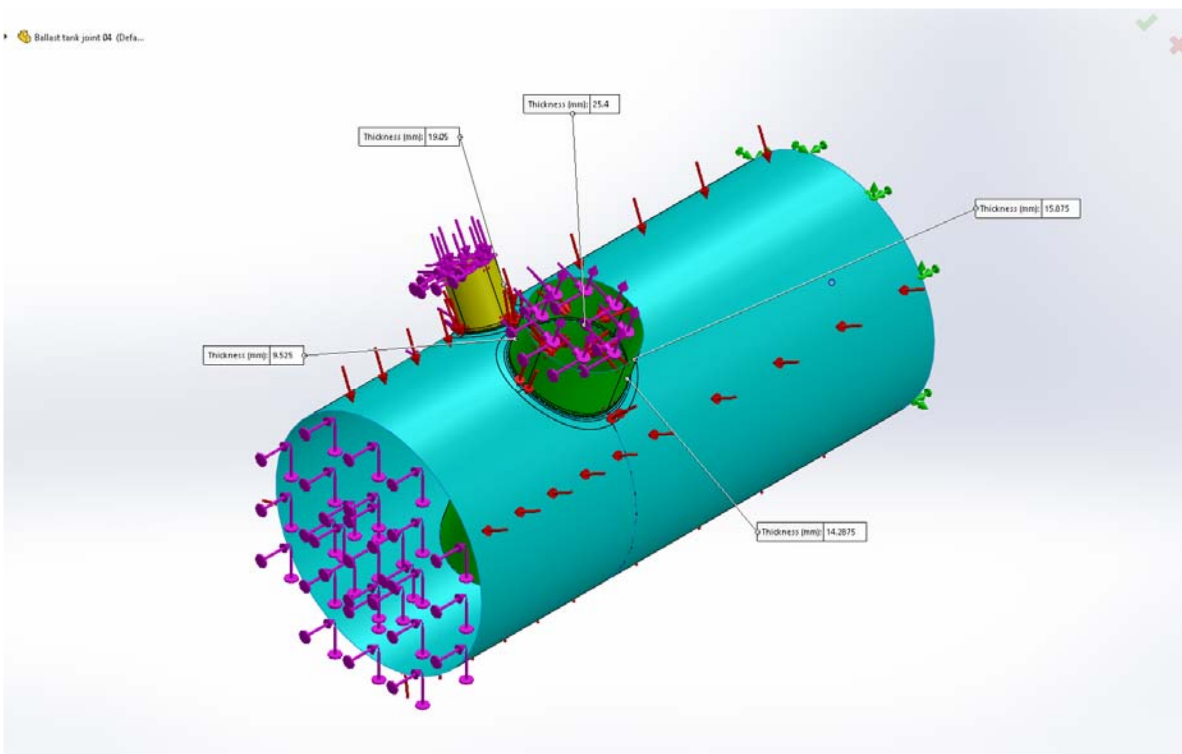

Figure A10. Ballast Tank Joint (J1).

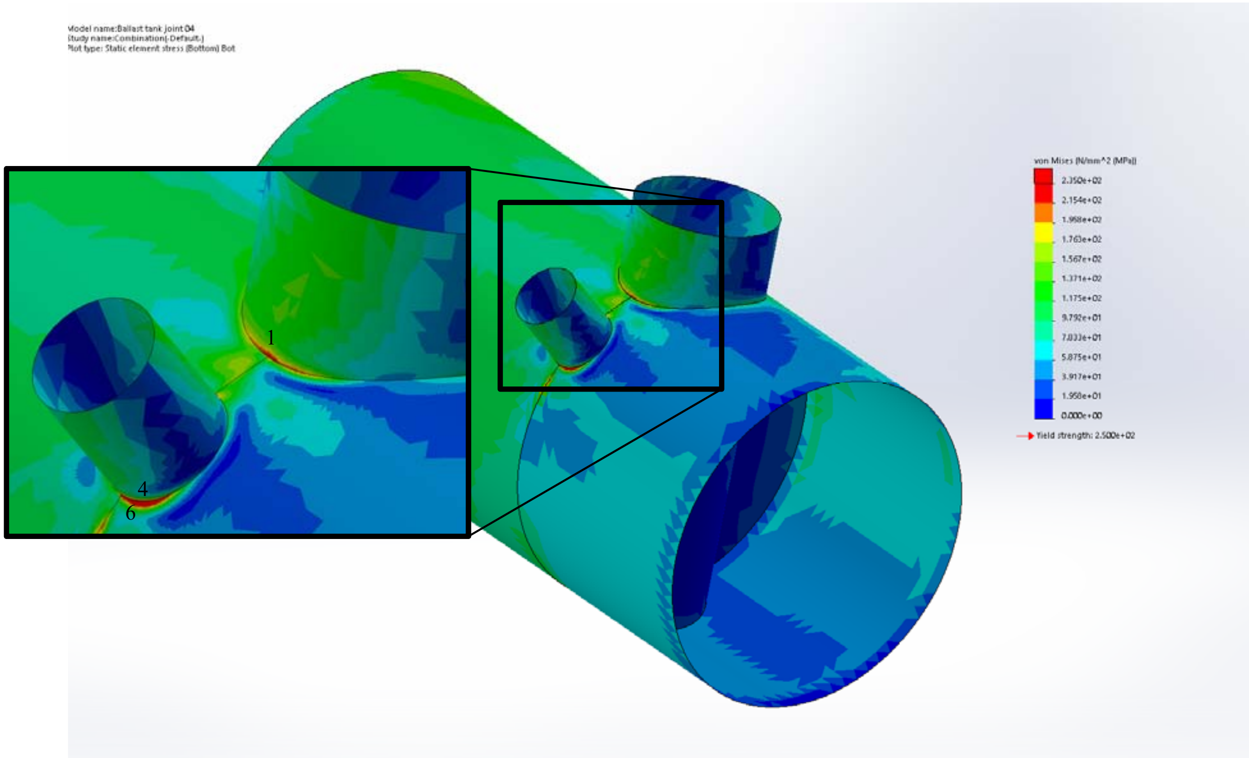

Figure A11a. Von Mises stress of Ballast Tank Joint; closeup of hot spot at Group 1, 4 and 6.

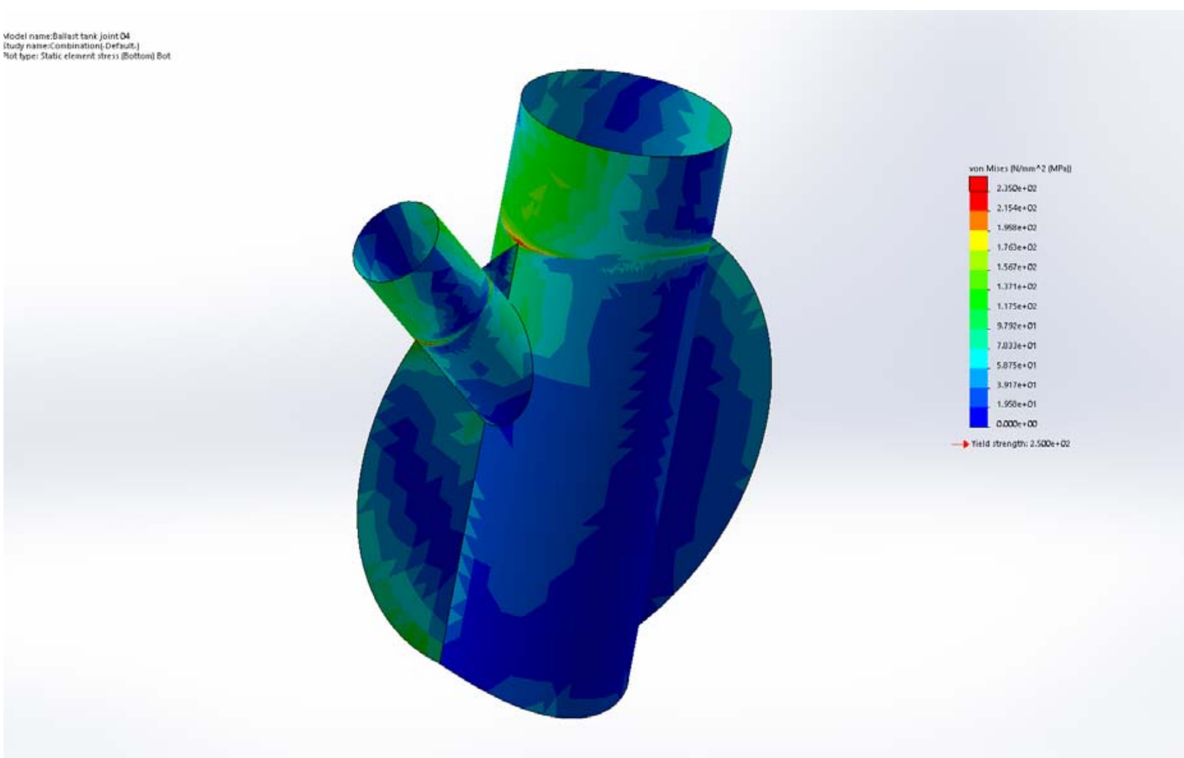

Figure A11b. Von Mises stress of Ballast Tank Joint; shell removed for clarity.

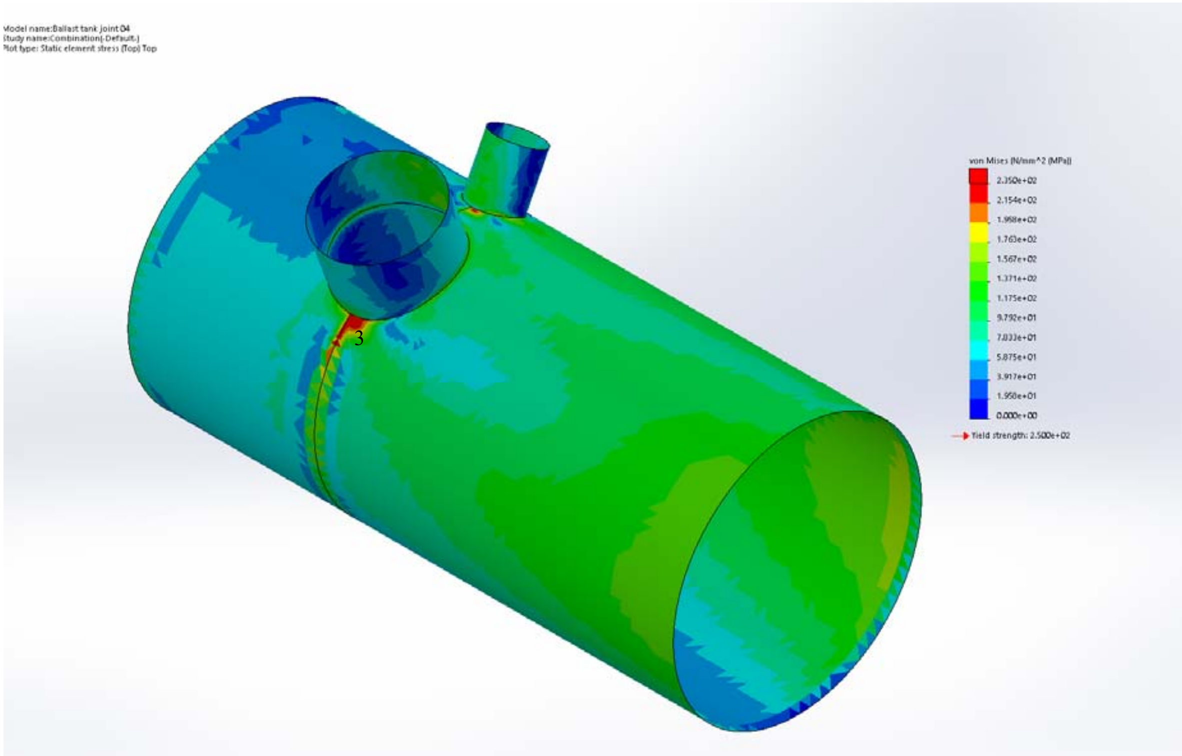

Figure A12. Von Mises stress of Ballast Tank Joint showing hot spot at Group 3.

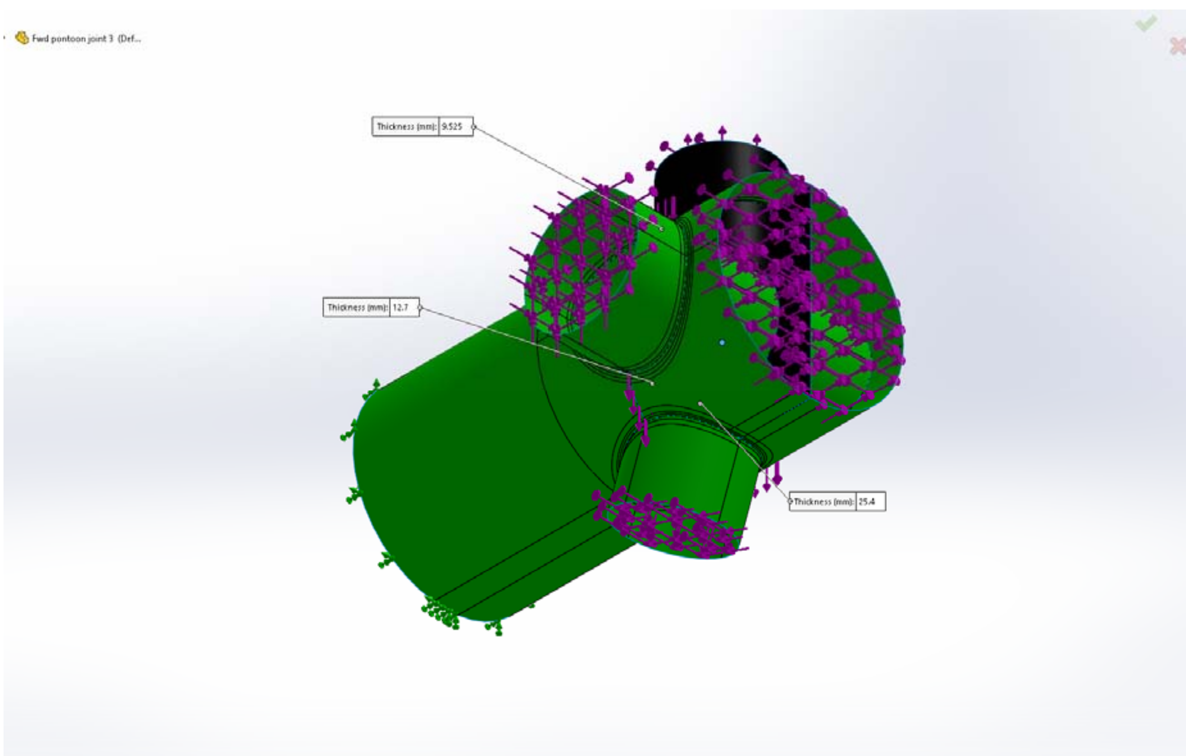

Figure A13. Forward Pontoon Joint (J3).

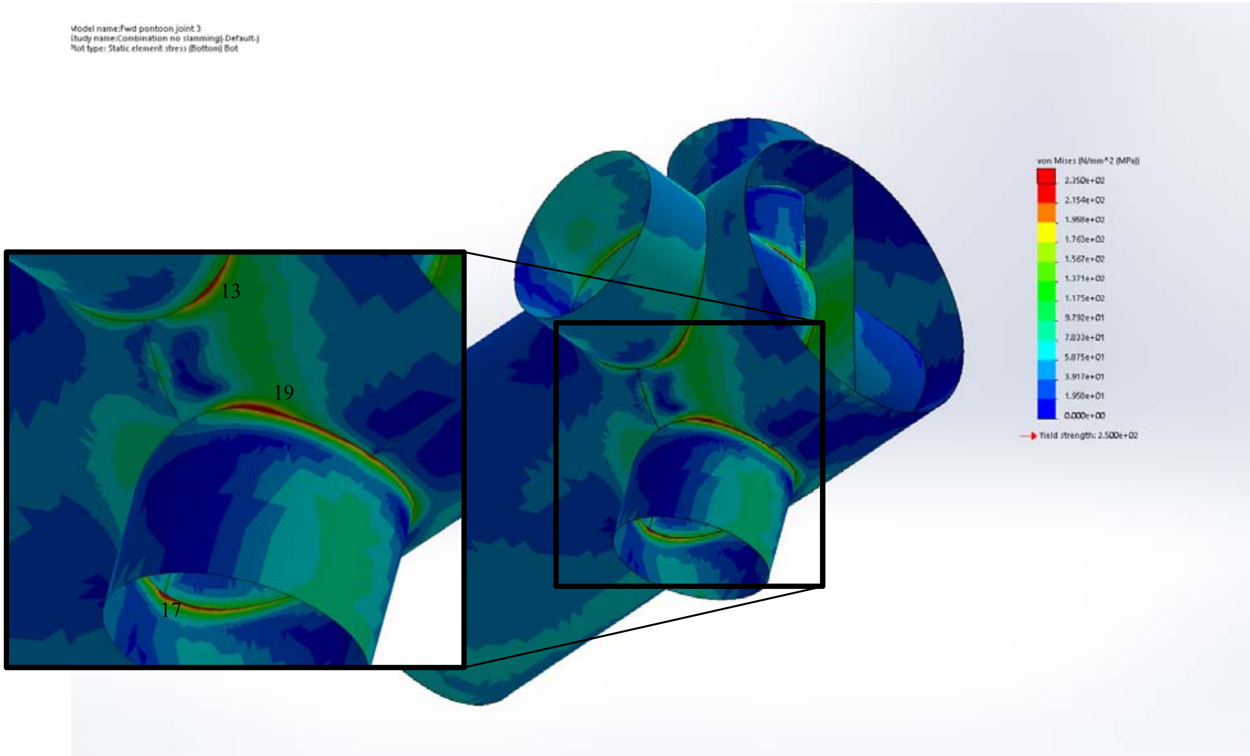

Figure A14. Von Mises stress of forward, inboard side of Forward Pontoon Joint; ; closeup of hot spot at Group 13, 17 and 19.

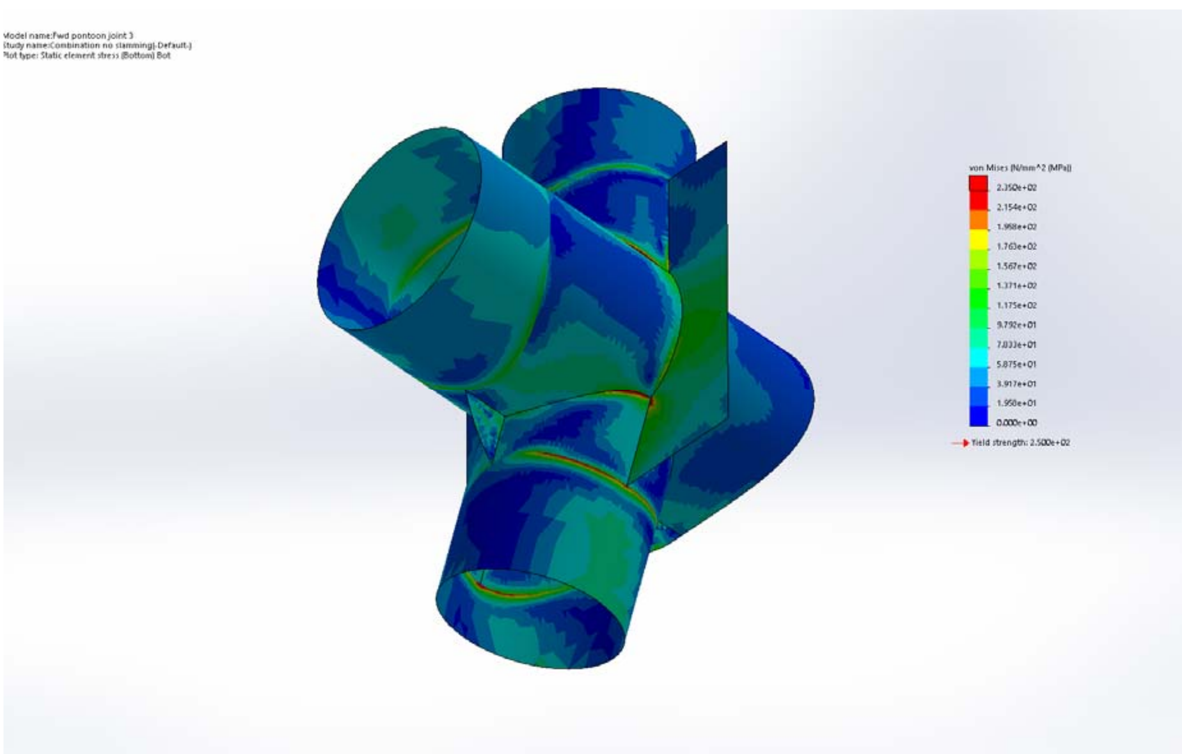

Figure A14a. Von Mises stress of forward, inboard side of Forward Pontoon Joint (shell removed for clarity).

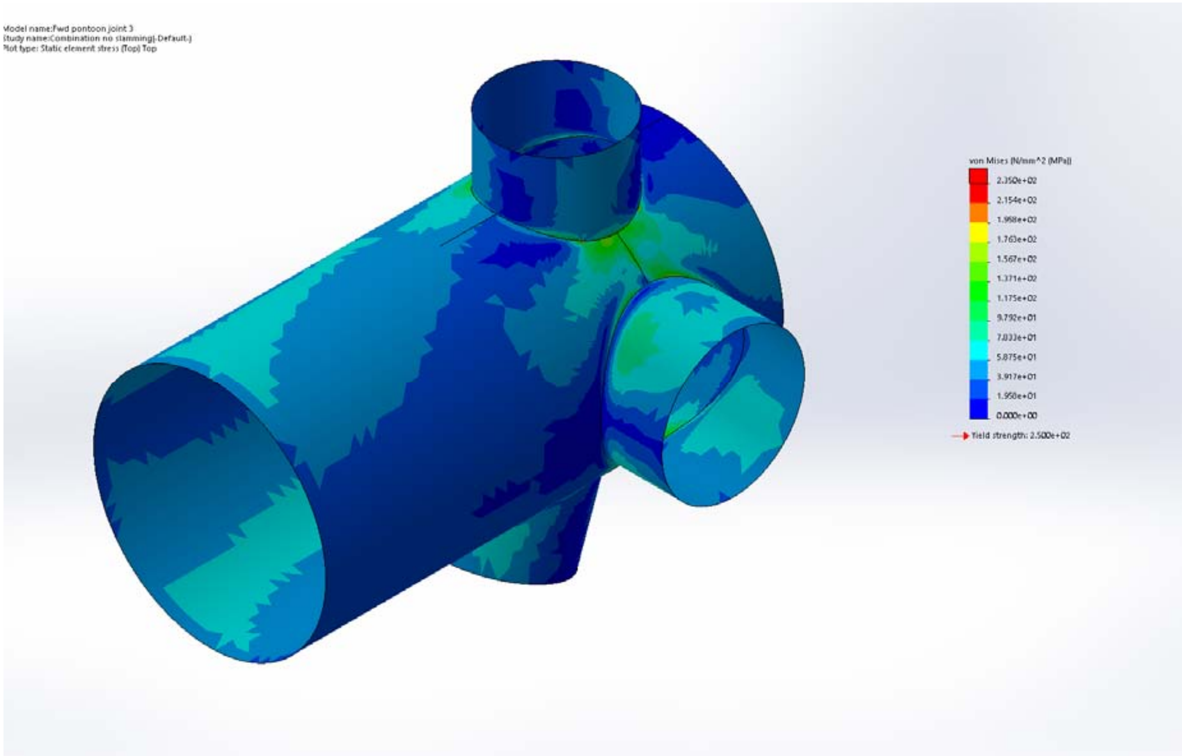

Figure A15. Von Mises stress of aft, inboard side of Forward Pontoon Joint.

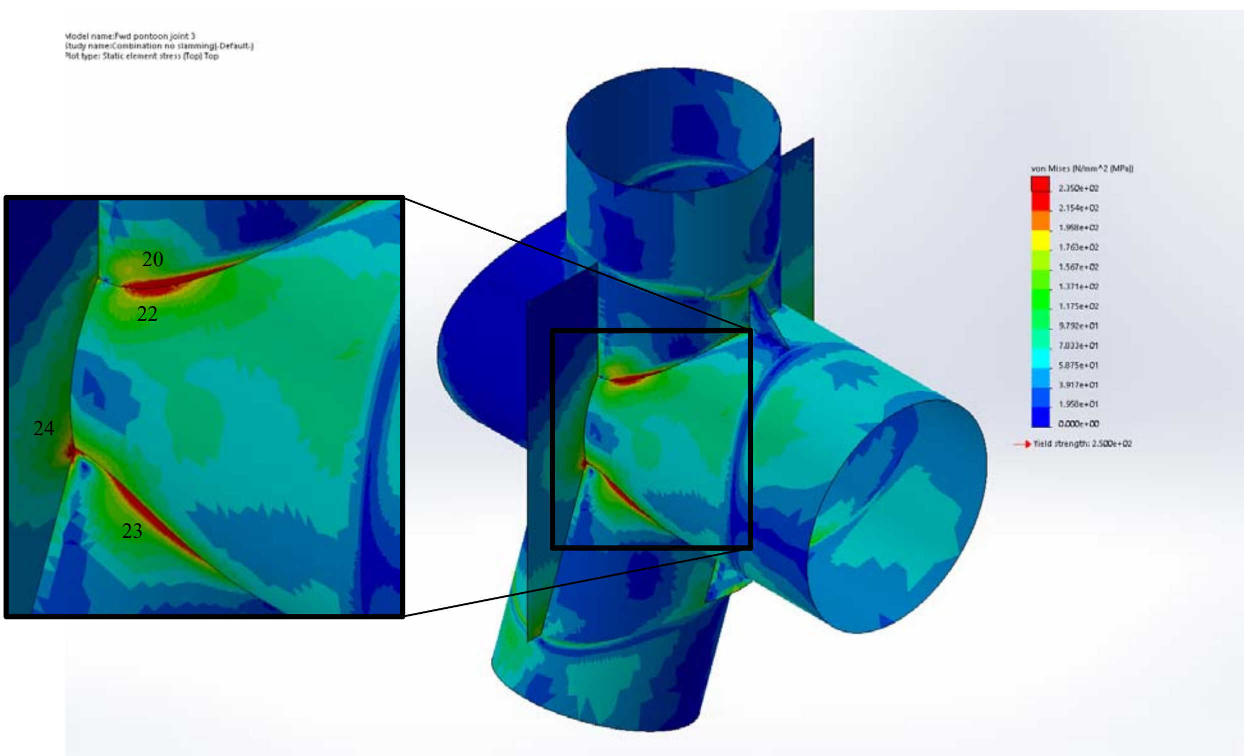

Figure A15a. Von Mises stress of aft, inboard side of Forward Pontoon Joint; shell removed for clarity; closeup of hot spot at Group 20, 22, 23 and 24.

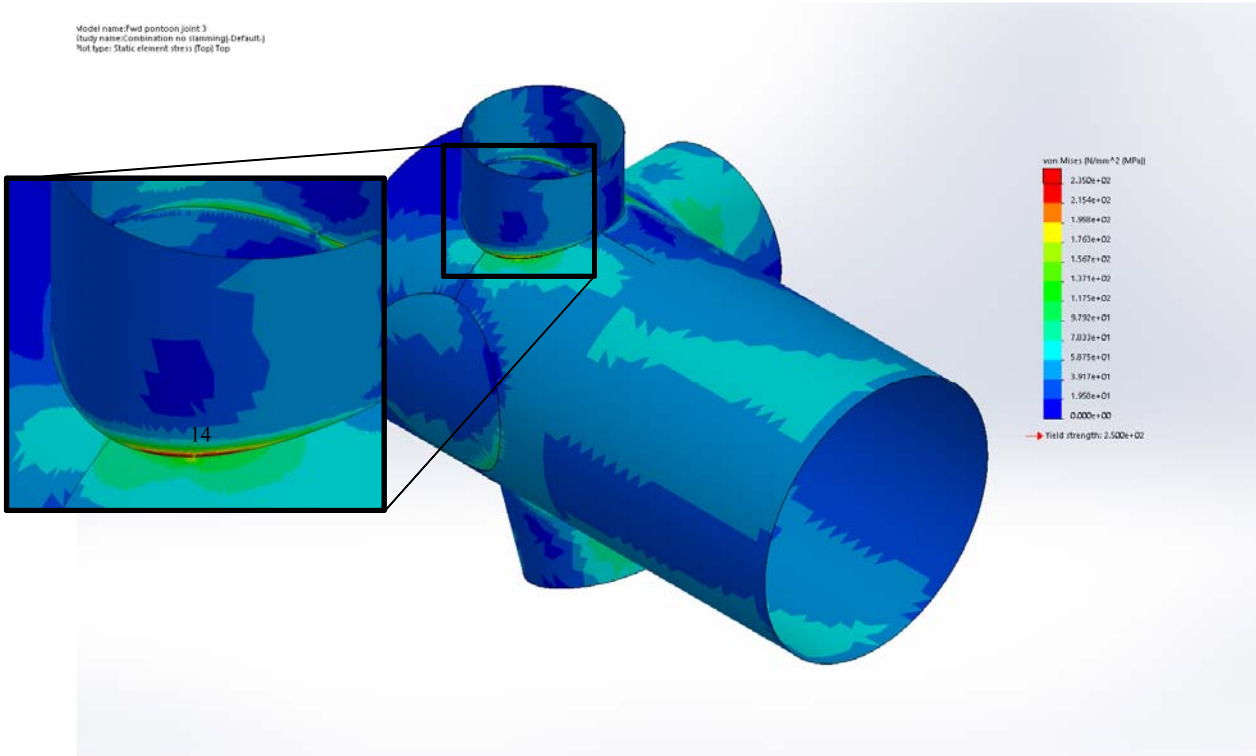

Figure A16. Von Mises stress of aft, outboard side of Forward Pontoon Joint; closeup of hot spot at Group 14.

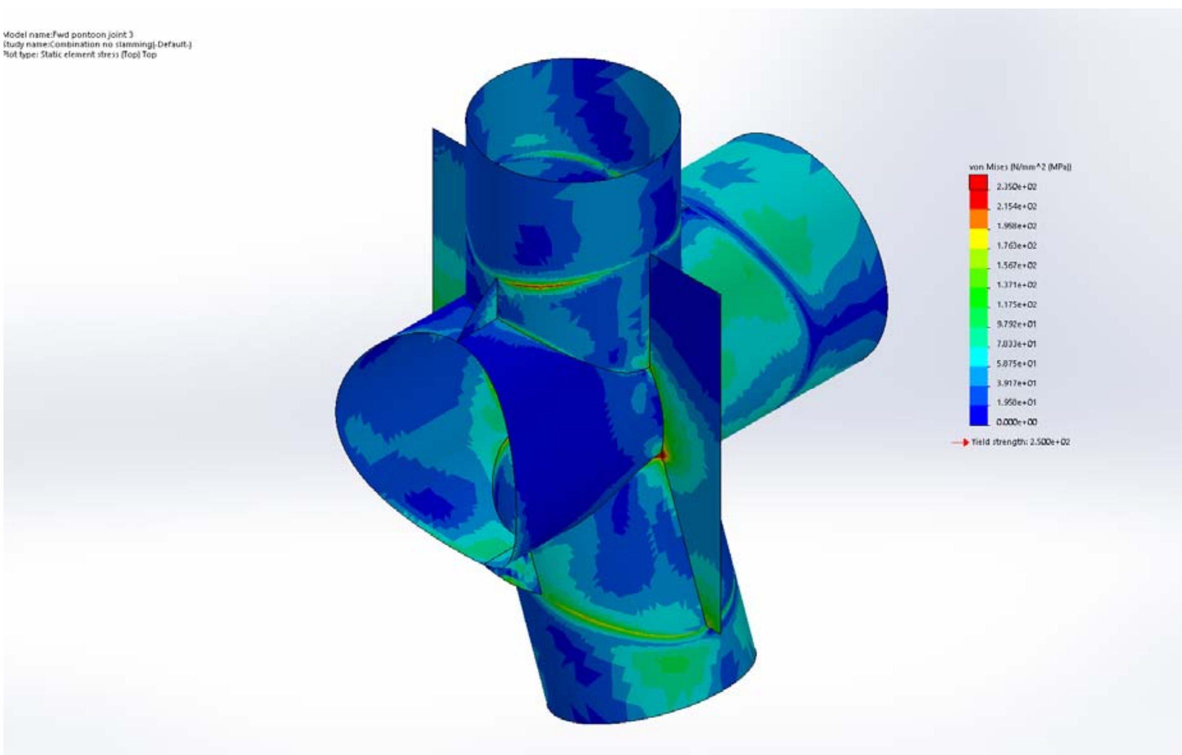

Figure A16a. Von Mises stress of aft, inboard side of Forward Pontoon Joint (shell removed for clarity).

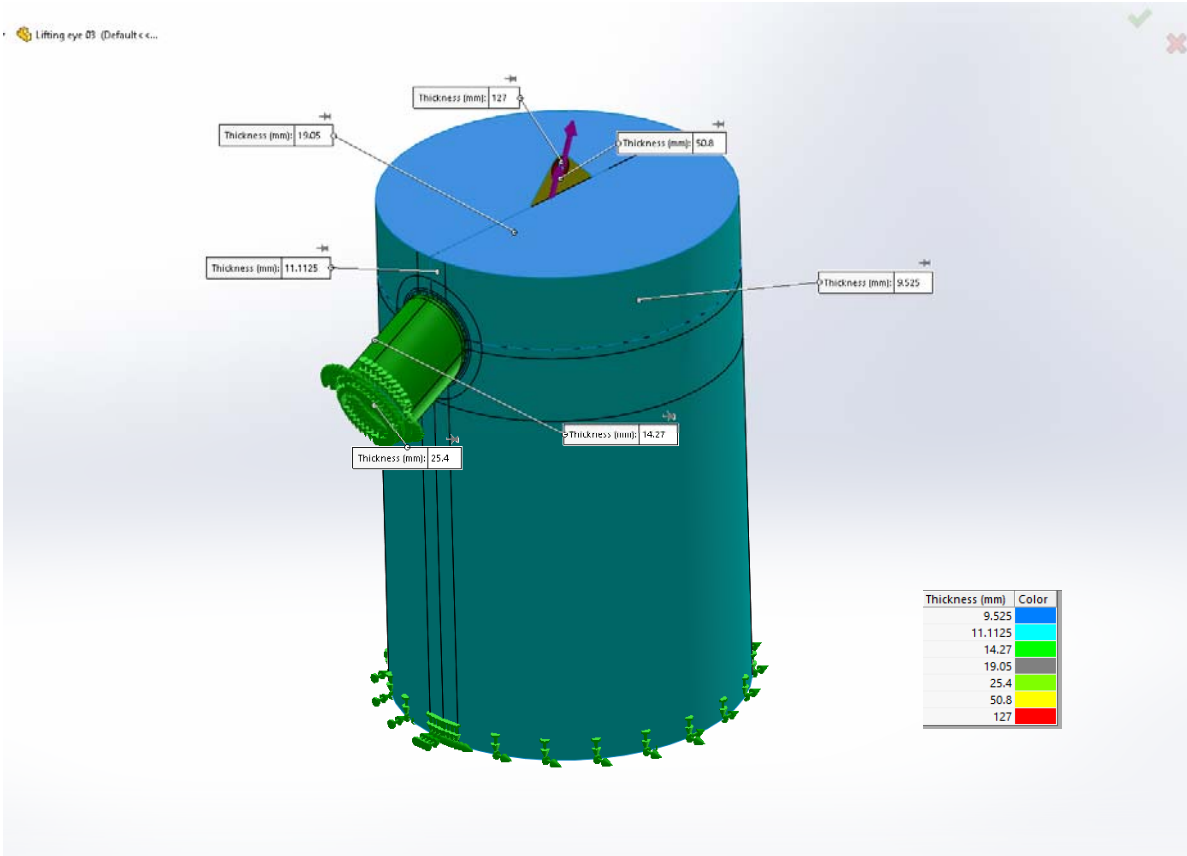

Figure A17. Lifting Eye and support structure.

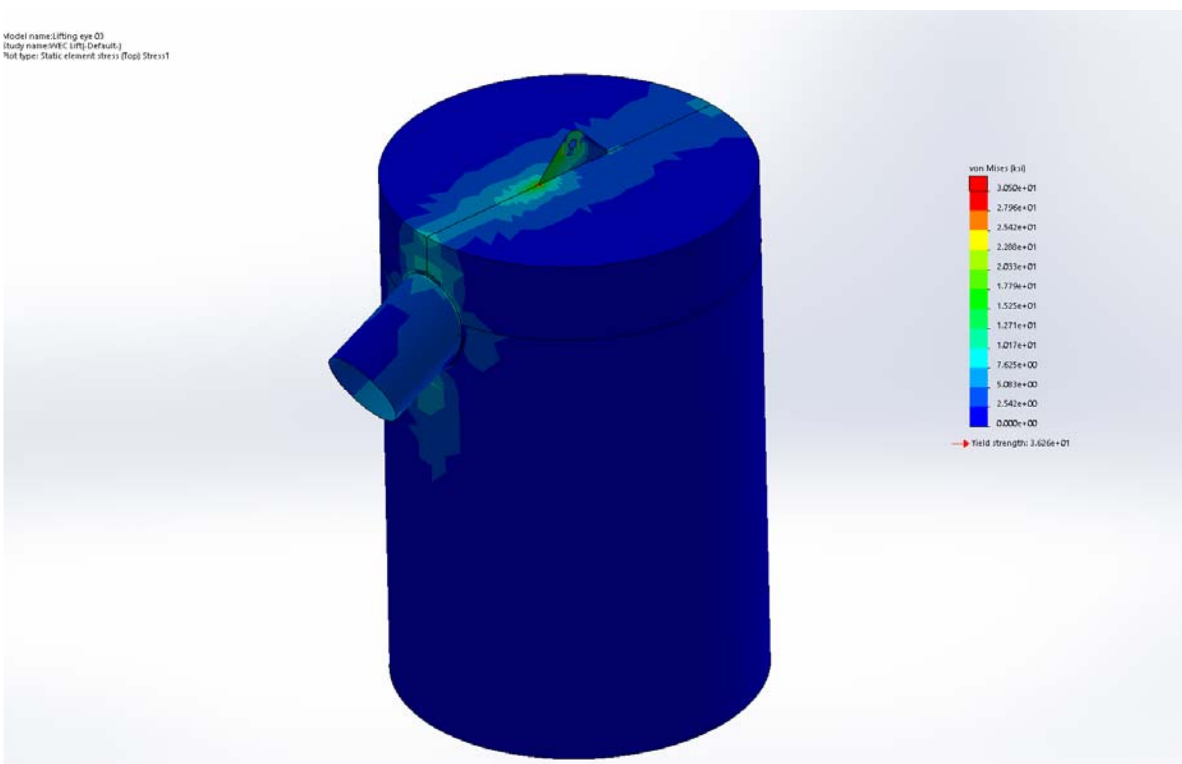

Figure A18. Von Mises stress of aft side of Lifting Eye and support structure.

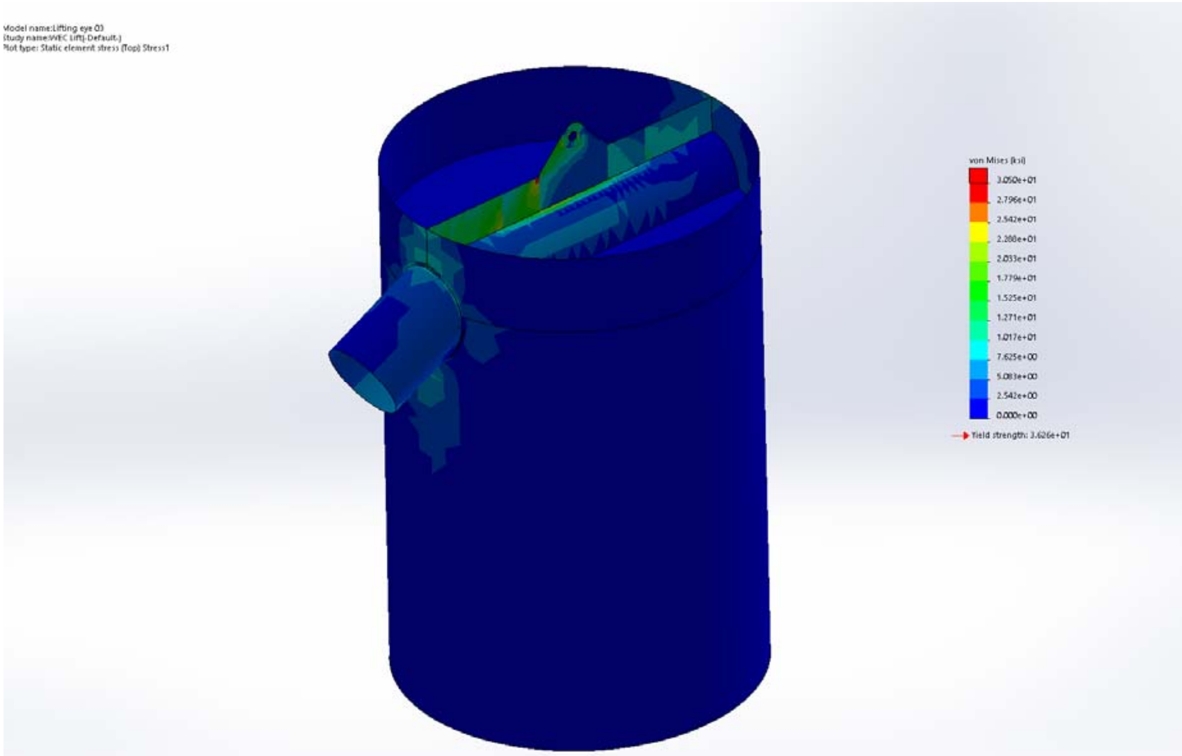

Figure A18a. Von Mises stress of aft side of Lifting Eye and support structure (end bulkhead removed for clarity).

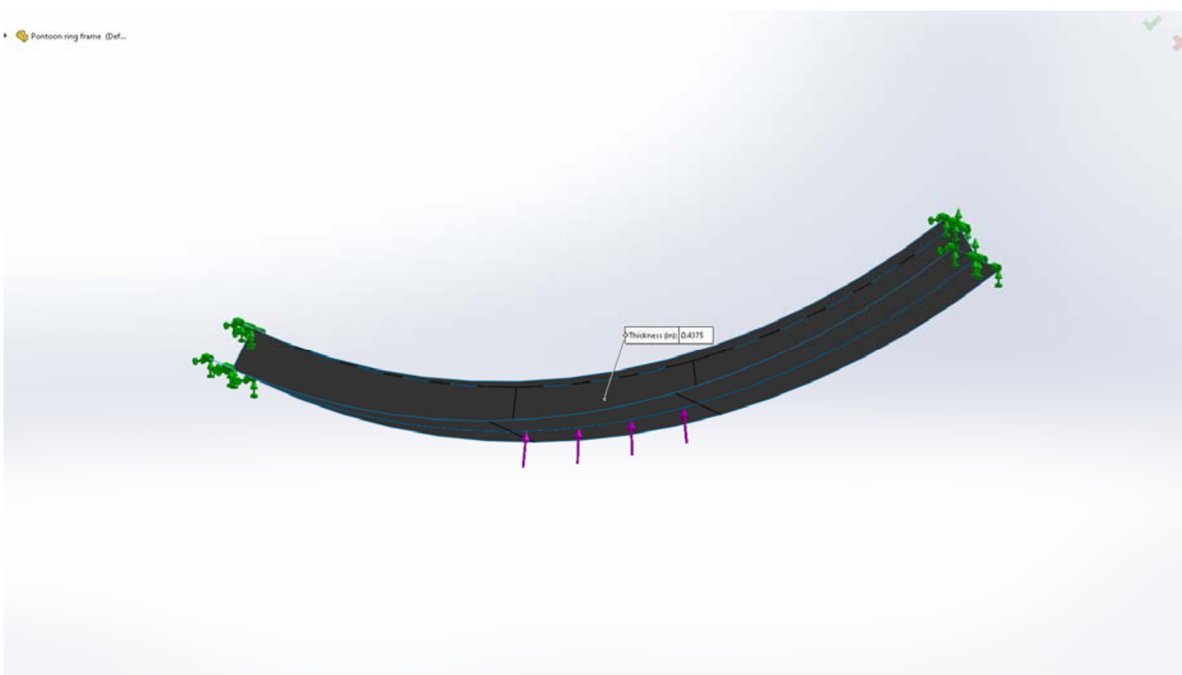

Figure A19. Typical Pontoon Ring Frame.

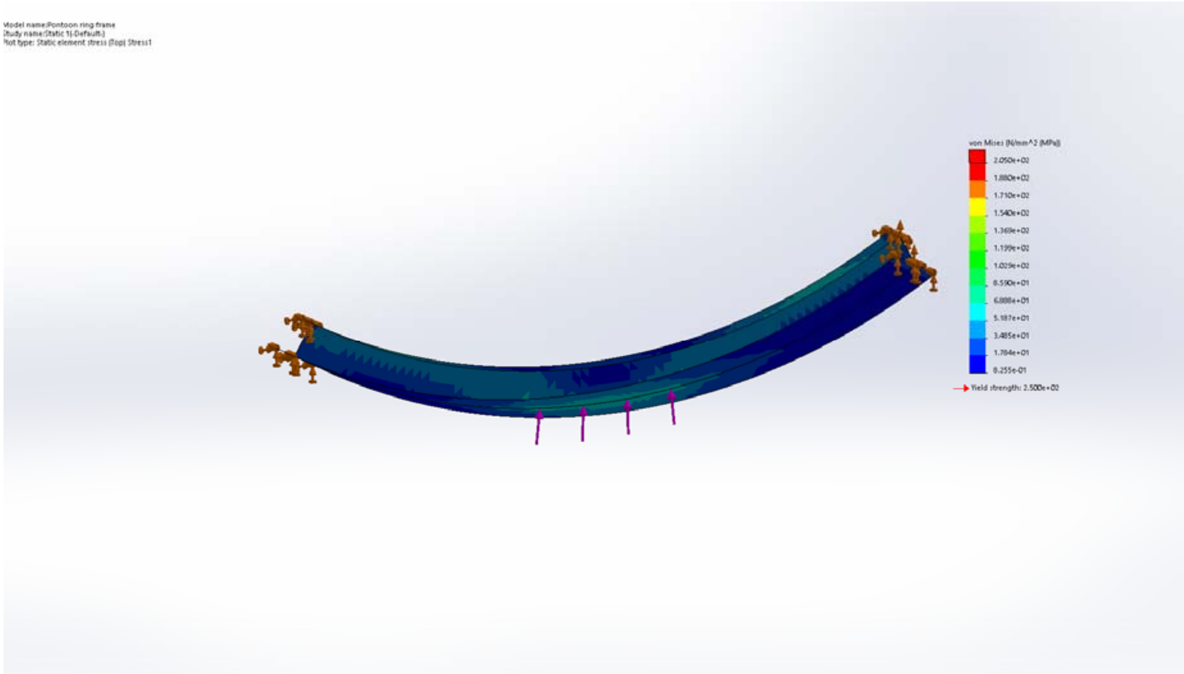

Figure A20. Von Mises stress in Typical Pontoon Ring Frame.

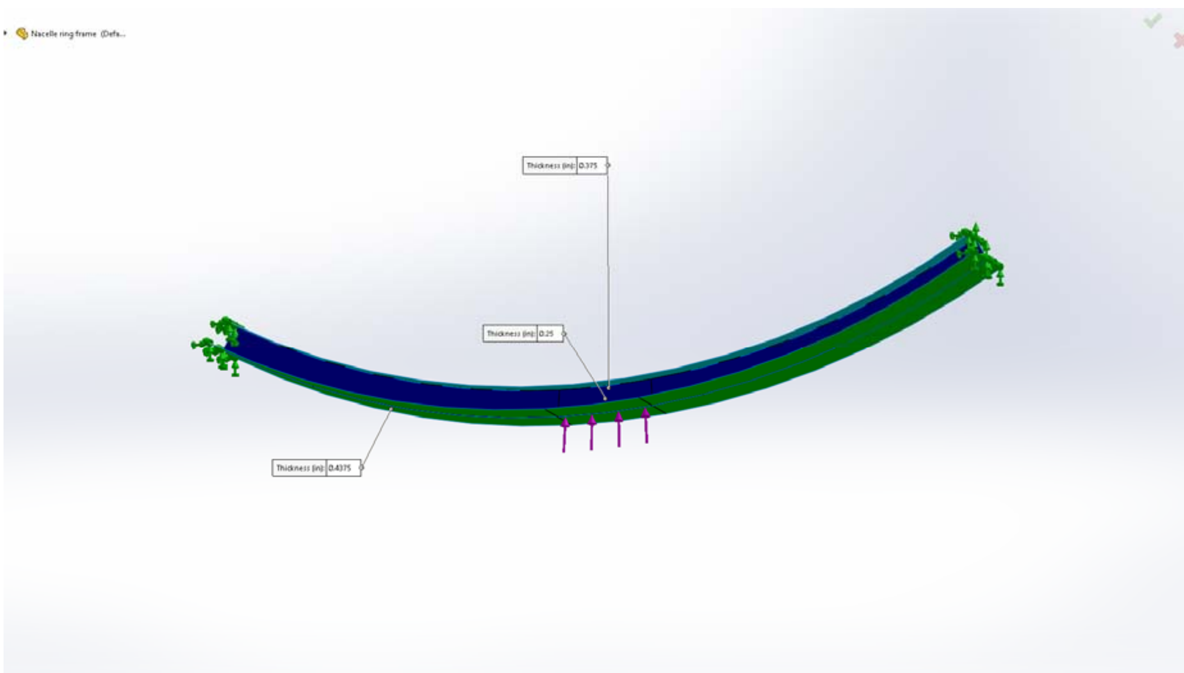

Figure A21. Typical Nacelle Ring Frame.

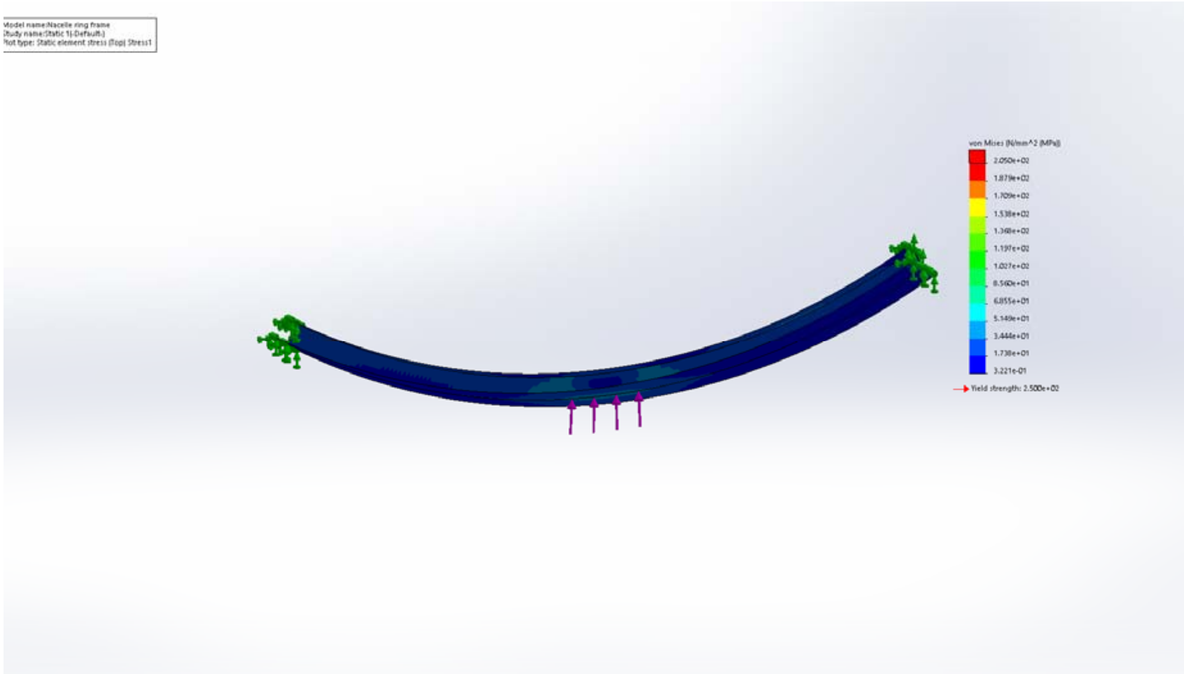

Figure A22. Von Mises stress in Typical Nacelle Ring Frame.

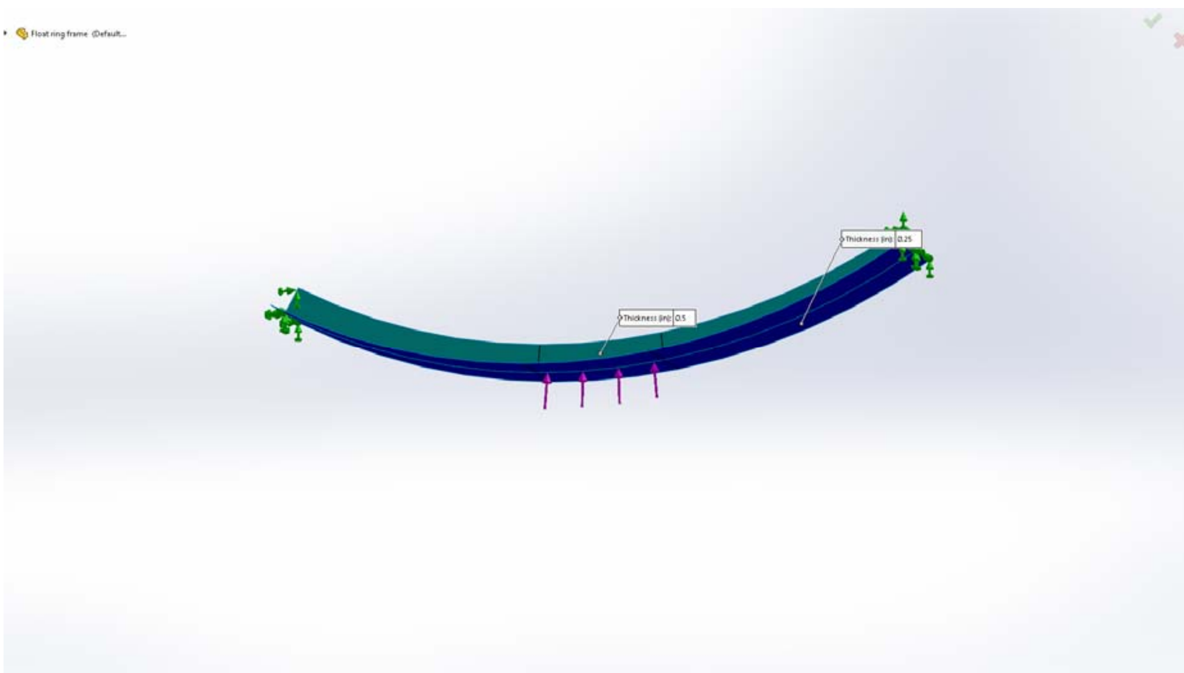

Figure A23. Typical Float Ring Frame.

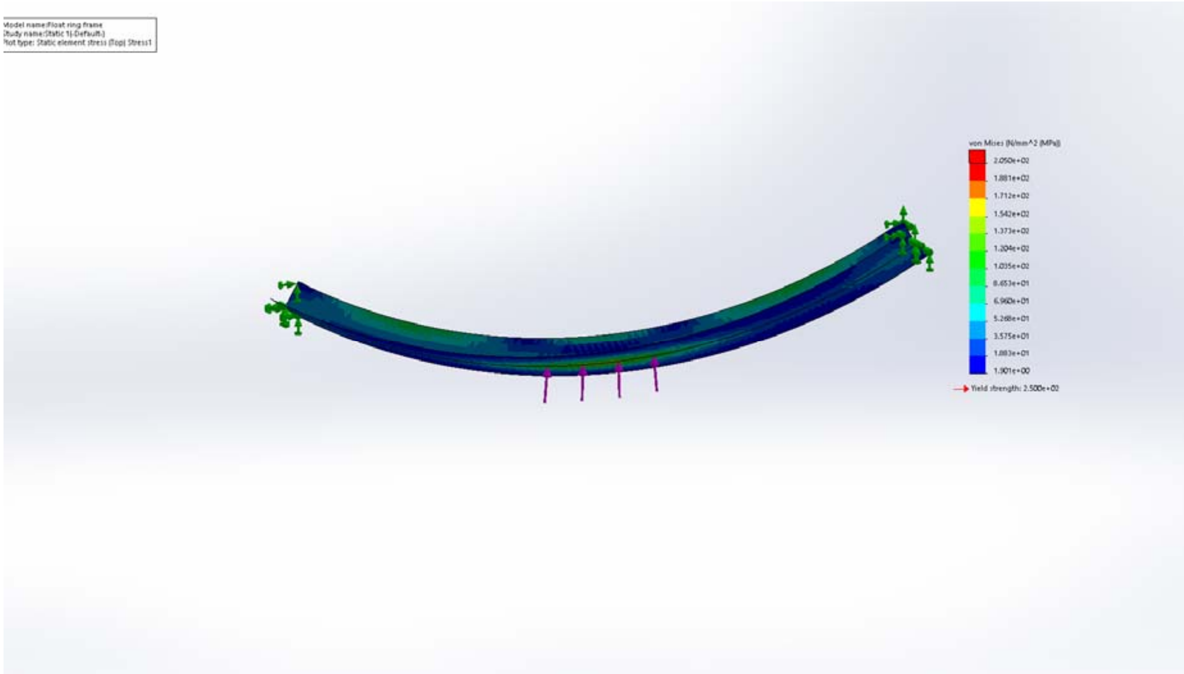

Figure A24. Von Mises stress in Typical Float Ring Frame.