

DELETED DUE TO ANONYMITY CHECK

Closed Ballast Dynamic Depth (CBDD) Ship Appendices

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Annexes:

Annex A – Ship Selection Statistic

(Port of Rotterdam, 2014)



ARRIVALS OF SEA-GOING VESSELS BY GT-CLASS AND THROUGHPUT

GT-class	January - December 2014		January - December 2013	
	Number	Throughput	Number	Throughput
0	-	1000	582	98
1001	-	1500	617	535
1501	-	2000	1.414	1.665
2001	-	2500	1.672	3.824
2501	-	3000	1.901	4.133
3001	-	3500	683	1.735
3501	-	4000	1.438	4.935
4001	-	4500	662	2.718
4501	-	5000	452	1.989
5001	-	6000	894	4.191
6001	-	7000	772	4.574
7001	-	8000	2.208	15.735
8001	-	9000	808	7.621
9001	-	10000	492	2.789
10001	-	20000	3.341	33.462
20001	-	30000	3.223	51.340
30001	-	40000	1.271	22.986
40001	-	50000	623	23.013
50001	-	60000	963	30.943
60001	-	70000	1.382	45.761
70001	-	80000	301	15.263
80001	-	90000	499	41.818
90001	-	100000	552	38.508
100001	-	110000	96	7.383
110001	-	120000	95	5.782
120001	-	130000	28	1.175
130001	-	140000	39	1.146
140001	-	150000	369	20.018
150001	-	180000	332	44.174
> 180000			81	5.418
Not specified				19
				16
Total		27.790		444.733
			29.448	440.464

Unit: Number of sea-going vessels, Throughput x 1.000 tons
Source: Port of Rotterdam Authority

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Annex B – Basis Ship and Alterations

DRAGONERA: First Seahorse 35 bulk carrier (Significant Ships of 2011)

Shipbuilder:.....Qidong daoda Heavy
Industry Co., Ltd (DDHI)
Vessel's Name: Dragonera
Hull No: DD-017
Owner/operator:Falcon Maritime
Country:Denmark
Designer:Grontmij A/S
Country:Denmark
Model test establishment used:
Force Technology
Flag:Malta
IMO number:9536428
Total number of sister ships already
completed (excluding ship presented):
.....2
Total number of sister ships still on
order:.....4

Dragonera is the first in a series of six green ships built to Grontmij's design. The Seahorse 35 bulk carriers have an optimised hull that has no bulbous bow as Grontmij say this is unnecessary and can be counter-productive if the vessel's bow is not always in the water.

The first six vessels will be delivered to Falcon Maritime of Denmark and were designed around four years ago. Grontmij say they have substantially upgraded the latest versions of the Seahorse 35 design and later models will sport a Mewis duct, a larger propeller and a smaller more efficient engine. Overall the Dragonera offers a 10% improvement in fuel efficiency on comparable vessels, but Dragonera is at the beginning of the changes and later models will be a further 25% more efficient compared to the early Seahorse design.

Improvements in efficiency can be added through the addition of waste heat recovery systems from Aalborg and exhaust gas recirculation units added to the latest engines from MAN Diesel & Turbo. Scrubbers also from Aalborg will allow the ships to operate in emission control areas (ECA) using more cost effective HFO rather than the significant higher priced MDO or MGO.

The Seahorse 35 is designed for 'economical and efficient operation, environmental friendliness and maintenance, safety, loading flexibility and to meet the latest regulations for bulk carriers, say the designers.

The vessel is designed with a shallow draught that maximises the cargo space.

It has five flush double-skinned cargo holds, wide hatches and no hopper tank tops, which provides easy access and cargo storage in all the vessel's hold.

Short turn-around times are achieved through effective cargo hold cleaning by portable washing machines, 'Outlets of water and compressed air are arranged in all cargo holds at tank top-level. A permanent washing water return line is arranged in each cargo hold at the tank top level. A portable pneumatic pump can, via the return line, transfer dirty washing water to two cargo hold washing water holding tanks arranged for temporary storage to enable cargo hold cleaning in sensitive and restricted areas," say the design team.

The double-skin configuration allows easy access to staff for structural inspections of the vessel, even when the vessel is loaded. All fire main line, hydraulic and fresh water (FW) piping and electrical cables in cargo the area are arranged in upper deck pipe ducts, avoiding cargo and green water damage and thereby requiring a minimum of maintenance work. A slender afterbody and a highly efficient NPT propeller offers optimal performance with minimum fuel consumption and a verticle stem has been adopted to improve the fuel efficiency in adverse weather conditions.

TECHNICAL PARTICULARS

Length oa:180m
Length bp:176.75m
Breadth moulded:30m
Depth moulded to upper deck:.....14.7m
Width of double skin
Side:1.70m
Bottom:.....1.60m
Draught
Scantling:10.10m
Design10.10m
Gross:24,210gt
Displacement:45,009tonnes
Light weight:10,396tonnes
Deadweight
Design:34,613dwt
Scantling:34,613dwt
Speed, service:14knots
Cargo capacity
Bale:45,654m³
Grain:45,733m³
Bunkers
Heavy oil:1586m³
Diesel oil:402m³

Water ballast:12,943m³
Daily fuel consumption
Main engine: 26.4tonnes/day
Auxiliaries:2tonnes/day
Classification society and notations:
..... DNV + 1A1 Bilk Carrier ESP ES(D),
ICE-1C. BC-A, DG-B, BWM(s,f) Holds(2,4)
May Be Empty, GRAB(20) ha(+) dk(+)
TMON BIS FUEL 9380cSt. 991kg/m³, -
15°C) BWM-E(s,f), COAT-PSCB(B), ICE-1C
Main engines
Design:MAN
Model:5S50MC-C7
Manufacturer:STX
Number:1
Type of fuel:HFO
Output of each engine:7500kW
Propellers
Designer/manufacturer:.....Stone
Marine Propulsion
Number:4
Diameter:5.6m
Speed:124rpm
Diesel-driven alternators
Number:3
Engine make/type:Hyundai
Heavy Industries Co., Ltd/ 6H17/28
Type of fuel:HFO and MDO
Output/speed of each set:.....
654kW x 900rpm
Alternator make/type:
marine three-phase
Output/speed of each set:
600kW x 900rpm
Boilers
Number:1
Type:MC5901R1
Make:Kangrim
Output, each boiler:1800kg/h
Cargo cranes/ cargo gear, Number:4
Other cranes, Number:3
Mooring equipment, Number:4
Stern thruster
Make:Changzhouzhonghao
Number:1
Output:7500kW
Waste disposal plant
Incinerator: ..Nanjing Luzhou Co., Ltd
Waste compactor:Hansun Marine
Technology Co., Ltd
Contract date:17 November 2008
Launch/float-out date:5 March 2011
Delivery date:26 August 2011

Annex C – First Principles of Ship Design (Structural Strength)

IACS section modulus equations and variables:

IACS section modulus [m ³]	$Z_{IACS_MIN} = 0.9 \times C \times L^2 \times B \times (C_B + 0.7) \times k \times 10^{-6} = 9.8750$
C	$C = 10.75 - \left(\frac{300 - L}{100} \right)^{1.5} = 9.4355$
Length [m]	$L = 180$
Breadth [m]	$B = 30$
Block coefficient	$C_B = \frac{\Delta/\rho}{L \times B \times T} = 0.8338$
Displacement [t]	$\Delta = 46613$
Salt water density [t/m ³]	$\rho = 1.025$
Draught [m]	$T = 10.1$
Material factor	$k = 0.78$

Table 2: Material factor k

Minimum yield stress R_{eH} , in N/mm ²	k
235	1.0
315	0.78
355	0.72
390	0.68

Global load calculation equations and variables:

Area moment of inertia [m ⁴]	$I_{OWN} = \frac{bd^3}{12}$
Neutral axis location from the bottom of the keel [m]	$Y_n = \frac{\sum 1st\ Mmt}{\sum Area}$
Second moment of area about the bottom of the keel [m ⁴]	$I_{BASE} = \sum I_{OWN} + \sum 2nd\ Mmt$
Second moment of area about the neutral axis [m ⁴]	$I_{NA} = \textcircled{2} \times \left(I_{BASE} - \sum Area \times Y_n^2 \right)$
Section modulus [m ³]	$Z = \frac{I_{NA}}{Y}$

Global load calculations

Expanded case:

Item	Plate Thickness (m)	Thickness factor	Assumed Stiffening (40%)	Length b (m)	Breadth d (m)	Area (m ²)	Lever (m)	1st Mmt (m ³)	2nd Mmt (m ⁴)	Iown (m ⁴)
Bottom Shell	0.028	0.9	1.4	15	0.03528	0.5292	0	0.000000	0.000000	0.000055
Bottom Girder 1	0.028	0.6	1	0.0168	1.6	0.02688	0.8	0.021504	0.017203	0.005734
Bottom Girder 2	0.028	0.6	1	0.0168	1.6	0.02688	0.8	0.021504	0.017203	0.005734
Bottom Girder 3	0.028	0.6	1	0.0168	1.6	0.02688	0.8	0.021504	0.017203	0.005734
Bottom Girder 4	0.028	0.6	1	0.0168	1.6	0.02688	0.8	0.021504	0.017203	0.005734
Bottom Girder 5	0.028	0.6	1	0.0168	1.6	0.02688	0.8	0.021504	0.017203	0.005734
Tank Top	0.028	0.8	1.4	15	0.03136	0.4704	1.6	0.752640	1.204224	0.000039
Lower Inner Side Shell	0.028	1	1.4	0.0392	8.5	0.3332	5.05	1.682660	8.497433	2.006142
Lower Outer Side Shell	0.028	1	1.4	0.0392	10.1	0.39592	5.05	1.999396	10.096950	3.365650
Inner Lower Side girder	0.028	0.8	1.4	1.7	0.03136	0.053312	5.5	0.293216	1.612688	0.000004
Upper Inner Side Shell	0.028	1	1	0.028	4.8	0.1344	12.3	1.653120	20.333376	0.258048
Upper Outer Side Shell	0.028	1	1	0.028	4.8	0.1344	12.3	1.653120	20.333376	0.258048
Wing Tank	0.028	0.8	1.4	2.286	0.03136	0.071689	14.13	1.012965	14.313196	0.007809
Deck	0.028	1.3	1.4	4.44	0.05096	0.2262624	14.7	3.326057	48.893042	0.000049
Hatch Coaming	0.028	2	1	0.056	3.2	0.1792	16.3	2.920960	47.611648	0.152917
Total						2.6623834		15.401654	172.981948	6.077433

Neutral Axis location from the bottom of the keel (m)	5.78
Second Moment of area about the bottom of the keel (m ⁴)	179.0594
Second moment of area about the neutral axis (m ⁴)	179.9243

	Distance between Yn and Yown	Section Modulus (m ³)
Hatch	12.12	14.8513
Keel	5.78	31.1023
Deck	8.92	20.1820

Collapsed case:

Item	Plate Thickness (m)	Thickness factor	Assumed Stiffening (40%)	Length b (m)	Breadth d (m)	Area (m ²)	Lever (m)	1st Mmt (m ³)	2nd Mmt (m ⁴)	Iown (m ⁴)
Bottom Shell	0.028	0.9	1.4	15	0.03528	0.5292	0	0.000000	0.000000	0.000055
Bottom Girder 1	0.028	0.6	1	0.0168	1.6	0.02688	0.8	0.021504	0.017203	0.005734
Bottom Girder 2	0.028	0.6	1	0.0168	1.6	0.02688	0.8	0.021504	0.017203	0.005734
Bottom Girder 3	0.028	0.6	1	0.0168	1.6	0.02688	0.8	0.021504	0.017203	0.005734
Bottom Girder 4	0.028	0.6	1	0.0168	1.6	0.02688	0.8	0.021504	0.017203	0.005734
Bottom Girder 5	0.028	0.6	1	0.0168	1.6	0.02688	0.8	0.021504	0.017203	0.005734
Tank Top	0.028	0.8	1.4	15	0.03136	0.4704	1.6	0.752640	1.204224	0.000039
Lower Inner Side Shell	0.028	1	1.4	0.0392	8.5	0.3332	5.05	1.682660	8.497433	2.006142
Lower Outer Side Shell	0.028	1	1.4	0.0392	10.1	0.39592	5.05	1.999396	10.096950	3.365650
Inner Lower Side girder	0.028	0.8	1.4	1.7	0.03136	0.053312	5.5	0.293216	1.612688	0.000004
Upper Inner Side Shell	0.028	1	1	0.028	4.8	0.1344	7.7	1.034880	7.968576	0.258048
Upper Outer Side Shell	0.028	1	1	0.028	4.8	0.1344	7.7	1.034880	7.968576	0.258048
Wing Tank	0.028	0.8	1.4	2.286	0.03136	0.071689	9.5285	0.683088	6.508806	0.007809
Deck	0.028	1.3	1.4	4.44	0.05096	0.2262624	10.1	2.285250	23.081027	0.000049
Hatch Coaming	0.028	2	1	0.056	3.2	0.1792	11.7	2.096640	24.530688	0.152917
Total						2.6623834		11.970170	91.554985	6.077433

Neutral Axis location from the bottom of the keel (m)	4.50
Second Moment of area about the bottom of the keel (m ⁴)	97.6324
Second moment of area about the neutral axis (m ⁴)	87.6282

	Distance between Yn and Yown	Section Modulus (m ³)
Hatch	8.80	9.9533
Keel	4.50	19.4901
Deck	5.60	15.6368

Comparing section moduli with IACS minimum criteria:

IACS Z _{MIN} (m ³)		Z (m ³)	
		Expanded Case	Collapse Case
9.875038776	Hatch	14.8513	9.9533
	Keel	31.1023	19.4901
	Deck	20.1820	15.6368

Validation of the Stiffener coefficient (40%)

Deck dimensions:

Total area (m ²)	0.226262
Length (m)	4.44
Thickness (without stiffener factor) (m)	0.0364
Area without stiffening (m ²)	0.161616
Area for stiffening (m ²)	0.064646

Flat bar stiffener spacing calculation and comparison:

	Dragonera Seahorse	Capesize ship
Web height h_w (m)	0.39	
Web thickness t_w (m)	0.027	
Area of stiffener (m ²)	0.01053	
Deck length (m)	4.44	12.32
Number of stiffeners	7	14
Stiffener spacing (m)	0.6343	0.88

IACS flat bar stiffener criteria:

Stiffener dimension criteria	$\frac{h_w}{t_w} \leq 20\sqrt{k}$
Web height [m]	$h_w = 0.39$
Web thickness [m]	$t_w = 0.027$
Material factor	$k = 0.78$

Compliance with IACS rules:

k	0.78
h_w/t_w	14.44444
$20\sqrt{k}$	17.66352

Annex D – First Principles of Ship Design (Stability)

Parameters:

Parameters	Dimension (m)
Length Overall	180
Breadth	30
$T_{Extended}$	10.1
$T_{Collapsed}$	2.276
Depth _{Extended}	14.7
Depth _{Collapsed}	10.1

Using these parameters, the stability of the ship has to be calculated in the extended and collapsed draught. It is important to note here that following IACS rules, the GM of the ship must be greater than 0.15m.

Using the equation $GM=KB+BM-KG$ it is possible to find if the vessel is floating upright.

To calculate KB, the equation $KB = T/2$ is used due to the high block coefficient of the vessel, meaning that for stability calculations it could be considered as a block shape.

Calculating BM I used $BM = \frac{B^2}{12 \times T}$ as described in Ship Stability for Masters Mates by D.R. Derrett.

And finally for KG, I estimated the centres of gravity to all heavy parts of the ship and then combined them to give a single value.

Extended Case:

$$KB = \frac{T_{Extended}}{2}$$

$$KB = \frac{10.1}{2}$$

$$KB = 5.05 \text{ m}$$

$$BM = \frac{B^2}{12 \times T_{Extended}}$$

$$BM = \frac{30^2}{12 \times 10.1}$$

$$BM = 7.426 \text{ m}$$

For KG to be calculated, the centroids of the main components of the ship had to be estimated. These were split up into 3 sections, which were as follows. The distance from the keel for the steel mass was calculated when finding the second moment of area of the steel structure during the structural strength calculations.

Component	Centre of Gravity Distance from Keel [m]	Mass [Tonnes]	Moment [Tonne Metres]
Outfit Machinery	5	1060.5	5302.5
Cargo	10	34613	346130
Steel (Expanded)	5.785	9335.5	54005.87
Steel (Collapsed)	4.496	9335.5	41972.41

The KG of the ship could now be found by summing the moments of the extended case of the ship and dividing by the summation of moments.

$$KG = \frac{\sum Moments}{\sum Mass}$$

$$KG = \frac{5302.5 + 346130 + 54005.87}{1060.5 + 34613 + 9335.5}$$

$$KG = 9.008 \text{ m}$$

This now allows for GM to be calculated as follows

$$GM = KB + BM - KG$$

$$GM = 5.05 + 7.426 - 9.008$$

$$GM = 3.47 \text{ M}$$

As stated in the IMO Code on Intact Stability, the initial metacentric height, GM_0 should not be less than 0.15m for any vessel. This regulation is shown to have been met.

Collapsed Case

Firstly it needs to be explained how the draught of the vessel in the collapsed case was calculated. This was done by calculating the Tonnes Per Centimetre immersion for the ship.

$$TPC = \frac{Water \ Plane \ Area \times \ Density}{100}$$

$$Water \ Plane \ Area = B \times LBP \times Block \ Coefficient$$

$$WPA = 30 \times 176.75 \times 0.84$$

$$WPA = 4456.34m^2$$

$$TPC = \frac{4456.34 \times 1.025}{100}$$

$$TPC = 45.7 \text{ tonnes}$$

Finding the tonnes required to alter the draught of the ship by one centimetre, it was possible to calculate the draught of the ship when the cargo on board was discharged.

$$\text{Collapsed Draught} = \frac{\text{Lightweight /TPC}}{100}$$

$$\text{Collapsed Draught} = \frac{10396 /45.7}{100}$$

$$\text{Collapsed Draught} = 2.28m$$

This now allows KB to be calculated as KB for the extended case was calculated.

$$KB = \frac{T_{\text{Collapsed}}}{2}$$

$$KB = \frac{2.28}{2}$$

$$KB = 1.16m$$

$$BM = \frac{B^2}{12 \times T_{\text{Extended}}}$$

$$BM = \frac{30^2}{12 \times 2.28}$$

$$BM = 32.95m$$

For KG to be calculated in the collapsed case, the original extended case of KG was used, and then the problem was modelled as a removal of mass with regard to the removed cargo, and a shift of mass with regard to the mass of the steel hull structure of the ship being moved downwards.

$$G_0G_1 = \frac{(\text{Mass being moved}) \times (\text{Distance from COG of ship})}{\text{New Displacement of ship}}$$

$$G_0G_1 = \frac{(\text{Mass of Cargo}) \times (\text{COG distance from keel} - KG_{\text{Extended}})}{\text{Displacement} - \text{Cargo}}$$

$$G_0G_1 = \frac{34613 \times (10 - 9.01)}{45009 - 34613}$$

$$G_0G_1 = 3.30m$$

$$G_1G_2 = \frac{(\text{Mass of Steel}) \times (\text{COG distance from keel}_{\text{Expanded}} - \text{COG distance from keel}_{\text{Collapsed}})}{\text{Lightship}}$$

$$G_1G_2 = \frac{9335.5 \times (5.785 - 4.496)}{10396}$$

$$G_1G_2 = 1.16m$$

Finding these shifts in the centre of gravity of the ship, it is now possible to calculate the new collapsed case KG

$$KG_{\text{Collapsed}} = KG_{\text{Extended}} - G_0G_1 - G_1G_2$$

$$KG_{\text{Collapsed}} = 9.001 - 3.30 - 1.16$$

$$KG_{\text{Collapsed}} = 4.55m$$

This now allows for GM to be calculated as follows

$$GM = KB + BM - KG$$

$$GM = 1.16 + 32.95 - 4.55$$

$$GM = 29.54m$$

As stated in the IMO Code on Intact Stability, the initial metacentric height, GM_0 should not be less than 0.15m for any vessel. This regulation is shown to have been met.

Carl Bro a/s - DMC
NAPA/D/LD/060614
P40357500/CXS4210
P40357500

DIAMOND 53
SUMMARY OF
LOADING CONDITIONS

DATE 2006-09-29
TIME 10:47
USER JAN
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LOADING AND STRENGTH INFORMATION

CASE	TEXT	DISP t	CARGO t	BUNKER t	BALLAST t	T(M) m	TRIM m	GM(F) m	SHMIN t	SHMAX t	BMMIN tm	BMMAX tm
L-00	LIGHT WEIGHT - NO SUPPLY	11044	0	0	0	2.63	3.96	22.43	-1319	1940	-25	78289
L-01	LIGHT BALLAST - DEP 100 %	28910	0	2441	15424	6.10	1.67	8.09	-2252	3732	-3	120996
L-02	LIGHT BALLAST - ARR 10 %	29249	0	392	17813	6.16	1.60	8.43	-2171	2392	-1	107837
L-03	HEAVY BALLAST - DEP 100 %	44203	0	2441	30718	8.93	1.82	4.91	-4473	3471	-108962	60686
L-04	HEAVY BALLAST - ARR 10 %	43703	0	392	32268	8.83	1.04	5.20	-4273	3744	-123982	43496
L-05	DOCKING KEEL - ARR 10 %	14848	0	392	3412	3.25	0.00	15.88	-2965	2531	0	130158
L-06	PROPELLER IMMERSION - ARR 10 %	22626	0	392	11191	4.64	-3.59	10.27	-1797	2523	-4	105985
L-07	HOMOGEN DESIGN - DEP 100 %	56439	40766	2441	2187	11.09	1.26	3.21	-1163	1132	-10540	43072
L-08	HOMOGEN DESIGN - ARR 10 %	53402	40766	392	1200	10.57	0.16	3.37	-393	819	-13316	7239
L-09	HOMOGEN SCANT - DEP 100 %	64772	51287	2441	0	12.54	1.70	2.97	-591	1056	-30511	1669
L-10	HOMOGEN SCANT - ARR 10 %	62722	51287	392	0	12.21	-0.16	2.97	-1551	1194	-53113	1248
L-11	C0.80 PANA FW - DEP 100 %	60007	46522	2441	0	12.02	0.00	3.58	-1470	997	-36121	18713
L-12	C0.80 PANA FW - DEP 10 %	59993	48557	392	0	12.02	0.00	3.57	-1242	1019	-56534	1890
L-13	C1.35 CH 1245 - DEP 100 %	64603	51118	2441	0	12.52	1.14	4.76	-4447	4062	-9409	164629
L-14	C1.35 CH 1245 - ARR 10 %	63212	51118	392	658	12.30	0.01	4.83	-4228	4059	-10120	151724
L-15	C3.00 CH 135 - DEP 100 %	64682	51196	2441	0	12.53	1.15	6.51	-4860	5119	-6670	143209
L-16	C3.00 CH 135 - ARR 10 %	63290	51196	392	658	12.31	0.03	6.61	-4684	4771	-6990	128796
L-17	C3.00 CH 135 - DEP 50 %	64539	52200	1295	0	12.52	0.61	6.56	-4694	5001	-8490	127361
L-18	C3.00 CH 135 - ARR 10 %	64046	52200	392	410	12.44	0.01	6.63	-4564	4970	-19592	121510
L-19	STEEL COILS - DEP 100 %	64236	50750	2441	0	12.47	0.39	8.16	-1407	1407	-54512	6790
L-20	STEEL COILS - ARR 10 %	64264	50750	392	2078	12.48	0.12	8.36	-1534	1503	-58997	8414
L-21	TIMBER LOAD - DEP 100 %	63437	46256	2441	3695	12.33	0.55	1.30	-1377	1346	-33560	7998
L-22	TIMBER LOAD - ARR 10 %	63666	47331	392	4899	12.37	0.10	1.27	-1710	1564	-63131	9102

(STABILITY INFORMATION MANUAL

Including longitudinal strength

DIAMOND 53 – HANDYMAX BULK CARRIER

M/S SPAR SCORPIO, Page 7-1

Annex E – Lifting Mechanism

Appendix I to Annex E - Engineering Requirements (Hydraulic system calculations)

All the calculations are done in excel and the assumptions that have been made can be changed if necessary.

It was assumed that half of the lightship was being lifted.

$$\text{Mass lifted} = \text{Lightship} \times 50\%$$

$$\text{Mass lifted} = 10396t \times 50\%$$

$$\text{Mass lifted} = 5198t$$

A large safety factor of 3 is added, this means that if a single cylinder fails then the cylinders on either side of it should be able to cope with the extra force without reaching their limitations, the safety factor is also to make sure that the assumption of 50% of the lightship being lifted is appropriate.

$$\text{Mass lifted with SF} = 5198t \times 3$$

$$\text{Mass lifted with SF} = 15594t$$

To find the force required to lift the top half of the ship not only must gravity be taken into account but also acceleration of movement force must be considered. To do this an assumed lifting time of 6 hours was taken, this seemed to be appropriate for the ship size. The assumed distance that the top deck should be moving relative to the keel should be 4.6m.

To work out the acceleration force suvat equations were used:

$$S = ut + \frac{1}{2}at^2$$

Rearranging for acceleration, with zero starting velocity so u will equal to zero and converting 6 hours to seconds:

$$a = \frac{2S}{t^2}$$

$$a = \frac{2 \times 4.6}{(6 \times 60 \times 60)^2}$$

$$a = 1.97188 \times 10^{-8}ms^{-2}$$

This acceleration force is so small it is negligible, especially when compared to gravity (9.81), however if a much shorter lift time is required then this may become important.

Force required to move the top of the ship;

$$F = ma$$

$$F = 15594 \times 1000 \times 9.81$$

$$F = 1.529 \times 10^8N$$

The total number of cylinders needed to lift this was assumed to be 300, this was then changed to 280 at a later stage because for maintenance 280 seemed more appropriate as there would be an approximate gap of 2 meters between each cylinder and 14 systems with approximately 20 per system seemed more appropriate.

The internal diameter of the cylinder is estimated to be 0.25 meters. This is variable but it seemed appropriate for maintenance, the ability to change or check the cylinders.

The total internal area of all the cylinders is then calculated.

$$A = \text{number of cylinders} \times \pi \left(\frac{d}{2} \right)^2$$

$$A = 280 \times \pi \left(\frac{0.25}{2} \right)^2$$

$$A = 13.744 \text{m}^2$$

Using the total force to move the top deck and the number of cylinders, force per cylinder can be calculated.

$$F_{\text{cylinder}} = \frac{\text{Force to move the ship}}{\text{Number of cylinders}}$$

$$F_{\text{cylinder}} = 5.463 \times 10^5 \text{N}$$

To work out the pressure required per cylinder to lift the weight of the top of the ship

$$P = \frac{F}{A}$$

$$P = \frac{1.529 \times 10^8}{13.744}$$

$$P = 1.113 \times 10^7 \text{Pa}$$

$$P_{\text{absolute}} = 1.123 \times 10^7 \text{Pa} (112.31 \text{bar})$$

To find a pump a flow rate of the hydraulic fluid needs to be found. To do this the volume being moved needs to be calculated.

$$\text{Moving volume} = \text{Area of cylidners} \times \text{moving height}$$

$$\text{Moving volume} = 13.744 \times 4.6$$

$$\text{Moving volume} = 63.225 \text{m}^3$$

To get the flow rate the moving volume should be divided by a time.

$$\text{flow rate} = \frac{\text{moving volume}}{t}$$

$$\text{flow rate} = \frac{63.225}{6}$$

$$\text{flow rate} = 10.537 \text{m}^3/\text{hr}$$

This is how the chosen pump displays there pump flow rate characteristics. The chosen pump which will be required to pump all the cylinders is Seim POF series triple screw pump.

To calculate the thickness of the cylinder wall and the thickness of the end plug a material must be selected. This material should already be used in similar situations and should be strong enough to resist the internal pressure of the cylinder. The material was selected using the EduPack software, the material is; Stainless Steel, austenitic, A151 205, wrought, annealed (F3K20S) for the reasons above it is also resistant to wear, which will be important in the marine environment.

*note: all the material properties taken from EduPack are assumed to be worst scenarios; minimum strength properties and maximum density and cost.

Material properties:

	Young modulus (GPa)	Yield strength (MPa)	Tensile strength (MPa)	Compressive strength (MPa)	Density (kg/m^3)	Cost GBP/Kg
Value	193	425	725	425	7.90×10^3	1.05

Cylinder thickness

To calculate the cylinder thickness:

$$t = \frac{P d_0}{2\sigma}$$

$P = \text{pressure}$

$d_0 = \text{cylinder internal diameter}$

$\sigma = \text{material yield strength}$

$$t = \frac{1.113 \times 10^7 \times 0.25}{2 \times 425 \times 10^6}$$

$$t = 3.303mm$$

End plug

To calculate the thickness of the end plug:

$$t = d_0 \sqrt{\frac{kP}{\sigma}}$$

$k = \text{An empirical coefficient}$

$$t = 0.25 \times \sqrt{\frac{0.162 \times 1.113 \times 10^7}{425 \times 10^6}}$$

$$t = 16.358mm$$

(Q.S.Khan)

To calculate the thickness of the piston the following calculations took place and an estimated diameter of 0.2m was chosen.

$$F_{cylinder} = \frac{\pi^2 EI}{4L^2}$$

Making I the subject:

$$I = \frac{4F_{cylinder}L^2}{\pi^2 E}$$

$$I = \frac{4 \times 5.463 \times 10^5 \times 4.6^2}{\pi^2 \times 193 \times 10^9}$$

$$I = 2.4277 \times 10^{-5} m^4$$

$$I = \frac{\pi}{64} (D^4 - (D - 2t)^4)$$

Rearranging to make t the subject.

$$\begin{aligned}\frac{64I}{\pi} &= D^4 - (D - 2t)^4 \\ D^4 - \frac{64I}{\pi} &= (D - 2t)^4 \\ D - 2t &= \sqrt[4]{D^4 - \frac{64I}{\pi}} \\ 2t &= D - \sqrt[4]{D^4 - \frac{64I}{\pi}} \\ t &= 0.5 \left(D - \sqrt[4]{D^4 - \frac{64I}{\pi}} \right) \\ t &= 0.5 \left(0.2 - \sqrt[4]{0.2^4 - \frac{64 \times 4.685 \times 10^{-5}}{\pi}} \right) \\ t &= 8.8296 \text{ mm}\end{aligned}$$

Checking this thickness will not buck with Euler buckling.

$$\begin{aligned}A &= \frac{\pi}{4} (D^2 - (D - 2t)^2) \\ A &= \frac{\pi}{4} (D^2 - (D - 2t)^2) \\ A &= \frac{\pi}{4} (0.2^2 - (0.2 - 2 \times 8.8296 \times 10^{-3})^2) \\ A &= 2.7127 \times 10^{-3} \text{ m}^2 \\ \sigma_E &= \frac{F_{cylinder}}{A} \\ \sigma_E &= \frac{5.463 \times 10^5}{2.7127 \times 10^{-3}} \\ \sigma_E &= 2.014 \times 10^8\end{aligned}$$

So the safety factor against buckling is $= \frac{\text{Yield strength}}{\sigma_E}$ which gives a safety factor against buckling of 2.11.

Using the thicknesses and the heights of the system the total weight of a unit would be 350.25Kg. So the total weight of the whole system will be 98.070 tonnes. The total cost of the units would be £102,973

Bibliography

Q.S.Khan. (n.d.). Design of a Hydraulic Cylinder. In Q.S.Khan, Volume-2. Design and Manufacturing of Hydraulic Cylinders (pp. 20-28). Mumbai: TANVEER PUBLICATIONS.

Appendix II to Annex E – Engineering Requirements (Material Selection)

When designing the hydraulics system material selection was essential. It needed to be able to cope with the required forces with a very good durability against water and oil. In addition, cost and weight should be limited if possible. CES EduPack software was used to find an acceptable material with these requirements. Criteria was set on the software in order to reduce the available material for selection. The criterial set were as follows:

Property	Value
Durability in water (sea)	Excellent
Durability in water (fresh)	Excellent
Organic solvents	Excellent

The composition was also limited to non-honeycomb, powder, foam and wire as these will not resist the expected forces. In the end the selected material was Stainless Steel, austenitic, A151 205, wrought, annealed. This was selected because as well as meeting the above properties it has a very high strength (see table below). It is also non-flammable, and can be manufactured by hot metal forming with a relatively low cost.

Physical property	Value
Tensile strength	745 MPa
Young's modulus	193 GPa
Yield strength	425 MPa
Compressive strength	425 MPa

Appendix III to Annex E – Engineering Requirements (Pump Selection)



60 Frame Plunger Pump

Nickel Aluminum
Bronze Model

6747

FEATURES

- Triplex plunger design delivers high efficiency and low pulsation.
- Durable dual V-Packings are lubricated and cooled by pumped liquid.
- Pre-set Lo-Pressure Seals provide secondary protection against external leaks and require no packing adjustment.
- Special high-density, polished, concentric plungers provide a true wear surface and extended seal life.
- Specially formulated Seals and V-Packings offer unmatched performance and extended life.
- Wet-end easily serviced without entering crankcase.

SPECIFICATIONS

	U.S. Measure	Metric Measure
Flow	48 gpm	(182 l/m)
Pressure Range	100 to 2000 psi	(7 to 140 bar)
RPM	615 rpm	(615 rpm)
Inlet Pressure Range	Flooded to 50 psi	(Flooded to 3.5 bar)
Bore	1.772"	(45 mm)
Stroke	2.579"	(65.5 mm)
Crankcase Capacity	10 Qts.	(9.46 l)
Maximum Liquid Temperature	140°F	(60°C)
Above 130°F call CAT PUMPS for inlet conditions and elastomer recommendations.		
Inlet Ports (2)	2" NPTF	(2" NPTF)
Discharge Ports (2)	1-1/4" NPTF	(1-1/4" NPTF)
Shaft Diameter	1.772"	(45 mm)
Weight	335 lbs.	(151.95 kg)
Dimensions	31.14 x 24.37 x 12.87" (791 x 619 x 327 mm)	

ELECTRIC HORSEPOWER REQUIREMENTS

FLOW		PRESSURE				RPM
		psi 900	psi 1000	psi 1200	psi 2000	
U.S. gpm	l/m	bar 60	bar 70	bar 85	bar 140	
48	182	29.6	32.9	39.4	65.8	615

CAUTIONS AND WARNINGS

All High Pressure Systems require a primary pressure regulating device (i.e. regulator/unloader) and a secondary pressure relief device (i.e. pop-off valve, relief valve). Failure to install such relief devices could result in personal injury or damage to pump or property. CAT PUMPS does not assume any liability or responsibility for the operation of a customer's High pressure system.

Read all CAUTIONS and WARNINGS before commencing service or operation of any high pressure system. The CAUTIONS and WARNINGS are included in each service manual and with each Data sheet. CAUTIONS and WARNINGS can also be viewed online at www.catpumps.com/cautions-warnings or can be requested directly from CAT PUMPS.

DETERMINING THE PUMP R.P.M.	Rated gpm	=	"Desired" gpm
	Rated rpm	=	"Desired" rpm
DETERMINING THE REQUIRED H.P.	gpm x psi	=	Electric Brake
	1480	=	H. P. Required
DETERMINING MOTOR PULLEY SIZE	Motor Pulley O.D.	=	Pump Pulley O.D.
	Pump rpm	=	Motor rpm

See complete Drive Packages (include Pulleys, Belts, Hubs, Key) Tech Bulletin 608.
Refer to pump Service Manual for repair procedure and additional technical information.



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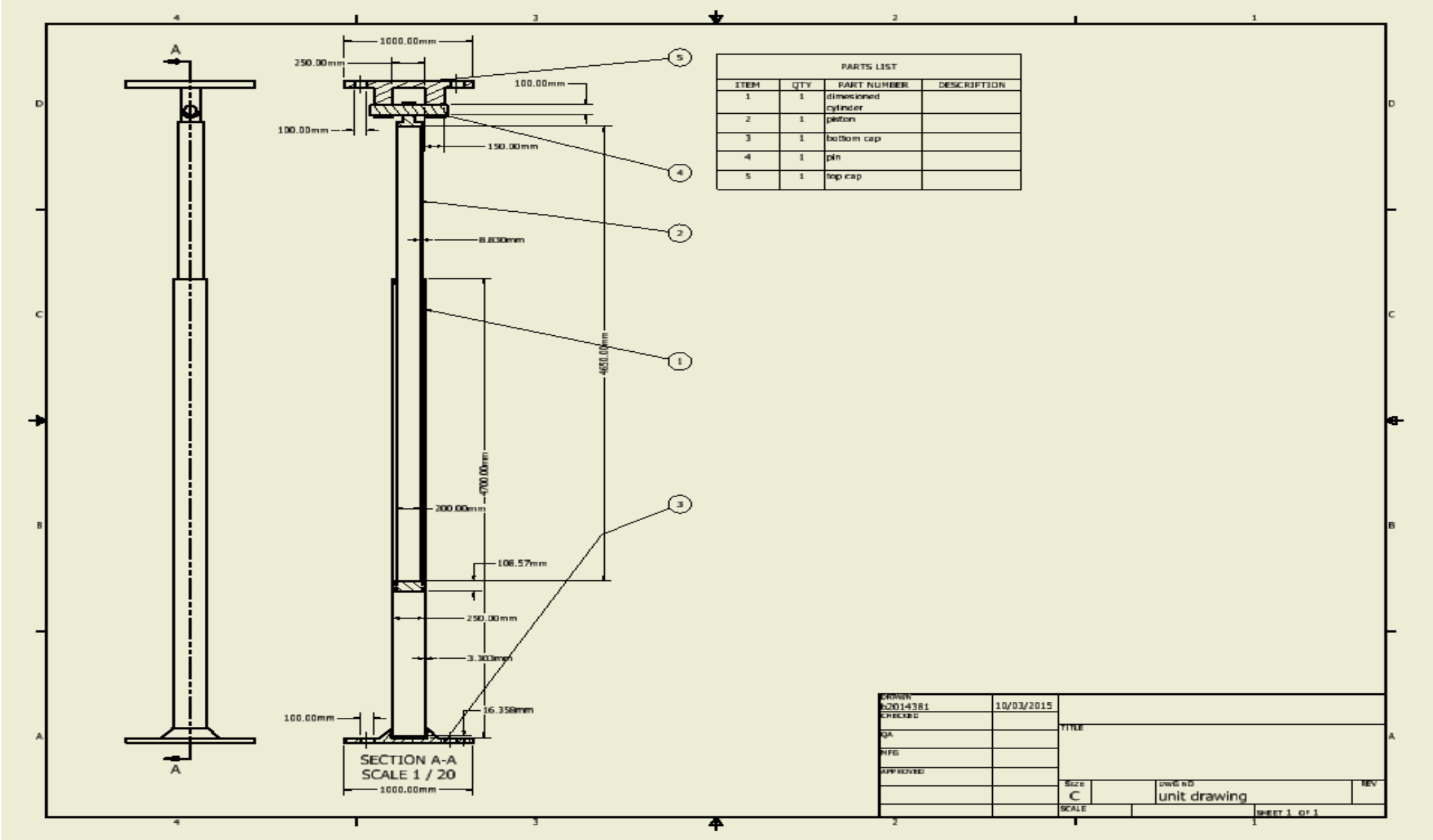
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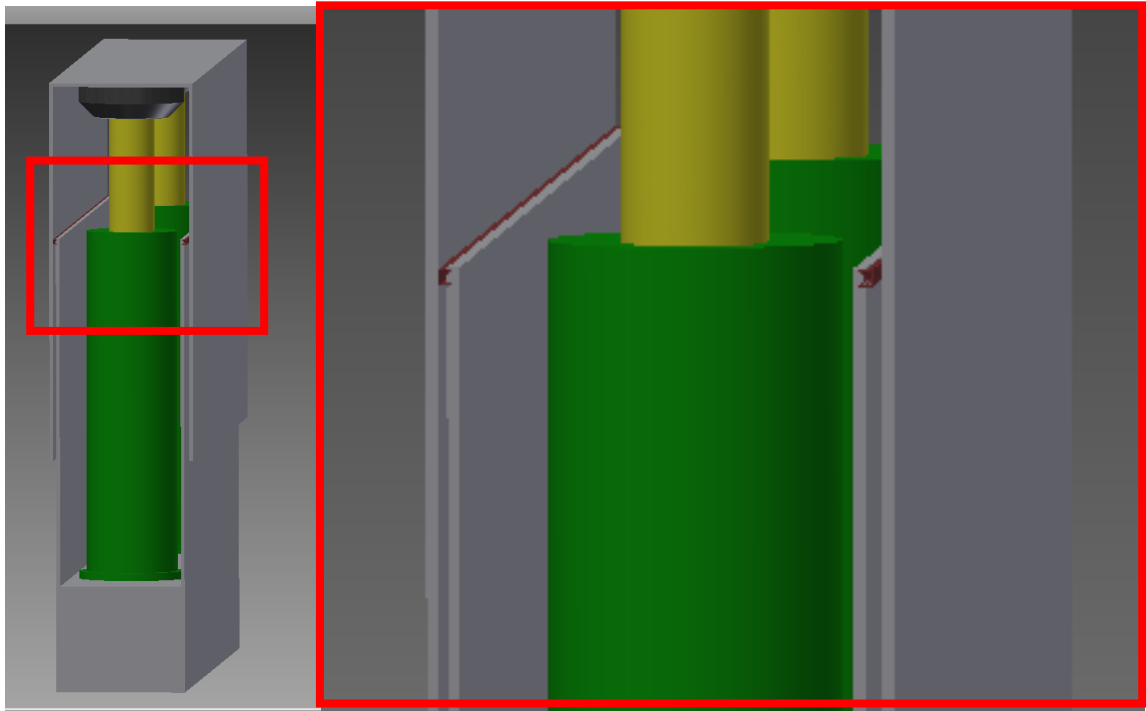
Appendix IV to Annex E – Engineering Requirements (Cylinder Drawing)



Annex F – Water Tight Integrity

Initial Parameters

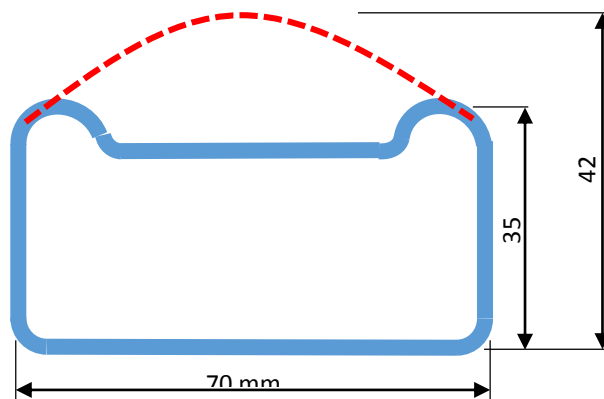
It is sensible to assume that the ship should have two water tight seals so that one was used as a backup in case one were to fail. The locations of these seals were at the top of the stationary bulkheads on either side as shown below.



The volume of the seals has to be found in order to calculate the required capacity of the system. The ship is divided into 13 sections and each system will have a dedicated supply. This means that initial calculations should be carried out to determine the requirements of each systems.

1) Volume required to be filled by each system

To calculate the volume, the size of the seals had to be determined. Using standard seals from the company Sealing Projex, the cross sectional area could be calculated.



The seal shown above measures 70mm x 35mm in the collapsed case, using an inflation of 20% which is a reasonable assumption, this gives an extended dimension of 70mm x 42mm and thus giving a required area to be filled with gas of 0.00294m^2 .

The length of the ship that will be covered by the ship can be estimated by the following procedures:

- 5 bulkheads at 30m each: $30 \times 5 = 150\text{m}$
- Length of ship times 2 for a rough perimeter: $180 \times 2 = 360\text{m}$
- Add a reasonable excess of 60m: $360 + 150 + 60 = 600\text{m}$

This value of 600m must now be multiplied by two, as there are two seals running around the ship.

$$600 \times 2 = 1200\text{m}$$

The cross sectional area of the seal (0.00294m^2) can now be multiplied by the length of the system (1200m) in order to calculate the volume of the seals.

$$1200\text{m} \times 0.00294\text{m} = 3.528\text{m}^3$$

This value in m^3 can now be converted into litres to allow for more conventional readings giving a value of 3528 litres, which should be rounded to a value of 3600 litres. This volume is the volume required to be filled around the whole ship and therefore should be divided by 13 to find the required volume to be serviced by each system.

$$3600 \div 13 = 277 \text{ litres}$$

The capacity of a compressor is measured in Cubic Feet per Minute (CFM), and is a measure of the amount of air that the compressor can pump out. This volume of 277 litres needs to be transferred into a flow rate of CFM. The air needs to be supplied as quickly as possible, so it is reasonable to state a flow rate of 277 litres per second.

$$277 \text{ litres per second} = 16620 \text{ litres per minute}$$

Transferring this to CFM:

$$16620 \text{ litres per minute} = 590 \text{ CFM}$$

2) Pressure of each system

From research into the area of inflatable seals, it is found that these seals are filled to a pressure of 8 bar. This is very convenient as most rotary screw air compressors are designed to work at 7-15 bar.

3) Type of Air Compressor

The final thing to consider when picking an air compressor is the type. There are two main types of compressor and these are:

- a) Reciprocating (piston) air compressors
- b) Rotary Screw air compressors

a) **Reciprocating (piston) air compressors**

These compressors compress air with the use of one or more cylinders/pistons, with the pistons reciprocating inside the cylinders compressing the air within the cylinders.

These are for low to very high pressure systems (up to 1000 bar), they are low capacity and are designed for intermittent use. This is not a good type of compressor for our project, as a capacity of 590 CFM is very high, and therefore a reciprocating air compressor would not cope with the needs of our system.

b) Rotary Screw air compressors

These compressors compress air by two screws that rotate in opposite directions, trapping and compressing air in these screws.



Rotary Screw Compressors work at low pressures (7-15 bar), at high capacity and are designed for continuous use. These reasons make this compressor more appealing and therefore a Rotary Screw Air Compressor should be used.

Required Parameters and compressor selection

After calculating the requirements for the air compressor, one should be found to meet the needs of the system.

Parameter	Value	
Type of Compressor	Rotary Screw Compressor	
Capacity	590 CFM	
Pressure	8 bar	116 psi

A compressor that meets all of these requirements is the 135 horsepower variable frequency drive rotary screw compressor from Almig. Its properties are given below:

Parameter	Value
Manufacturer	Almig
Type of Compressor	Rotary Screw Compressor
Capacity	152 – 602 CFM
Pressure	75 – 190 psi



The price of one of these compressors is \$60,500 and as 13 are needed, this would cost \$786,500. The weight of one compressor is 4510 kg, resulting in an addition of weight across the ship of 58,630kg.

4) **Material Selection**

The material of the seal requires significant thought as this will be in contact with the elements of the sea and it is very important that this will not perish or degrade. However, at the same time the chosen material must be able to expand with the compressed air system and provide a watertight seal around the ship.

In order to find such a material to cope with these conditions, CES EduPack a materials information software. In order to find the correct material there is set out criteria that the software then used to search the database for a suitable solution.

Material Properties

The material that would be used for the seals must be resistant to both fresh and salt water as at this stage it is unrealistic to determine if the vessel will operate in just salt, or just fresh water conditions or both. The seal may come into contact with some form of lubrication oil, or grease during operation so this must be accounted for, and finally during manufacture of the seal, it is likely that the product will be injection moulded as this is the most common form of seal production. These properties were then selected within the CES EduPack software.

Property	Value
Durability in Water (Fresh)	Excellent
Durability in Water (salt)	Excellent
Durability in oils and fuels	Excellent
Ability to be Injection Moulded	Excellent

After applying these properties to the software I was left with a selection of thermoplastics, and thermoset elastomers. Knowing that the thermoset elastomers are rubber I was left with Perfluoro elastomer. This is used for seals in extreme environments such as offshore oil production. This made me confident in the choice of material with respect to its current operational use.

Annex G – Power Requirement and Distribution

Appendix I to Annex G – Empirical Power Requirements

Since Seahorse 35 has an installed power of 7.5MW, it is a safe assumption that with our additional mass on board due to hydraulics and additional steel, and therefore we have chosen to install 8.125MW of power. The installed power is generally 25% more than the required power (15% for sea margin and 10% for engine margin). Due to that we have found that the minimum required power for our ship is 6.5MW. Since power and resistance are related by the formula:

$$P = VR$$

It is possible to find the resistance of the ship to be 464.29KW at 14 knots.

Frictional resistance accounts for roughly 45% (according to the ITTC57 method values found on similar ships) of the power requirement in the collapsed case was assumed to be in the formula:

$$P_{Collapsed} = P_{Expanded} \frac{T_{Collapsed}}{T_{Expanded}} \times 0.45$$

From this we have found that in the collapsed case the ship will only require 3.25MW of power. This has led to a 50.06% drop in power required which is then directly proportional to the fuel consumption which will be reduced as well.

Since it is essential to design the ship to the worst scenario, we must use the expanded power requirements for propeller design. Firstly the wake and thrust coefficient must be found:

$$w = w_1 + w_2 + w_3$$

$$w_1 = a + \frac{b}{c(0.98-\delta)^3+1}$$

$$a = 0.1 \frac{B}{L} + 0.149$$

$$w_2 = \frac{0.025 F_a}{100(\delta-0.7)^2+1}$$

$$b = 0.05 \frac{B}{L} + 0.449$$

$$w_3 = -0.18 + \frac{0.00756}{D/L+0.002}$$

$$c = 585 - 5027 \frac{B}{L} + 11700 \left(\frac{B}{L}\right)^2$$

$$t = t_1 + t_2 + t_3$$

$$t_1 = d + \frac{e}{f(0.98-\delta)^3+1}$$

$$d = 0.625 \frac{B}{L} + 0.08$$

$$t_2 = -0.01 F_a$$

$$e = 0.165 - 0.25 \frac{B}{L}$$

$$t_3 = 2 \left(\frac{D}{L} - 0.04\right)$$

$$f = 825 - 8060 \frac{B}{L} + 20300 \left(\frac{B}{L}\right)^2$$

Where F_a is the form at the aft end, where a U shaped hull is 3 and a V shaped hull is -3.

This is used to find the apparent velocity, V_a , using the formula:

$$V_a = V_s(1 - w)$$

This was found to be 4.71 at 14 knots in the expanded case.

Another parameter to calculate is the area ratio A_E/A_O which is found using the midpoint of the inequality:

$$0.25 + 0.05(z - 2) < A_E/A_O < 1.10 - 0.075(z - 5)^2$$

Where z is the blade number between 3 and 7 inclusively. A_E/A_O was found to be 0.64. The propeller is assumed to be similar to that of Seahorse 35 and so is 5.2m

Appendix II to Annex G – Propeller Requirements

From the resistance found earlier, it is possible to find the thrust and therefore the thrust loading coefficient:

$$C_{Th} = \frac{T}{\frac{1}{2}\rho V_a^2 \frac{\pi}{4} D^2}$$

Using the thrust loading coefficient, we can finally get the advanced coefficient, J, and the pitch ratio, P/D.

$$P/D = (0.0144z^2 - 0.04z + 0.335)C_{Th}^{(0.025(z-0.35)^2-0.6)} - 0.0025(z-4)^4 + 0.01z + 0.53 + 0.1(A_E/A_O)^2$$

The pitch of the propeller is then found through:

$$P = D^{P/D}$$

$$J = \left((0.0183(7.5 - z)^2 - 0.2) \left(A_E/A_O - 1.1 \right) + 0.75 \right) C_{Th}^{-0.44} + (0.12 - 0.028(z - 5.2)^2) (A_E/A_O - 1)$$

From this value of J and the knowledge of the formula below, it is easily possible to find the rotational speed of the propeller in rpm:

$$J = \frac{V_a}{nD}$$

$$n = \frac{V_a}{JD}$$

Annex H – Cost Saving calculations

Based on the Fuel priced at 1600GMT on 14th April 2015, the cost of 1 tonne of fuel oil is \$304.5. The engine selected for the CBDD Ship could be the MAN S46ME-B8. At 75% load the engine requires 167g/KWh and produces 6.210MW. Assuming that the loading of the ship occurs for 3 days in a 2 week period, a 5 day maintenance period per year and the rest the ship is traveling at sea. The total fuel cost per year can be calculated at \$2.145M USD per year. Assuming a 2% inflation in the price of fuel over a 25 year ships lifespan means the fuel total cost is estimated at \$53.62M USD. With a saving of 25% required power, the total saving due to fuel is in the region of \$13.4M USD.

cost of Fuel	SFC @ 75%	Days in Port per year	Engine power @ 75% load	Days at sea	hours at sea	fuel consumption	cost per year
304.5	167	82	6210	283	6792	7,043.78	2,144,830.84
\$/mt	g/Kwh	days	Kw			tonnes per year	USD
Estimated average interest rate							
2.00%							
Estimated overall cost of fuel in its lifetime							
Lifespan of Ship							
25							
years							
Required propulsion Power saving							
25.03%							